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Multi-objective optimisation of vapour-compression refrigeration systems using Microsoft Excel

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ABSTRACT

This paper presents a methodology for using a general-purpose spreadsheet application, which is Microsoft Excel, as an effective modelling platform for multi-objective optimisation analyses of vapour-compression refrigeration (VCR) systems. Since Excel itself does not provide functions for determining the thermodynamic properties of the refrigerants, the VBA programming language that comes with Microsoft applications has been used for developing such functions for various conventional and alternative refrigerants. The other limitation of Excel regarding the intended purpose is that the solver that comes with it (Frontline's Solver) is not suitable for multi-objective analyses. Fortunately, the developers of the MIDACO solver have made a limited version of this multi-objective solver available for Excel users for free. The paper demonstrates the capability of Excel with these additional capabilities for exergetic, economic, and environmental (3E) analyses of VCR systems by considering a simple system with the environment-friendly R-152a as the refrigerant.

Keywords: *Multi-Objective Optimisation, Vapour-Compression Refrigeration Systems, Microsoft Excel, MIDACO*

1. Introduction

The cooling demand around the world consumes about 15% of electricity production and contributes by around 10% to the global greenhouse gas emissions [1]. Therefore, the synthetic refrigerants with high global-warming and ozone-layer depletion potentials have to be phased out and replaced by more benign ones. In this respect, computer aided models play the important role of scrutinising the potential fluids to identify the most suitable ones. Since the energy consumed during system operation accounts for the majority of their carbon emissions, maintaining or improving the efficiency of refrigeration and air-conditioning systems is an important consideration when developing the equipment for refrigerants with low global warming potential (GWP) [2]. The characteristics of low GWP refrigerants may require design changes that could significantly raise the costs of VCR systems and/or undermine their efficiency. In this respect, computer-aided models provide a safe, low-cost, and accurate method for initial assessment of the efficiency and economics of the modified refrigeration systems.

A considerable number of publications in the last two decades dealt with computer-aided analyses of VCR systems using synthetic and natural refrigerants. Arshad [3] developed a model for exergy analysis of the simple VCR system with refrigerants R-12, R-22, and R-407C by using REFPROP software [4]. He analysed the performance of these three refrigerants by varying the evaporator temperature and showed that the second law efficiency increased while the exergy destruction decreased with the increase in evaporator temperature. Exergy loss for R-12 and R-22 were almost the same, but that for R-407C was less. Exergetic efficiency was maximum for R-12 followed by R-22 and then R-407C. Karakurt et al. [5] also analysed the performance of the simple VCR system, but using R-152a, R-134a and R-290. By using the Engineering Equation Solver (EES) [6] to determine the refrigerants' properties, they studied the effects of sub-cooling and superheating on the system's performance in terms of coefficient of performance (COP), exergetic efficiency, exergetic performance coefficient, and ecological coefficient of performance (ECOP). Their results showed that the COP, exergetic, and ECOP parameters of the refrigeration system strongly depend on a number of operational and design factors that include the condenser and evaporator temperatures, pressure losses in heat exchangers and isentropic efficiency of compressor.

Ahamed et al. [7] studied the energy and exergy performances of a refrigerator using R-134a as a baseline with its performances using butane and isobutene.

Exergy losses in the individual components were obtained at different evaporating and condensing temperatures from experimental data. They obtained the properties of the refrigerants by using the REFPROP software. Their analyses showed that the exergy efficiency of isobutene was 50% higher than that of R-134a. They also found that the exergy losses were minimal for the refrigerants in the four components at the higher evaporating temperatures and that the maximum exergy loss of about 60% occurred in the compressor. Gulloa et al. [8] also investigated the use of a natural refrigerant, which is carbon dioxide (R-744), for air-conditioning in warm climates in order to assess its real potential in such weather conditions. By using EES, they analysed a refrigeration system with parallel compression as one of the proposed modifications to enhance the performance of a single-stage refrigeration system. They compared the thermodynamic efficiency and the final cost of the product for the modified refrigeration system that uses the auxiliary compressor with that of a conventional system, both of them operating in transcritical conditions. Their results showed that the adoption of an auxiliary compressor resulted in an increase of the COP by approximately 18.7% over the investigated temperatures range and the final cost of the product associated with this solution was on average 6.7% lower than that of the conventional system.

Kadam et al. [1] investigated a VCR system with R-134a as the refrigerant which is used in an actual/existing district cooling system (DCS) and studied the feasibility of replacing the synthetic refrigerant with a natural fluid, which is ammonia (R-717). They implemented their model by using the ASPEN Plus^{TM} process simulator [9] with R-134a or R-717 as the primary working fluid (refrigerant) and water as the secondary fluid. To determine the fluid properties, they selected the REFPROP model. By assessing the impact of the evaporation temperature on the total equivalent warming impact (TEWI), coefficient of performance (COP), exergy efficiency and cost rate, it was observed that the COP and exergy efficiency of the VCR system with R-717 was around 3 % higher compared to that with R-134a. They also observed that the total global warming impact and combined cost of R-717 were lower by 1.6 % compared to R-134a.

Sun et al. [10] developed computer-aided models using MATLAB for a single-stage VCR system with an economiser (SSRS+E), a two-stage VCR system (TSRS), and a cascade VCR system (CRS) to conduct an energetic and economic (2E) analysis of these systems using R-744 and R-717. The data for fluid properties was obtained by using REFPROP. The corresponding optimal inter-stage pressure was calculated by iteration at each evaporating temperature and the rest of the system parameters were calculated based on the optimal inter-stage pressure. Their results showed that SSRS+E could save energy by 13.6% and 7.1% compared to TSRS at evaporating temperatures of -20°C and -25°C, respectively. The R744/R717 CRS was found to be superior to TSRS in terms of energy consumption and refrigeration unit investment costs. Compared to TSRS, R744/R717 CRS could save energy by 14.1% and 18.8%, at the evaporating temperatures of -45°C and -50°C, respectively. Based on their analysis, SSRS+E was recommended for use for evaporating temperatures above -25°C, TSRS was recommended for use in the evaporating temperature range of -45°C to -25°C, and R744/R717 CRS was recommended for use at the evaporating temperatures below -45°C.

