An Excel Add-In for Refrigerants Properties and its Use for Optimising Multi-Stage Compression Refrigeration Cycles

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Abstract

The paper deals with the development and verification of an educational add-in for Microsoft Excel that determines the thermo-physical properties of various synthetic and natural refrigerants used in vapour-compression refrigeration (VCR) systems. To determine the properties of saturated and compressed liquid refrigerants, the add-in uses a linear interpolation function to interpolate the data published by ASHRAE. For superheated refrigerants, the add-in determines the enthalpy and entropy by adopting ideal-gas relationships in which the specific heat is also taken from ASHRAE data, but at a reduced pressure. The paper tests the accuracy of these relationships with different options for evaluating the reduced pressure by comparing their estimates against ASHRAE data for refrigerant R134a. The paper then shows how the add-in can be used to analyse a two-stage compression VCR system and compares the results with those obtained by using another Excel add-in for R134a developed at the University of Alabama. Finally, the paper shows how the add-in can be used with Solver to perform optimisation analyses of a three-stage compression VCR system also using R134a.

Keywords: Excel add-in, multi-stage refrigeration systems, optimisation, thermodynamic properties

1. Introduction

Cascade and multi-stage vapour-compression refrigeration (VCR) systems are more efficient than single-stage systems particularly when there is a large difference between the evaporator and condenser temperatures. However, in order to get the best performance of these more expensive systems the intermediate pressure between the low-pressure and the high-pressure stages or circuits has to be optimised. The concept of design optimisation of engineering systems for physical, economical, or environmental constraints is an important concept in engineering education and optimisation analyses of energy systems offer numerous cases for its illustration. Since optimisation analyses are time consuming, they need to be performed with the help of computer software and, to conduct them, the students can use one of the specialised items of commercial software that are available. However, there is a growing awareness among academicians that general-purpose spreadsheet applications, such as Microsoft Excel, can be more effective as teaching aids [1,2]. Compared to the specialised software, Excel allows the students to develop white-box models for their analyses; which improves their understanding of the underlying concepts.

As a computational platform for engineering analyses, Excel offers a rich library of statistical and mathematical functions for model development and a powerful iterative tool for optimisation analyses; which is the Solver add-in. What limits its capability with respect to optimisation analyses of energy-systems is its lack of built-in functions for determining the thermo-physical properties of working fluids. To solve this problem for industrial and research applications, a number of fluid-property add-ins have been developed for Excel such as the REFPROP add-in provided by the American National Institute of Standards and Technology (NIST) [3] and the set of add-ins developed by Megawatsoft [4]. One of the earliest property Excel add-ins developed for teaching purposes is the *TPX* add-in of Goodwin [5]. The team at the mechanical engineering department of the University of Alabama also developed a number of Excel add-ins for thermodynamic analyses [6-8] that include add-ins for ideal gases, water and superheated steam, and for refrigerants R134a and R22.

El-Awad [9] developed an add-in, called Thermax, that provides nine groups of property functions for the fluids mostly used in energy systems. Its group for refrigerants properties determine the thermo-physical properties of 27 synthetic and natural refrigerants based on the data given by the American Society of Heating, Refrigeration and Air-conditioning Engineers (ASHRAE) [10]. To determine the enthalpy (h) and entropy (s) of superheated refrigerants, the add-in adopts ideal-gas formulae in which the specific heat (C_p) is taken as that of the saturated refrigerant at the given pressure (P). This paper shows that the accuracy of these functions deteriorates far from the saturation line, but that the accuracy can be improved by evaluating C_p at a reduced pressure $P^* = zP$, where z is an adjustment factor. The paper examines different options for the adjustment factor and compares their accuracy by comparison with the data given by ASHRAE for refrigerant R134a. The paper then demonstrates the use of the add-in, together with Solver, for optimisation analyses of two-stage and three-stage compression VCR cycles.

2. Thermax functions for superheated refrigerants

The Appendix lists the nine groups of fluid property functions provide by Thermax. Its functions in the refrigerants group deal with 27 synthetic and natural refrigerants which are: R12, R22, R23, R32, R123, R124, R125, R134A, R143A, R152A, R245fa, R1234yf, R1234ze, R404A, R407C, R410A, R507A, methane, ethane, propane, butane, iso-butane, ethylene, propylene, ammonia, water, and carbon dioxide. Thermax functions for the properties of saturated refrigerants simply store and interpolate the data provided by ASHRAE [10]. Considering the difficulty of using the Helmholtz equation of state to determine the thermodynamic properties of all the supported refrigerants in the superheated region, the add-in uses simpler relationships based on those of ideal gases [9,11]. This section assesses the accuracy of these relationships by comparison with the data given by ASHRAE for superheated refrigerant R134a.