As the previous review of the literature shows, most of the previous computer-aided studies focussed on analysing the thermodynamic (energetic and/or exergetic) and economic performance of VCR systems. However, the increasing concerns about global environment change nowadays make the environment factor as important as the economic factor [1]. Most research also analysed the performance of VCR systems without using multi-objective optimisation (MOO) techniques to determine the optimum evaporator/condenser temperatures or pressures. However, these techniques are needed for evaluating the performance of modified VCR systems compared to conventional systems from energetic, economic, and environmental viewpoints. For their analyses, most researchers also used commercial and dedicated software, but the use of general-purpose software allows more researchers and engineering students to join the search for environment-friendly refrigerants and contribute to the development of innovative refrigeration systems. This paper demonstrates the use of the widely-available Microsoft Excel as a modelling platform for MOO analyses of VCR systems. The refrigerant selected for this purpose is R-152a for its superior performance efficiency and shorter atmospheric life-time [5,11].

While Excel itself is equipped with numerous mathematical functions, the Solver add-in [12] that comes with it enables single-objective optimisation (SOO) analyses to be conducted by using both gradient-based and evolutionary methods. For MOO analyses, the free-to-download version of the MIDACO solver [13] is used. The refrigerant's properties can be obtained by using the various Excel add-ins developed by the academic institutions and individual researchers or by using VBA to develop the functions. The Excel-aided model presented in this paper uses the Thermax add-in [14] to determine the refrigerant properties and uses the data provided by Roy and Mandal [11] for validating the relevant functions. By taking advantage of these two added features, the paper shows that the Excel-based platform is capable of MOO analyses of VCR systems.

2. The Analytical Model for the VCR System

Figure 1 shows schematic and *T-s* diagrams of the simple VCR system in which the refrigerant leaves the evaporator as dry saturated vapour without superheating and leaves the condenser as dry saturated liquid without subcooling. Roy and Mandal [11] conducted a thermo-economic assessment of this system using three low GWP refrigerants which are R-152a, R-1234ze, and R-600a. By comparing the optimum results obtained for these refrigerants they showed that the refrigerant R-152a offers the best performance. In the present analysis only R-152a will be considered and the basic data for the analysis, as given by Roy and Mandal [11], are shown on Table 1.

Thermodynamic model

Basic assumptions:

- Steady operation
- Changes in kinetic and potential energies are negligible.
- Adiabatic compression and throttling processes.
- Piping pressure losses are ignored.

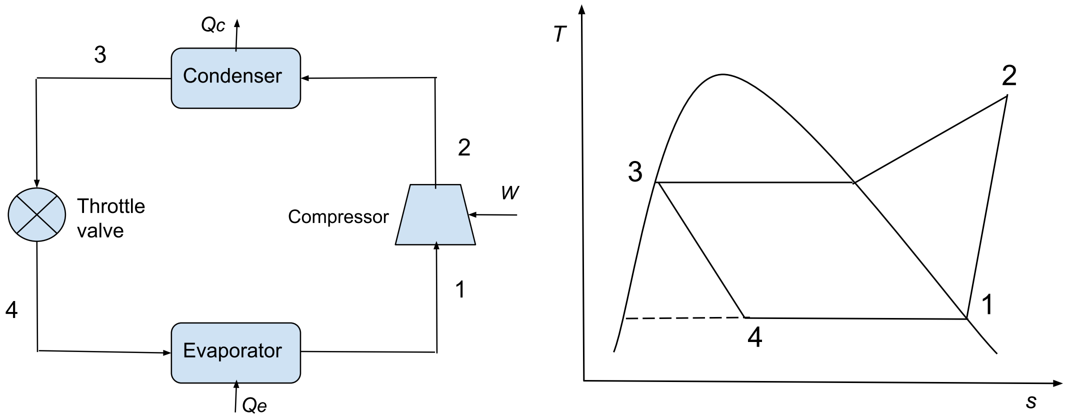
Given the cooling capacity (*CC*) of the system, in kW, the mass flow rate of the refrigerant, in kg/s, is determined from;

$$\dot{m} = \frac{CC}{h_1 - h_4} \tag{1}$$

The compression work and heat rejection in the condenser are given by:

$$\dot{W}_c = \dot{m}(h_2 - h_1) \tag{2}$$

$$\dot{Q}_{con} = \dot{m}(h_2 - h_3) \tag{3}$$



(a) (b)

Figure 1: (a) Schematic and (b) *T-s* diagram of the simple VCR system.

Table 1. Assumed input parameters of the simple system [12]

Parameter	Value
Cooling capacity of the system, CC	10 kW
Isentropic efficiency of compressor, η_c	80%
Overall heat transfer coefficient for evaporator, U_{eva}	0.03 kW/m^2.K
Overall heat transfer coefficient for condenser, U_{con}	0.04 kW/m^2.K
Ambient temperature, T_0	25^{o}C
Temperature change for air in evaporator and condenser	± 5^{o}C
Temperature of the inlet air to evaporator	0^{o}C
Maintenance factor, φ	1.06
Interest rate, i	14%
Plant life time, n	15 years
Annual operation hours, N	4266 hours
Electrical power cost, c_{elec}	0.09 \$/kWh
Emission factor, μ_{CO_2E}	0.968 kg/kWh
Cost of CO_2 *avoided,* c_{CO_2}	0.09 \$/kg of CO_2 emission

The coefficient of performance (COP) is defined as:

$$COP = \frac{CC}{W_c} \tag{4}$$

Adiabatic throttling is a constant-enthalpy process and, therefore:

$$h_4 = h_3 \tag{5}$$

The rate of exergy destruction in the four system components are determined as follows:

Evaporator:

$$\dot{E}_D^{Eva} = T_0\left[\dot{m}(s_1 - s_4) - \frac{CC}{T_E}\right] \tag{6}$$

Compressor:

$$\dot{E}_D^{Com} = \dot{m}T_0(s_2 - s_1) \tag{7}$$

Condenser:

$$\dot{E}_D^{Con} = T_0\left[\dot{m}(s_3 - s_2) + \frac{\dot{Q}_{con}}{T_C}\right] \tag{8}$$

Throttling valve:

$$\dot{E}_D^{TV} = \dot{m}T_0(s_4 - s_3) \tag{9}$$

The overall exergetic efficiency of the system is defined as:

$$\varepsilon = \left(\dot{W}_c - \dot{E}_D^{Tot}\right)/\dot{W}_c \tag{10}$$

Where is the total rate of exergy destruction in the system:

$$\dot{E}_D^{Tot} = \dot{E}_D^{Eva} + \dot{E}_D^{Com} + \dot{E}_D^{Con} + \dot{E}_D^{TV} \tag{11}$$

Economic model

The total annualised cost rate of the system is given by [11]:

$$\dot{C}_{total} = \sum \dot{C}_k + \dot{C}_{op} + \dot{C}_{env} \tag{12}$$

where, $\dot{C}_k$ is the capital investment and maintenance cost rate of individual components, $\dot{C}_{opt}$ is the operating cost rate of the system, and $\dot{C}_{env}$ is the CO_2 penalty cost rate of the system. The capital investment and maintenance cost rate of the individual component is given by:

$$\dot{C}_k = C_k.\phi.CRF \qquad c_{CO_2} \tag{13}$$

where, C_k is the capital investment of the component, φ is the maintenance factor, and CRF is the capital recovery factor as obtained from:

$$CRF = i(1+i)^n / \left[(1+i)^n - 1\right] \tag{14}$$

where i is the interest rate and n is the system's expected lifetime. The capital investment of the system components are estimated using the relations shown on Table 2.

Table 2. Capital cost functions of the different components [11]

Component	Capital cost function
Evaporator	$C_{eva} = 1397 \times A_{eva}^{0.89}$
Compressor	$C_{eva} = 10167.5 \times A_{com}^{0.46}$
Condenser	$C_{con} = 1397 \times A_{con}^{0.89}$
Throttle valve	$C_{TV} = 114.5 \times \dot{m}$

Operational cost rate of the system is the cost of electricity given by:

$$\dot{C}_{op} = \dot{W}.N.c_{elec} \tag{15}$$

where N is the annual operational hours and c_{elec} is the cost of electricity in \$/kWh. Following Wang et al. [14], the CO_2 penalty cost rate of the systems is calculated from:

$$\dot{C}_{env} = m_{CO_2e}.c_{CO_2} \tag{16}$$

where, is the penalty cost of the avoided CO_2 emission and is the amount of annual CO_2 emission from the system that can be estimated from:

$$m_{CO_2e} = \mu_{CO_2e} . E_{annual} \tag{17}$$

where is the emission factor and is the annual amount of energy consumed by the system. The values of *i*, *n*, *N*, μ_{CO_2e} , c_{elec}, and E_{annual} used in the analysis of the present systems are given in Table 1.

Total equivalent warming impact (TEWI)

Refrigerants have a direct global warming effect which results from the refrigerants being directly released or leaked into the atmosphere. They also have an indirect effect caused by the CO_2 emissions in thermal power plants that use fossil fuels to produce the energy needed for driving the refrigeration systems. TEWI is a non-monetary measure that evaluates the direct and indirect global warming effects of the refrigeration systems. It is calculated for different refrigerants using the following correlation [1].

$$TEWI = GWP_{ref}\left[m_{ref} \times L_{annual} \times n + m_{ref} \times (1-\alpha)\right] + (E_{annual} \times \beta \times n) \tag{18}$$

Where GWP_{Ref} is the GWP of the refrigerant, n is the system lifetime, m_{Ref} is the total refrigerant charge, L_{annual} is the refrigerant leakage rate, a is the recycling factor, E_{annual} is the energy consumed per year, and β is the electricity regional conversion factor. The comparison of TEWI provides a clear image of the global-warming effects during the service lifetime of the refrigeration system. Table 3 shows how m_{ref} and L_{annual} are calculated and gives the values of α, β, and GWP_{ref} for R-152a [1].

Table 3. TEWI analysis assumptions [1]

Parameter	m_{ref}	L_{annual}	α	β	GWP_{ref}
Assumed value	$\dot{m}_{ref}(240s)$	12.5	0.7	0.65	140

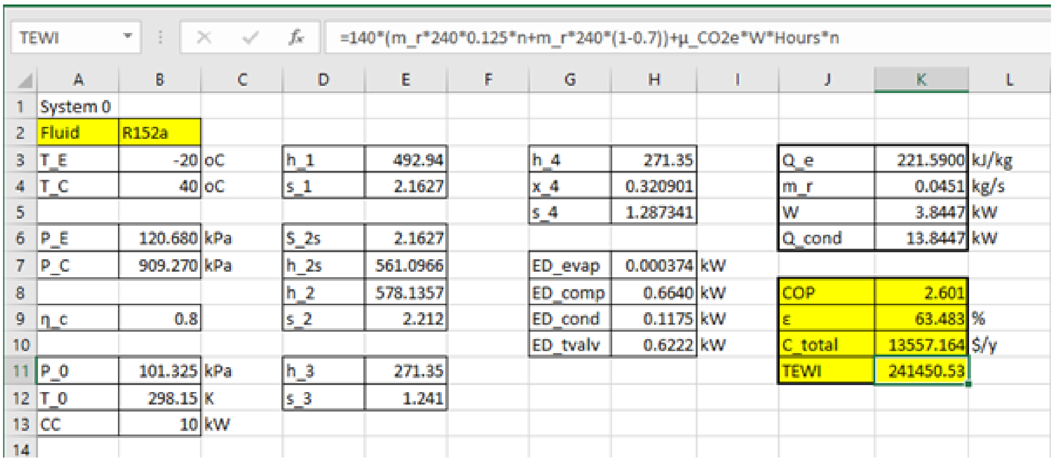
The MIDACO solver

MIDACO is a general-purpose software for solving mathematical optimisation problems [11]. The software implements an extended ant colony optimisation algorithm, which is a heuristic method that stochastically approximates a solution to the mathematical problem that is treated as black box. Although the capability of the free version is limited to four changing variables, it is still adequate for MOO analyses of simple, multi-stage compression, and cascade refrigeration systems. As a multi-objective solver, MIDACO does not give a single optimum solution like Solver, but produces a Pareto front that contains all the un-dominated optimum solutions. The best optimum solution is automatically selected by MIDACO.