A. Enthalpy of a superheated refrigerant

Enthalpy (h) of a superheated refrigerant is determined from the following relationship:

$$h = h_g + Cp_g^* \left(T - T_s \right) \tag{1}$$

Where h_g and T_s are the enthalpy and temperature of the saturated vapour refrigerant at the given pressure while Cp_g^* is the value of its specific heat evaluated at a reduced pressure (P^*) given by:

$$P^* = z \times P \tag{2}$$

Where z is an adjustment factor. Figure 1 shows the values computed with Equation (1) by using $z=1.0$ and $z=0.5$ compared to ASHRAE data for the enthalpy of superheated R134a at pressures of 0.1, 1.4, and 3.0 MPa. The corresponding saturation temperatures for these three pressures, which are -26.37°C, 55.22 °C and 86.17°C, represent the low and the high temperatures met in the evaporator and the condenser of a VCR system. Values of h_g, T_s, and Cp_g^* in Equation (1) were obtained by using the relevant Thermax functions for saturated refrigerants with refrigerant R134a as the input parameter. As Figure 1 shows, the values obtained with $z = 1.0$ deviate considerably from ASHRAE data, especially at $P = 3.0$ MPa. By using $z = 0.5$, Equation (1) leads to a good accuracy at P 3.0 MPa, but it is less accurate at $P = 0.1$ and 1.4 MPa. To improve the accuracy of Equation (1), the factor z was made a function of the deviation from the critical pressure. By fitting the computed enthalpy values to ASHRAE data for R134a at the three pressures, the following formula was obtained:

$$z = 1.3 \times e^{\left(-\frac{1.55P}{P_c} \right)} \tag{3}$$

Where P_c is the critical pressure for R134a (4.0593 MPa). Note that according to this formula, the value of z converges to 0.276 as the pressure approaches P_c and to a value of 1.3 at low pressures. Another pressure-dependent function for z that was investigated was to make it equal to the compressibility factor (Z), i.e.:
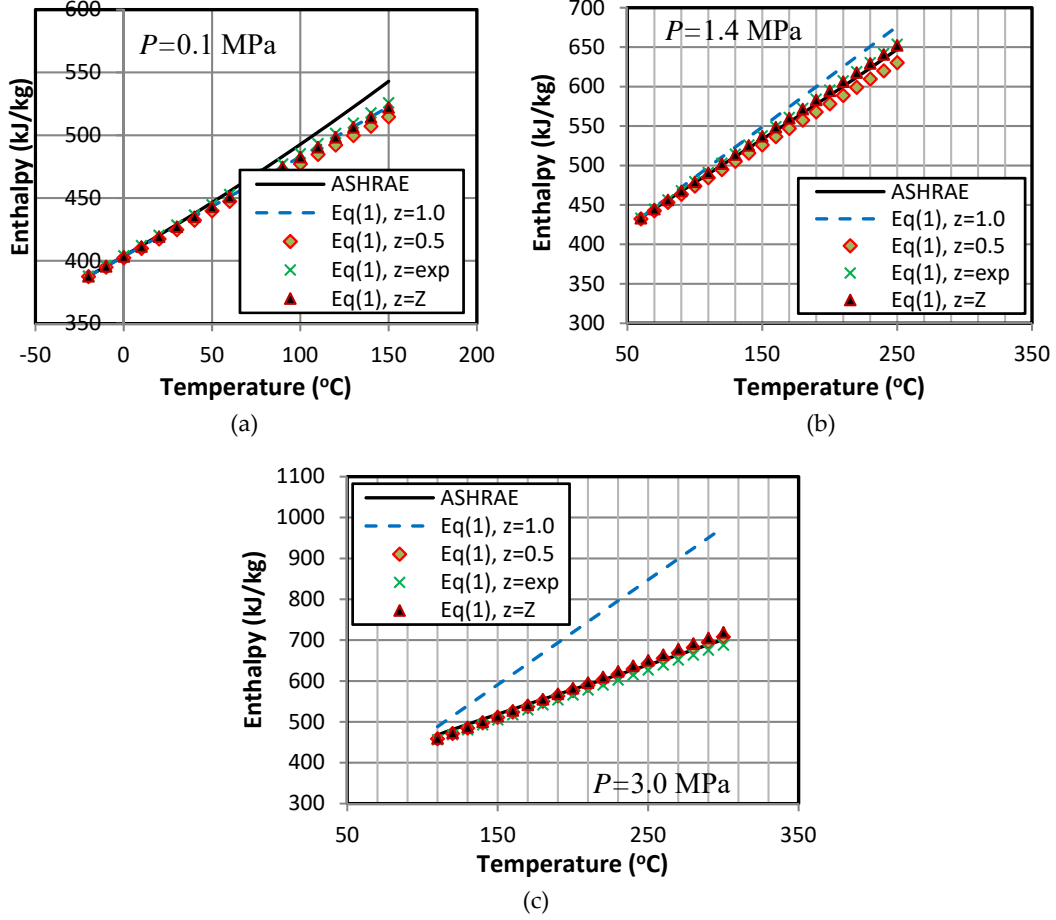
$$z = \frac{Pv_g}{RT_s} \tag{4}$$

Figure 1. The accuracy tests for different estimations of computed enthalpy values for R134a at: (a) 0.1 MPa, (b) 1.4 MPa, and (c) 3.0 MPa

Where T_s and v_g are the saturation temperature and specific volume of the saturated vapour at the given pressure, respectively, and R is the gas constant. The value of v_g was determined from ASHRAE data by using the relevant Thermax function. The enthalpy values computed with z given by Equation (3) "z=exp" and by Equation (4) "z=Z" are also show in Figure 1. The figure shows that both options give similar or better accuracy than $z = 0.5$ at the three pressure levels, but Equation (4) is more accurate than Equation (3) at $P = 3.0$ MPa. The largest deviations of the computed enthalpy values from ASHRAE data occur at P=0.1 MPa, which increase as the temperature increases, i.e. when the gas tends to behave more ideally.