3. Development and validation of the Excel-aided model

Figure 2 shows the first sheet of the Excel-aided model that performs the thermodynamic calculations. The sheet stores the name of the refrigerant in cell B2 as a variable (Fluid) so that it can be used to analyse the performance of the simple VCR system with alternative refrigerants by simply adjusting the name of the refrigerant in cell B2 from "R152a" to the intended fluid. The first block of cells on the left stores the specified data relevant to the thermodynamic model of the system as given in Table 1. The second and third blocks of cells in the sheet calculate enthalpy and entropy values at the four states and the rates of exergy destruction in the four components. The fourth block of cells determines the overall system parameters such as the refrigerant mass flow rate, the compressor power, etc. The formula bar reveals the formula that applies Equation (18) to calculate the system's TEWI. The value of the total cost rate (C_total) shown on Figure 2 is copied from sheet 2 that performs the economic calculations.

Figure 3 shows the second sheet of the model that performs the economic calculations. The data part of this sheet stores the various economic factors, the overall heat-transfer coefficients of the evaporator and condenser, temperatures and temperature changes of air entering the evaporator and condenser, cost of electricity, and CO_2 emission penalty. Based on these values the sheet determines the annualised costs of the four components and then determines the annualised costs of equipment, the annual cost of electricity, the annual penalty cost of CO_2 emissions, and the grand total of the system (C_total_an).



TEWI | =140*(m_r*240*0.125*n+m_r*240*(1-0.7))+μ_CO2e*W*Hours*n

	A	B	C	D	E	F	G	H	I	J	K	L
1	System 0											
2	Fluid	R152a										
3	T_E	-20	oC	h_1	492.94		h_4	271.35		Q_e	221.5900	kJ/kg
4	T_C	40	oC	s_1	2.1627		x_4	0.320901		m_r	0.0451	kg/s
5							s_4	1.287341		W	3.8447	kW
6	P_E	120.680	kPa	S_2s	2.1627					Q_cond	13.8447	kW
7	P_C	909.270	kPa	h_2s	561.0966		ED_evap	0.000374	kW			
8				h_2	578.1357		ED_comp	0.6640	kW	COP	2.601	
9	η_c	0.8		s_2	2.212		ED_cond	0.1175	kW	ε	63.483	%
10							ED_tvalv	0.6222	kW	C_total	13557.164	$/y
11	P_0	101.325	kPa	h_3	271.35					TEWI	241450.53	
12	T_0	298.15	K	s_3	1.241							
13	CC	10	kW									
14												

Figure 2: Sheet 1 of the Excel-aided model for the simple VCR system.

C_total_an | =C_eqip_an+C_elec_an+C_CO2e_an

	A	B	C	D	E	F	G	H	I	J	K	L
1												
2	SV	0		T_E	-20.000	oC	T_C	40.000	oC			
3	n	15										
4	i	0.14		PEC_com	18890.93		C_comp	3075.613		Z_comp	0.764217	
5	PWF	6.142168		PEC_evp	19360.10		C_evp	3151.998		Z_evp	0.78320	
6	CRF	0.162809		PEC_con	27170.77		C_con	4423.645		Z_con	1.09917	
7	ϕ	1.06		PEC_tval	5.17		C_tval	0.841267		Z_tval	0.00021	
8	Hours	4266										
9	U_eva	0.03		Evaporator			Condenser					
10	U_cond	0.04		ΔT_1	20.000		ΔT_1	15.000		C_eqip_an	10652.097	$/y
11	Tairin_eve	0	oC	ΔT_2	15.000		ΔT_2	10.000		C_elec_an	1476.152	$/y
12	Tairin_con	25	oC	LMTD_E	17.380		LMTD_C	12.332		C_CO2e_an	1428.915	$/y
13	ΔT	5	oC	A_ev	19.178805		A_con	28.06781		C_total_an	13557.164	$/y
14	Eleccost	0.09	$/kWh									
15	μ_CO2e	0.968	kg/kWh									
16	c_CO2e	0.09	$/kg									
17												

Figure 3: Sheet 2 of the Excel-aided model for the simple VCR system

The areas and capital costs of the evaporator and condenser are determined by using the log-mean temperature-difference (LMTD) method. Figure 3 shows the specified evaporator and condenser temperatures at the top of the sheet.

The functions that determine the enthalpy and entropy of superheated refrigerants use the ideal-gas equation for the specific heat in which the pressure is adjusted by a "compressibility factor". El-Awad et al. [13] compared the estimations of these functions for two ozone-friendly synthetic refrigerants, R-410A and R-1234yf, and two natural refrigerants, R-290 and R-744 with the values determined by Atalay and Conan [15] who used REFPROP. They also analysed a cascade VCR system with R-507A/R-23 as the pair of refrigerants and compared their results with those obtained by Parekh and Tailor [16] who analysed the same system with EES.

To verify the functions for R-152a, the present model was used to examine the effects of the evaporator and condenser temperatures on the system's *COP*, exergetic efficiency, and total annualised cost and the results are compared to those obtained by Ref [11] on Figures 4 to 9. Figure 4 shows that without introducing the "compressibility factor" the *COP* is considerably overestimated by the model. However, with this modification the *COP* is well estimated by the model which correctly predicts that it gradually increases with the evaporator temperature. Figure 5 also shows that the model correctly estimates the variation of the exergetic efficiency with the evaporator temperature.