B.   Entropy of a superheated refrigerant

Entropy (s) of a superheated refrigerant at a given pressure P and temperature T is determined from the following relationship:

$$s = s_g + \frac{Cp_g^*\left(T - T_s\right)}{\left(T_{av} + 273\right)} \tag{5}$$

Where s_g is the entropy of saturated refrigerant vapour at the given pressure, Cp_g^* is the specific heat at the reduced pressure P^*, and T_{av} is the average temperature:

$$T_{av} = \left(T + T_s\right)/2 \tag{6}$$

5

Figures 2 compares the values of entropy determined by Thermax on the basis of Equation (5) with the corresponding values given by ASHRAE for R134a at the three pressure levels of 0.1, 1.4, and 3.0 MPa. The figure shows the computed entropy values obtained with $z=1.0$, $z=0.5$, z determined from Equation (3), and z determined from Equation (4). As the figure shows, the last three options are reasonably accurate, but best agreement with ASHRAE data is obtained with z given by Equation (4).
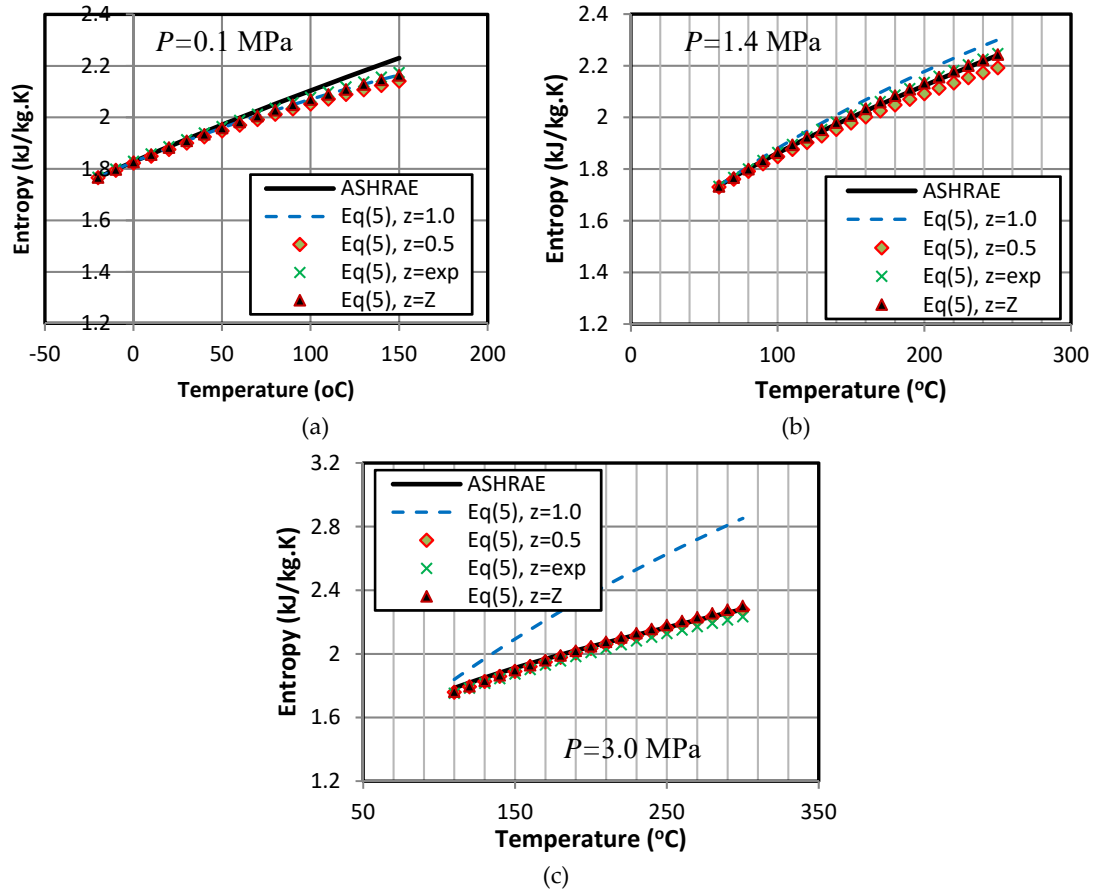

(a)

(b)

(c)

Figure 2. The accuracy tests for different estimations of computed entropy values for R134a at:
(a) 0.1 MPa, (b) 1.4 MPa, and (c) 3.0 MPa

C. Other relationships for a superheated refrigerant

The analysis of a VCR cycle may require the temperature of a superheated refrigerant to be determined given its pressure and enthalpy or entropy. For the first case, Equation (1) is rearranged as follows:

$$T = T_s + \left(h - h_g\right) / Cp_g^* \tag{7}$$

Where T_s and h_g are the saturation temperature and enthalpy of saturated refrigerant vapour at the given pressure P, but Cp_g^* is the value of the specific heat of saturated refrigerant vapour determined at the reduced pressure P^*.

Similarly, when the pressure and entropy of the superheated refrigerant are known and its temperature is to be determined, the following equation is used:

$$T = \left(T_g + 273\right) e^{\frac{s - s_g}{Cp_g^*}} - 273 \tag{8}$$

6

Where T_g and s_g are the temperature and entropy of saturated vapour refrigerant at the given pressure, while Cp_g^* is the value of the specific heat of saturated refrigerant vapour determined at the reduced pressure P^*. Note that the value of the temperature computed from Equation (8) is in °C. Thermax also provides additional functions for determining other properties of a superheated refrigerant, for example, one function directly determines the enthalpy given the pressure and entropy and another function determines the entropy given the pressure and enthalpy. The following sections show how the add-in can be used for optimisation analyses of two-stage and three-stage compression VCR cycles.