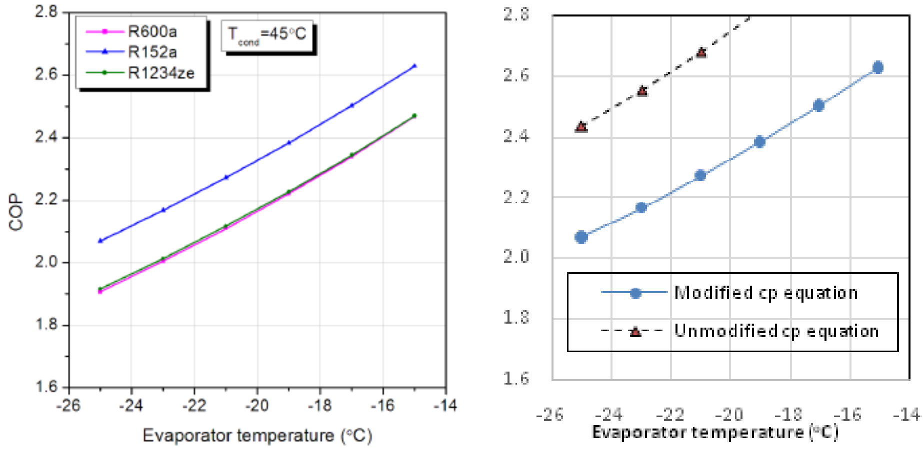


Figure 4: Effect of the evaporator temperature on the *COP*; Left, the reference model [11], Right, the present model.

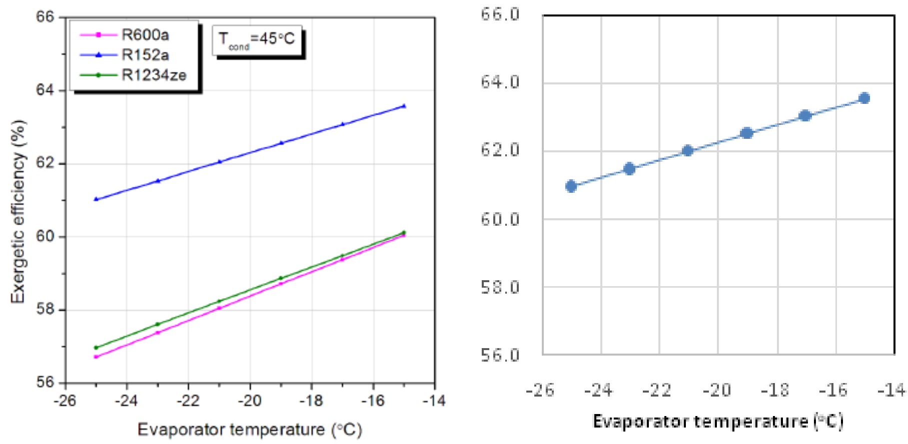


Figure 5: Effect of the evaporator temperature on the exergetic efficiency; Left, the reference model [11], Right, the present model.

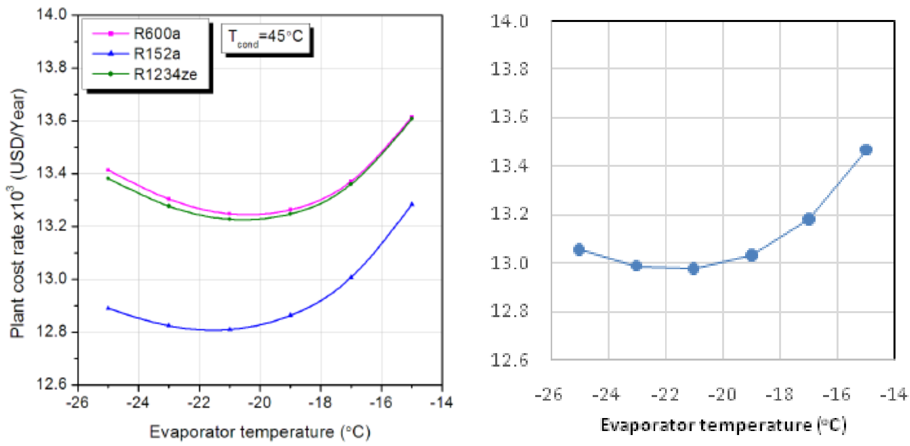


Figure 6: Effect of the evaporator temperature on the total annualised cost, Left, the reference model [11], Right, the present model

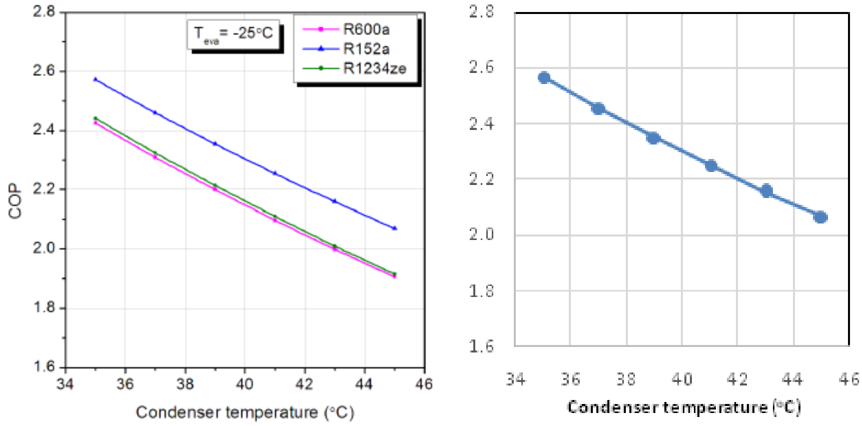


Figure 7: Effect of condenser temperature on COP, Left, the reference model [11], Right, the present model

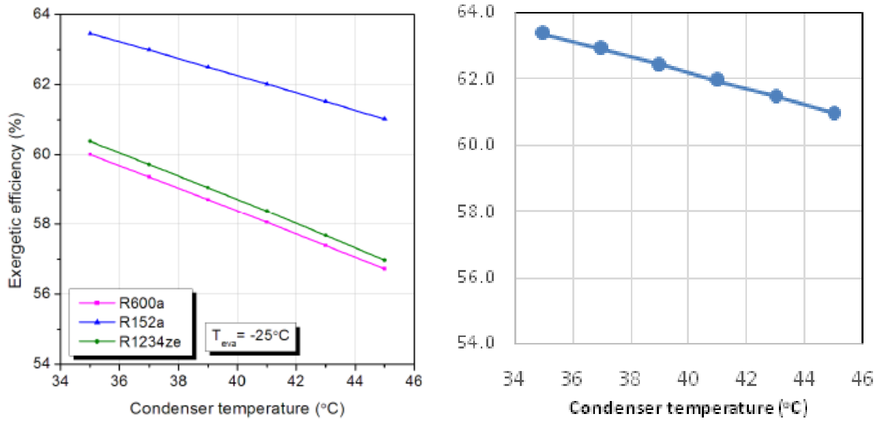


Figure 8: Effect of condenser temperature on exergetic efficiency, Left, the reference model [11], Right, the present model

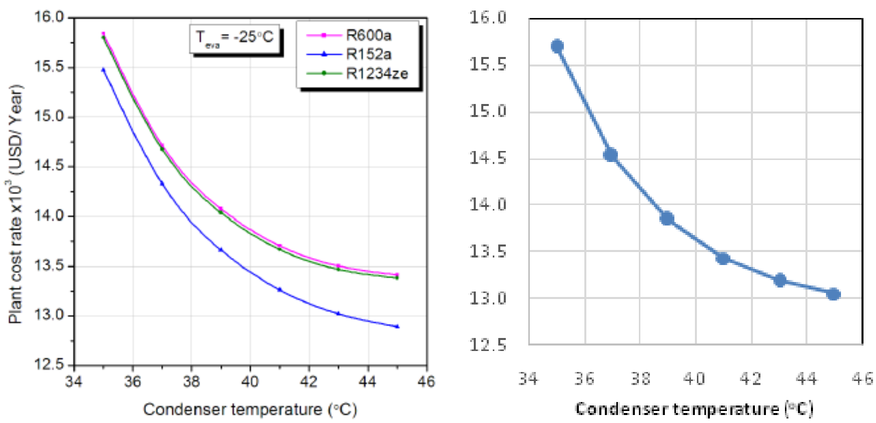


Figure 9: Effect of condenser temperature on the total cost, Left, the reference model [11], Right, the present model

Figure 6 shows that the model successfully predicted the general trend of the total cost as it varies with the evaporator temperature, but it slightly overestimated the total costs. This overestimation could be due to a slight increase in the refrigerant flow rate and, consequently, in the system's power and the cost of the compressor. Figures 7, 8, and 9 that show the variation of the three system parameters with the condenser temperature lead to similar conclusions; which gives confidence in the model to be used for the intended optimisation analysis of the VCR system.

4. 3E optimisation of the simple VCR system by using MIDACO

SOO of the simple VCR system can easily be reached by plotting the variations of the performance indicators with the evaporator and condenser temperatures. For example, Figure 5 and Figure 8 clearly show that maximising exergetic efficiency requires raising the evaporator temperature and reducing the condenser temperature. With respect to the total cost rate, Figure 6 shows that it has a minimum value at an evaporator temperature of about 21°C, while Figure 9 shows that it has its minimum value at the lowest condenser temperature. This is not the case with MOO that involves more than one performance indicator and, therefore, becomes difficult to predict without using a computer-aided method. MOO of the key performance indicators also requires a conflict between the selected indicators. For the present case, Figures 5 and 6 show that such a conflict exists between maximising the exergetic efficiency and minimising the annual plant cost since both of them increase with the evaporator temperature, while Figures 8 and 9 show that both of them decrease with the condenser temperature. Note that there is no such conflict between the exergetic efficiency and the COP that vary in the same manner with the evaporator or the condenser temperatures. Figure 10 shows the variation of the TEWI with the T_E and $T_C$from which we can see that the TEWI has the same trend like the exergetic efficiency and the COP.

Roy and Mandal [11] performed a dual-objective (2E) optimisation analysis the two objectives of which were to maximise the exergetic efficiency and minimise the annual plant cost rate. The optimum results obtained for the three refrigerants they considered identified R-152a as the best refrigerant from thermodynamic and economic considerations. In the following analysis the present Excel-aided model is used with the MIDACO solver [11] to optimise the simple VCR system also by maximising the exergetic efficiency and minimising the annual plant cost rate and TEWI. Figure 11 shows the set-up of MIDACO for the 3E optimisation analysis.