3. Optimisation analysis of the two-stage compression refrigeration cycle

The performance of single-stage VCR systems deteriorates when the evaporator temperature is very low and/or the condensing temperature is very high. This happens, for example, in frozen food industries where the required evaporator temperature can be as low as –40° C and in chemical industries for the liquefaction of gases where it can be as low as –150°C [12]. The very high condenser temperatures occur where the system is used as a heat pump for process heating, drying etc. In these situations, cascade or multi-stage compression refrigeration systems become more suitable. Cascade refrigeration systems enable different refrigerants to be used in the low and high-temperature circuits such that one of them suits the low-temperature stage while the other suits the high-temperature stage. However, the cascade system involves a heat exchanger that might add considerably to the cost of the system. In this case, the two-stage compression system shown in Figure 3.a can be more feasible by replacing the heat exchanger with a flash-chamber.
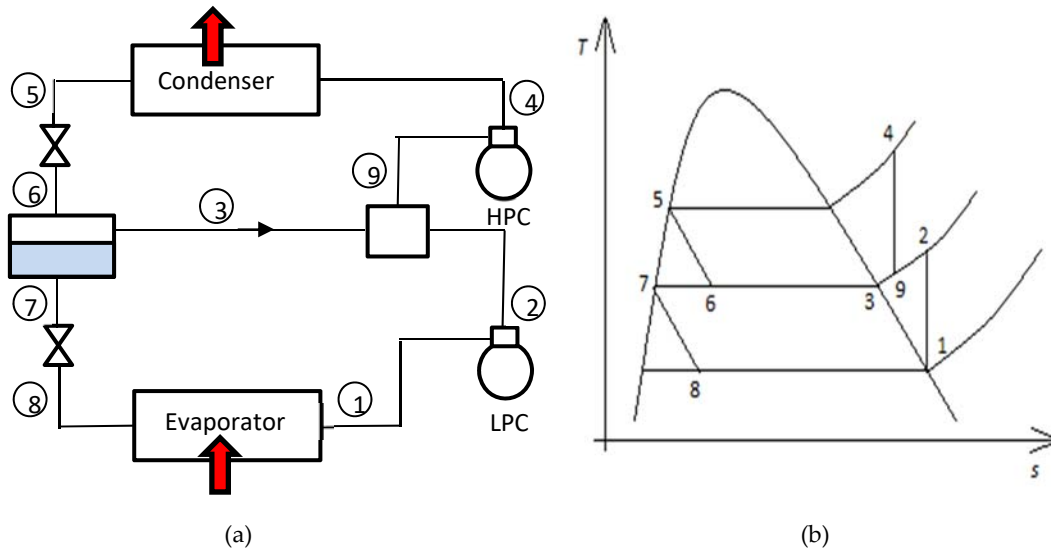

| (a) | (b) |

Figure 3. Schematic and *T-s* diagram for a two-stage compression VCR system

In the ideal *T-s* diagram shown in Figure 3.b the refrigerant is assumed to leave the evaporator as a saturated vapour and both compression processes are assumed to be isentropic. The refrigerant leaves the condenser as a saturated liquid and is throttled to a flash chamber pressure. The part of the refrigerant that evaporates during the flashing process is mixed with the refrigerant leaving the low-pressure compressor.

The mixture is then compressed to the condenser pressure by the high-pressure compressor. The liquid in the flash chamber is throttled to the evaporator pressure to cool the refrigerated space as it vaporizes in the evaporator. Cengel and Boles [13] considered a two-stage system with refrigerant R134a as the working fluid. Their system operates between evaporator and condenser pressures of 0.14 and 0.8 MPa, respectively, and its flash chamber has a pressure of 0.32 MPa. In what follows, their data and analysis results will be used to verify Thermax functions.

The analytical model

Taking the mass flow rate of the refrigerant in the condenser as $\dot{m}$, the fractions of this flow rate that are separated after the flash chamber as saturated vapour $\dot{m}_v$ and saturated liquid $\dot{m}_l$ are given by:

$$\dot{m}_v = x_6 \dot{m} \tag{9}$$

$$\dot{m}_l = (1 - x_6)\dot{m} \tag{10}$$

Where x_6 is the quality at state 6. Therefore, the rates of heat removal in the evaporator $\dot{Q}_e$ and heat rejection in the condenser $\dot{Q}_c$ are given by:

$$\dot{Q}_e = \dot{m}_l(h_1 - h_8) \tag{11}$$

$$\dot{Q}_c = \dot{m}(h_4 - h_5) \tag{12}$$

The compression work $\dot{W}_c$ which has two parts one in the low-temperature cycle and another in the high-temperature cycle, is given by.

$$\dot{W}_c = \dot{m}_l(h_2 - h_1) + \dot{m}(h_4 - h_9) \tag{13}$$

The COP of the system is then given by:

$$COP = \frac{\dot{Q}_e}{\dot{W}_c} = \frac{(1 - x_6)(h_1 - h_8)}{(1 - x_6)(h_2 - h_1) + (h_4 - h_9)} \tag{14}$$

Development of the Excel sheet and cycle optimisation

Figure 4 shows the Excel sheet developed for analysing the two-stage compression cycle using Thermax property functions. The three values of the evaporator, condenser, and flash-chamber pressures are stored in the data part that occupies the left side of the sheet. Calculations of the enthalpy values and the quality at state 6 are done in the two central columns using the add-in functions. Based on the calculated values of state enthalpies, the sheet determines the amount of heat absorbed in the evaporator (q_E), the heat rejected in the condenser (q_C), the total compression work (w_comp), and the cycle's COP (COP). Figure 5 shows the formulae entered in the three columns that calculate the intermediate and final results and Thermax functions they use. Note that only four Thermax function are used in the Excel sheet since most of the enthalpies appearing in Equation (14) are those of saturated liquid or saturated vapour at the given pressure. The required input parameters of the four functions are shown in Table 1. Also note that the refrigerant names are given as input arguments so that the same sheet can be used for other refrigerants by simply adjusting the

refrigerant name in the data part. This arrangement makes is easy to compare the performance of the system with various refrigerants.