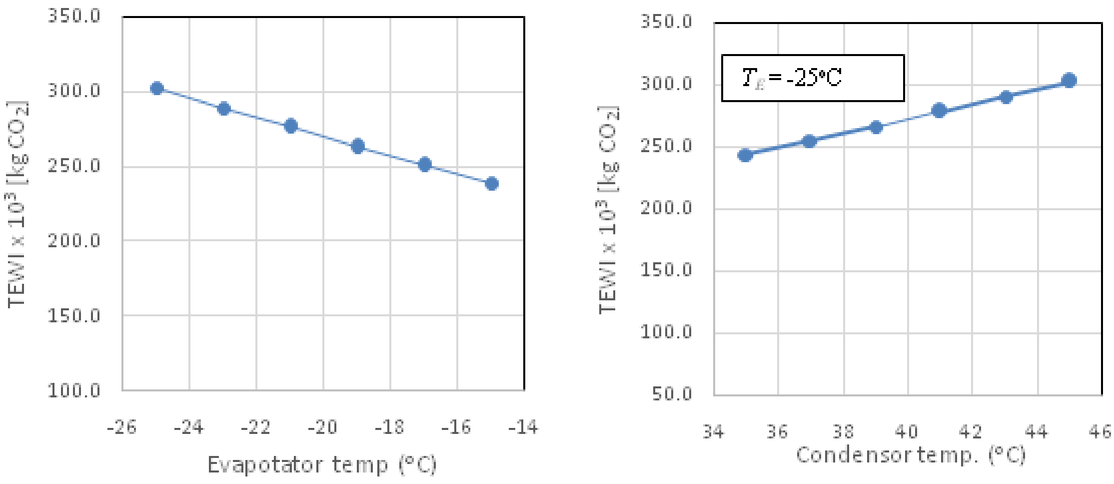


Figure 10: Effect of evaporator and condenser temperatures on the TEWI

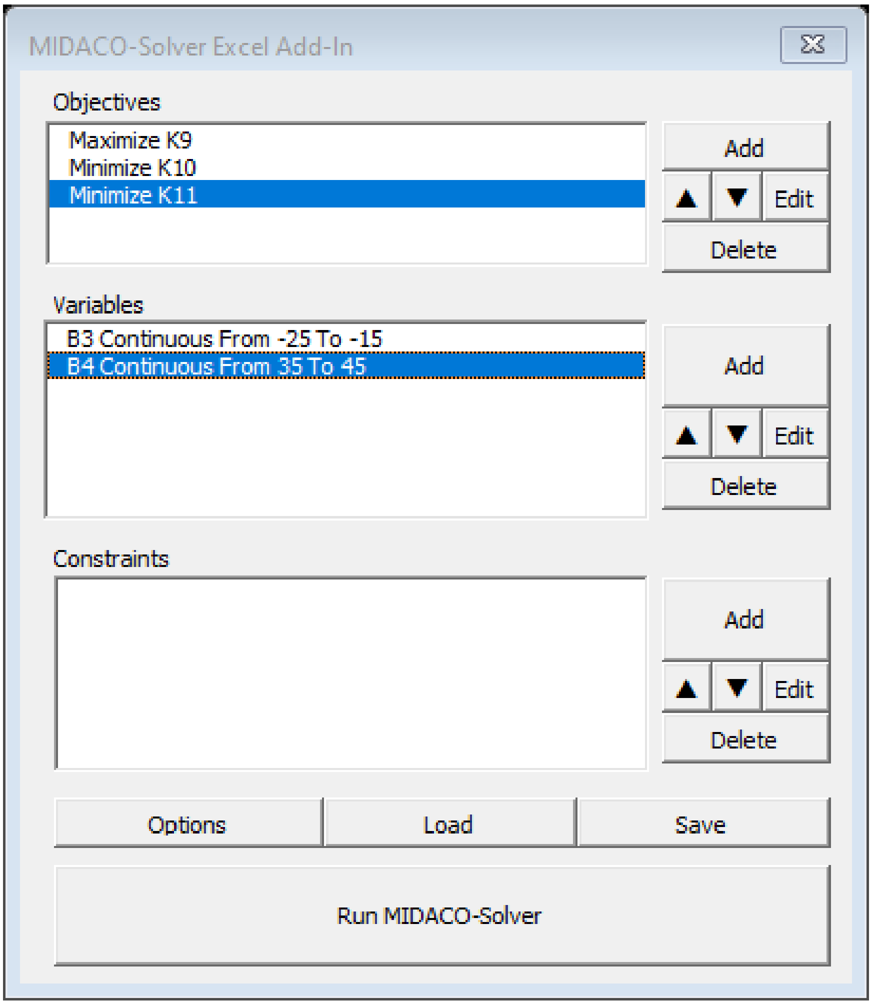


Figure 11: MIDACO set-up for the 3E-objective optimisation analysis

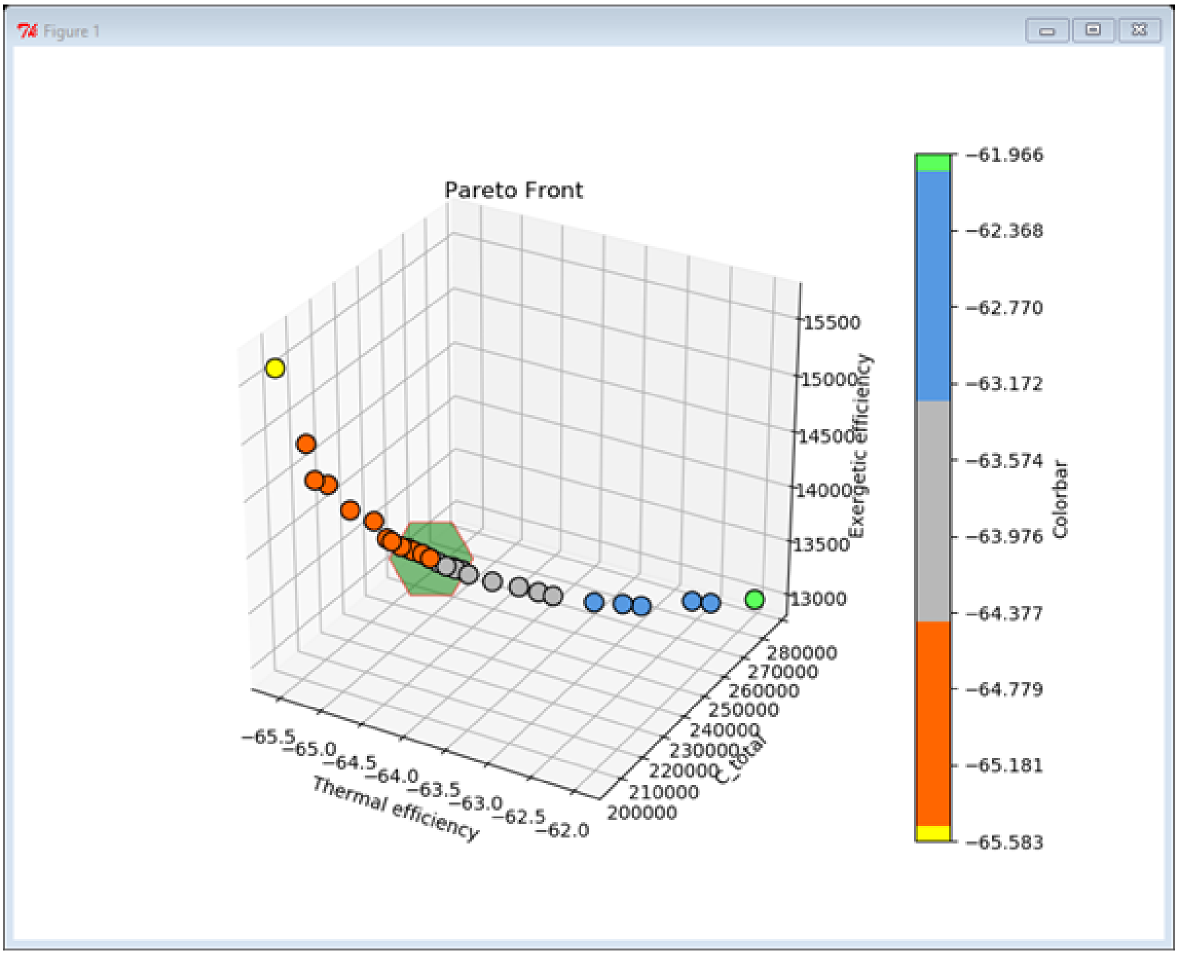


Figure 12: Pareto front obtained by MIDACO with the best 3E optimum solution

Fluid | R152a

	A	B	C	D	E	F	G	H	I	J	K	L
1	System 0											
2	Fluid	R152a										
3	T_E	-15.9038	oC	h_1	495.9192		h_4	272.0421		Q_e	223.8772	kJ/kg
4	T_C	40.36619	oC	s_1	2.153903		x_4	0.305915		m_r	0.0447	kg/s
5							s_4	1.283599		W	3.4930	kW
6	P_E	143.293	kPa	S_2s	2.153903					Q_cond	13.4930	kW
7	P_C	918.337	kPa	h_2s	558.4789		ED_evap	0.000275	kW			
8				h_2	574.1188		ED_comp	0.6078	kW	COP	2.863	
9	η_c	0.8		s_2	2.200		ED_cond	0.0963	kW	ε	64.455	%
10							ED_tvalv	0.5372	kW	C_total	13807.572	$/y
11	P_0	101.325	kPa	h_3	272.0421					TEWI	219626.9	
12	T_0	298.15	K	s_3	1.243							
13	CC	10	kW									
14												

Figure 13: Sheet 1 of the optimum 3E solution obtained by MIDACO

Following Roy and Mandal [11], the upper and lower limits imposed on the evaporator temperature are -15°C and -25°C, respectively, while those imposed on the condenser temperature are 45°C and 35°C, respectively. Figure 12 shows the Pareto front obtained by MIDACO for the present solution and Figure 13 shows the solution selected as the best optimum solution.

The optimum temperatures obtained by Roy and Mandal [11] for the 2E optimised solution were T_E = -18.8°C and T_C = 42.4°C at which ε = 70.2% and C_{total} = \$12,533/y. As Figure 13 shows, the optimum temperatures determined by MIDACO are T_E = -15°C and T_C = 40.47°C. Table 4 that compares the optimised system to the basic design shows that the optimised solution increased the exergetic efficiency by 1.53% and the total annualised cost by 1.85%, but reduced the TEWI by 9.0%.

Table 4. Comparison of the optimised system to the basic design

Parameter	**Base design**	**Optimised design**	**Change**
Evaporator temperature, °C	-20	-15.904	+4.1°C
Condenser temperature, °C	40	40.366	+0.4°C
COP	2.601	2.863	10.07%
ε %	63.483	64.455	1.53%
Total cost rate, \$/y	13,557.164	13,807.57	1.85%
TEWI, kG CO2	241,450.53	219,626.90	-9.04%