COP =q_E/w_comp

	A	B	C	D	E	F	G	H	I	J	K
1											
2	P_evap	140	kPa	h_1	387.3095		s_2	1.740259		q_E	146.287
3				s_1	1.740259		h_2	404.0627			
4	P_cond	800	kPa							q_C	178.9803
5				h_5	243.6307		h_3	400.0385			
6	P_flash	320	kPa							w_comp	32.69327
7				h_6	243.6307		h_9	403.2381			
8				x_6	0.204904		s_9	1.737308		COP	4.47453
9											
10				h_7	203.3228		s_4	1.737308			
11							h_4	422.611			
12				h_8	203.3228						
13											

Figure 4. The Excel sheet developed for the two-stage compression VCR system

fx =q_E/w_comp

D	E	F	G	H	I	J	K
h_1	=Refh_1Px("R134a",P_evap,1)		s_2	=s_1		q_E	=(1-x_6)*(h_1-h_8)
s_1	=Refs_1Px("R134a",P_evap,1)		h_2	=Refh_1Ps("R134a",P_flash,s_2)			
						q_C	=h_4-h_5
h_5	=Refh_1Px("R134a",P_cond,0)		h_3	=Refh_1Px("R134a",P_flash,1)			
						w_comp	=(1-x_6)*(h_2-h_1)+1*(h_4-h_9)
h_6	=h_5		h_9	=x_6*h_3+(1-x_6)*h_2			
x_6	=(h_6-h_7)/(h_3-h_7)		s_9	=Refs_1Ph("R134a",P_flash,h_9)		COP	=q_E/w_comp
h_7	=Refh_1Px("R134a",P_flash,0)		s_4	=s_9			
			h_4	=Refh_1Ps("R134a",P_cond,s_4)			
h_8	=h_7						

Figure 5. Excel formulae and Thermax functions used in the Excel sheet

Table 1. Thermax functions used in the modelling the two-stage compression system

Function	Purpose and input parameters
Refh_1Px(Ref,P,x)	determines the enthalpy of a saturated refrigerant given its pressure and quality (used to determine h_1, h_3, h_5, and h_7)
Refs_1Px(Ref,P,x)	determines the entropy of a saturated refrigerant given its pressure and quality (used to determine s_1, and s_9)
Refh_1Ps(Ref,P,s)	determines the enthalpy of a superheated refrigerant given its pressure and entropy (used to determine h_2)
Refs_1Ph(Ref, P, h)"	determines the entropy of a superheated refrigerant from the known values of pressure and enthalpy (used to determine s_9)

A similar Excel sheet was also developed using the relevant property functions provided by the add-in developed at the University of Alabama (UA) for refrigerant R134a [7]. Table 2 compares the values of various cycle parameters obtained by the two Excel add-ins with their corresponding values given by Cengel and Boles [13]. The figures in the table show that enthalpy and entropy values determined by the two add-ins are almost identical even for states 2 and 4 that lie in the superheated region. The enthalpy values given by the two add-ins are different from those given by Cengel and Boles [13] because the latter uses different reference values for enthalpy and entropy, but the values of $\dot{Q}_e$, $\dot{W}_c$, and COP given by both add-ins are very close to those given

by Cengel and Boles [13]. Table 2 shows that the COP of the two-stage compression cycle is 4.47, which is significantly higher than that of the simple cycle working under the same evaporator and condenser pressures, which is 3.97 [13]. Moreover, the COP of the two-stage cycle can be improved further by suitably adjusting the pressure at the flash-chamber. The results obtained by changing the value of this pressure in the Excel sheet are shown in Figure 6. The figure shows that the COP of the two-stage compression cycle can reach 4.49 at a flash-chamber pressure of around 400 kPa.

Table 2. Verification of Thermax functions for the three-stage cycle analysis

|	Cengel and Boles [13]	UA add-in	Thermax	Deviation (%)
h_1	239.16	387.3244	387.3095	-0.0038
s_1		1.740194	1.740259	0.0037
h_2	255.93	404.0133	404.0627	0.0122
h_3	251.88	400.0513	400.0385	-0.0032
h_4	274.48	422.6194	422.611	-0.0020
h_5	95.47	243.6611	243.6307	-0.0125
h_6	95.47	243.6611	243.6307	-0.0125
x_6	0.2049	0.204879	0.204904	0.0122
h_7	55.16	203.364	203.3228	-0.0203
h_8	55.16	203.364	203.3228	-0.0203
h_9	255.10	403.2016	403.2381	0.0091
s_9		1.737218	1.737308	0.0052
$\dot{Q}_e$	146.3	146.2708	146.287	0.0111
$\dot{W}_c$	32.71	32.68758	32.69327	0.0174
COP	4.47	4.474811	4.47453	-0.0063
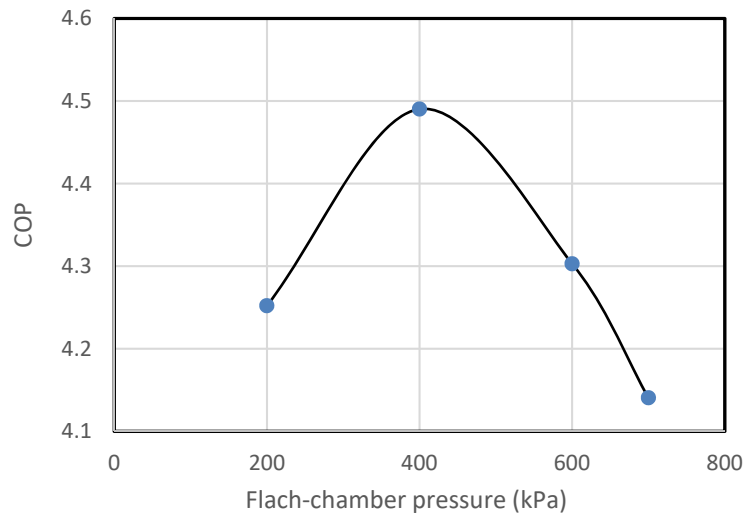
Figure 6. Variation of the COP with the flash-chamber pressure in the two-stage compression VCR system

The maximum COP and corresponding flash-chamber pressure can be determined more precisely by using Solver. Developed by Frontline Systems [14], Solver is a useful tool for "What-If" and optimisation analyses of energy systems that comes with Excel. It is found in the Data tab of Excel ribbon. Figure 7 shows Solver Parameters dialog

box that allows the user to find the maximum or minimum values of the formula in a certain cell, called the "Objective" cell, by changing the value/s of one or more variable cells that affect the Objective cell. It can also adjust the values of the Variable cells so that the Objective cell acquires a specified value. In the present analysis our objective is maximise the COP calculated in cell K8 by adjusting the pressures of the two flash-chambers stored in cells B6 as shown in Figure 4.
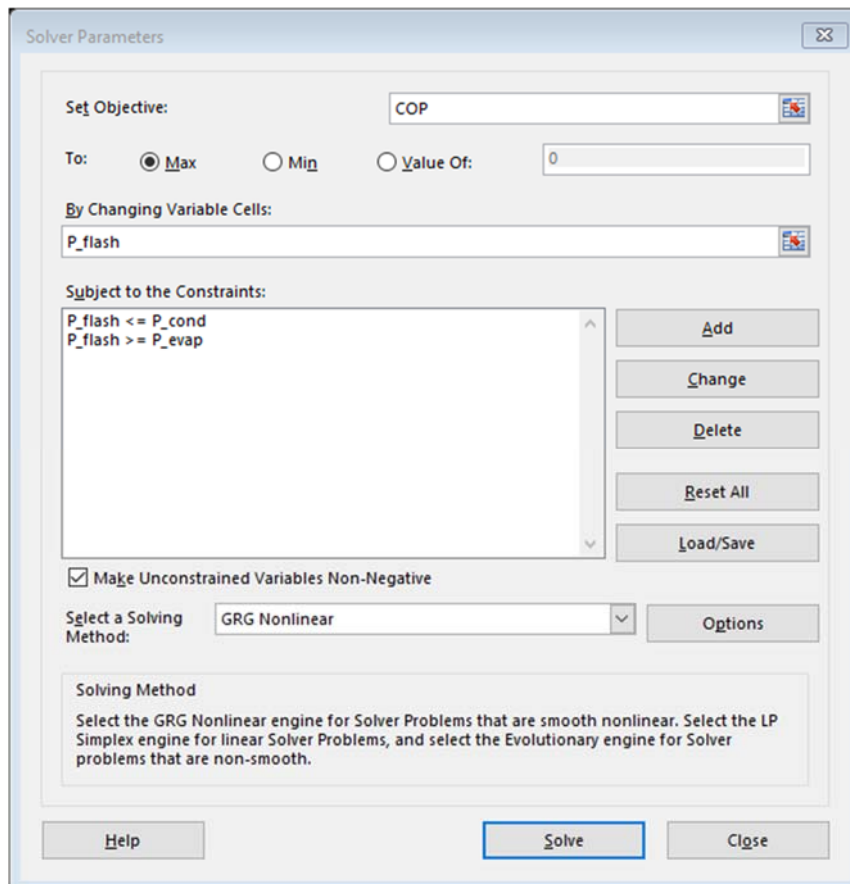

Figure 7. Solver Parameters dialog box for optimisation of the two-stage cycle

Solver allows the user to impose constraints on the solution such as upper and lower limits to the variable cells. As Figure 7 shows, two constraints have been imposed so as to keep the value of the flash-chamber pressure, P_flash, between the specified values of the evaporator ad condenser pressures. Another important feature of Solver is that it offers three solution methods that suit different types of problems. Two of these solution methods suit nonlinear problems, which are the GRG Nonlinear method and the Evolutionary method. Figure 7 shows the set-up for Solver to maximise the cycle's COP by using the GRG Nonlinear method. By pressing the "Solve" button, Solver will iterate to find the solution shown in Figure 8. The maximum COP is reached when the flash-chamber pressure is 375.45 kPa, at which the COP is 4.49. Further improvement of the system's COP is still possible by using more than one flash chamber at different pressure levels as shown in the following section.