5. Conclusions

A modelling platform based on Microsoft Excel for multi-objective optimisation analyses of vapour-compression refrigeration systems enables more researchers to join the effort for developing refrigeration and air-conditioning systems that are more environment-friendly. Excel-aided models are also ideal for educational purposes. This paper describes the development of an Excel-aided model for the simple VCR system with R-152a as the refrigerant. The model uses the Thermax add-in to determine the refrigerant properties and the MIDACO solver for a triple-objective (3E) analysis. The relevant Thermax functions are first verified with the data given by Roy and Mandal [11] for this case. The results of the 3E analysis show that the optimised system improves the exergetic efficiency, COP, and TEWI of the basic design at the expense of a slight increase in its total cost rate.

Nomenclature

A	Heat-transfer area	(m^2)
$\dot{C}$	Annual cost rate	($/year)
$\dot{E}^D$	Rate of exergy destruction	(kW)
$\dot{m}$	Mass flow	(kg/s)
$\dot{Q}$	Heat-transfer rate	(kW)
$\dot{W}$	Compressor power	(kW)
CC	Cooling capacity	(kW)
c	Unit cost (e.g. electricity price)	($/kWh)
COP	Coefficient of performance	(-)
CRF	Capital recovery factor	(-)
E	Elec. energy consumption	(kWh)
h	Specific enthalpy	(kJ/kg)
i	Interest rate	(%)
N	Number of operation hours per year	(hours)
n	Plant life time	(years)
p	Pressure	(kPa)
s	Specific entropy	(kJ/kg)
T	Temperature	(K or ^{o}C)
U	Heat transfer coefficient	($kW/m^2.K$)
x	Vapour quality	(-)

Greek Letters

η	Compressor's isentropic efficiency	(%)
μ_{CO_2}	Regional (country) electricity conversion factor	($/kg)
φ	Maintenance factor	(-)

Abbreviations

3E	Energetic, economic, and environmental
MOO	Multi-objective optimisation
SOO	Single-objective optimisation
TEWI	Total equivalent warming impact
VBA	Visual Basic for Applications language
VCR	Vapour-compression refrigeration

Subscripts

0	Ambient
CO_2	Avoided carbon dioxide emission
com	Compressor
con	Condenser
elec	Electrical
env	Environmental
eva	Evaporator
op	Operation
ref	Refrigerant
Total	Total work or exergy destruction
TV	Throttle valve

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