	A	B	C	D	E	F	G	H	I	J	K
						f_x	=q_E/w_comp				
1											
2	P_evap	140	kPa	h_1	387.3095		s_2	1.740259		q_E	146.3881
3				s_1	1.740259		h_2	407.4025			
4	P_cond	800	kPa							q_C	178.9833
5				h_5	243.6307		h_3	402.6593			
6	P_flash	375.4545	kPa							w_comp	32.59521
7				h_6	243.6307		h_9	406.5653			
8				x_6	0.176504		s_9	1.737318		COP	4.491092
9											
10				h_7	209.5452		s_4	1.737318			
11							h_4	422.614			
12				h_8	209.5452						
13											

Figure 8. Solver solution for optimisation of the two-stage cycle

4. Optimisation analysis of the three-stage compression refrigeration cycle

By using two flash chambers instead of one, three-stage compression VCR systems can be more feasible than two-stage compression systems if the temperature lift is sufficiently high. Figure 9 shows the schematic diagram for such a system and Figure 10 shows its ideal *T-s* diagram. In this case, selection of the two flash-chamber pressures that maximise the system's COP requires the use of an optimisation software since it is not as easy to determine them by varying the flash-chamber pressures as in the case of a single flash chamber. In what follows, the two flash-chamber pressures that maximise the system's COP will be determined by using Solver. The system to be analysed has the same evaporator and condenser pressures as in the two-stage cycle considered earlier, i.e., 0.14 and 0.8 MPa, respectively. The two flash-chamber pressures are initially assigned values of 320 kPa and 520 kPa.
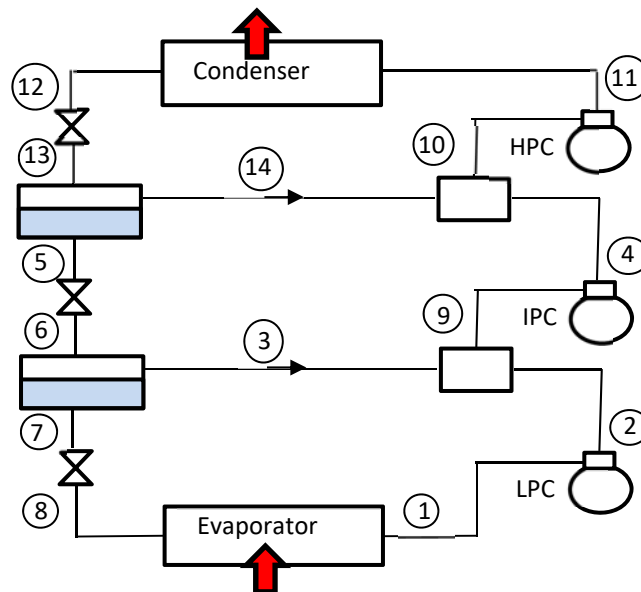
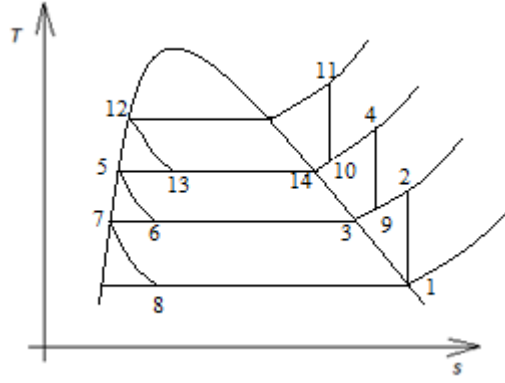Figure 9. Schematic of the three-stage compression VCR system

Figure 10. *T-s* diagram for the ideal three-stage compression VCR cycle

The analytical model

Note that the state points of the cycle in the low and intermediate pressure stages have the number as those in the two-stage compression cycle shown in Figure 3.b. Taking the mass flow rate of the refrigerant in the condenser to be $\dot{m}$, the same as that of the two-stage system considered earlier, the mass flow rates of the vapour and liquid fractions after the high-pressure flash chamber are given by:

$$\dot{m}_{14} = x_{13}\dot{m} \tag{15}$$

$$\dot{m}_5 = (1 - x_{13})\dot{m} \tag{16}$$

Accordingly, the mass flow rates of the vapour and liquid fractions after the low-pressure flash chamber are given by:

$$\dot{m}_3 = x_6(1 - x_{13})\dot{m} \tag{17}$$

$$\dot{m}_7 = (1 - x_6)(1 - x_{13})\dot{m} \tag{18}$$

The total compression work and the rate of heat rejection in the condenser are now given by:

$$\dot{W}_c = \dot{m}_7(h_2 - h_1) + (\dot{m}_3 \pm \dot{m}_7)(h_4 - h_9) + \dot{m}(h_{11} - h_{10}) \tag{19}$$

$$\dot{Q}_c = \dot{m}(h_{11} - h_{12}) \tag{20}$$

Development of the Excel sheet and cycle optimisation

Figure 11 shows the Excel sheet developed for this cycle by extending that of the two-stage compression cycle shown in Figure 5. The data part now includes two flash-chamber pressures, P_fc1 and P_fc2 instead of one. A third column has also been added to the calculations part to determine the enthalpy and entropy values in the high-pressure stage. At the specified values of the pressures at the two flash chambers, the COP is 4.65. Although this value is higher than the optimum pressure for the two-stage compression cycle determined earlier as 4.49, it is not the maximum possible value for the cycle's COP. The maximum COP can be achieved by optimising the two flash-chamber pressures by using Solver. Solver Parameters dialog box is shown in Figure 12.

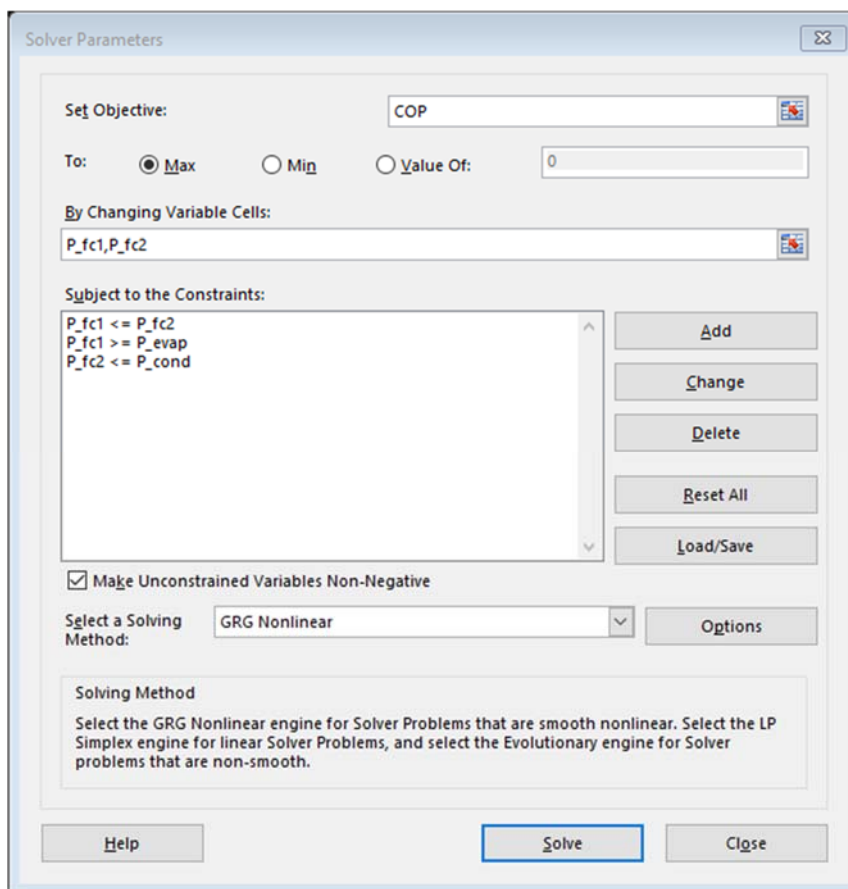Figure 11. Excel sheet for analysing the three-stage compressipon VCR cycle

Figure 12. Solver Parameters box for the optimisation of the three-stage cycle

As Figure 12 shows, three constraints have been imposed so as to keep the value of the two flash-chamber pressures, P_{fc1} and P_{fc2}, between the specified values of the evaporator pressure P_{evap}, and the condenser pressure, P_{cond}, i.e.:

$P_{evap} < P_{fc1}$

$P_{fc1} < P_{fc}2$

$P_{fc2} < P_{cond}$

Figure 13 shows the solution determined by Solver. The maximum COP is reached when the flash-chamber pressures are 283.465 kPa and 498.434 kPa. Comparison of the values in Figure 8 and Figure 13 shows that the cooling rate (q_E) has increased from 146.388 kJ for the two-stage system to 147.158 kJ for the three-stage system while the corresponding compressors work decreased from 32.595 kJ to 31.579 kJ. As a result, the COP of the three-stage system reached 4.66; which is higher than that of the two-stage compression cycle (4.49) by about 3.23% and higher than that of the simple VCR cycle (3.97) by about 14.44%. The advantage of the three-stage system over the simple or the two-stage system becomes clearer as the temperature lift increases.

	A	B	C	D	E	F	G	H	I	J	K	L	M	N
					f_x	=q_E/w_comp								
1														
2	P_evap	140	kPa	h_1	387.3095		s_2	1.740259		h_14	407.4098		q_E	147.1581
3				s_1	1.740259		h_2	401.5401						
4	P_cond	800	kPa							h_12	243.6307		q_C	178.7373
5				h_5	221.3546		h_3	398.0706						
6	P_fc1	283.465	kPa							h_13	243.6307		w_comp	31.57926
7				h_6	221.3546		h_9	401.1472		x_13	0.119729			
8	P_fc2	498.434	kPa	x_6	0.113242		s_9	1.738787					COP	4.659959
9										h_10	412.2944			
10				h_7	198.7873		s_4	1.738787		s_10	1.736527			
11							h_4	412.9587						
12				h_8	198.7873					s_11	1.736527			
13										h_11	422.368			
14														

Figure 13. Solution determined by Solver for the three-stage compression CVR cycle

5. Conclusions

This paper presented an educational Excel add-in that can be used for conducting parametric and optimisation analyses of various types of vapour-compression refrigeration systems. The add-in functions use the data provided by ASHRAE to determine the thermo-physical properties of several synthetic and natural refrigerants used in these systems. The paper focused on the functions that deal with the determination of enthalpy and entropy of superheated refrigerants by using ideal-gas relationships in which the specific heat of the superheated fluid is taken as that of the saturated vapour at a reduced pressure. The paper examines the accuracy of these functions by comparison with ASHRAE data for superheated refrigerant R134a at three pressure levels and different temperatures.

The paper also shows how the add-in can be used for analysing and optimising two-stage and three-stage compression VCR systems. Since most of the fluid states in these analyses lie in the compressed liquid or saturated mixture regions, the add-in function calculates the refrigerants properties in these states directly from the data provided by ASHRAE. Therefore, the accuracy of estimating the fluid properties in the superheated region is not expected to have a major effect on the accuracy of the overall cycle parameters determined by the present add-in. It should also be noted that the add-in functions for determining the enthalpy and entropy of superheated refrigerants are limited to the subcritical region and, therefore, for accurate estimation of these two properties by the relevant add-in functions, the pressure has to be reasonably below that at the critical point.

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Appendix: About Thermax

Thermax provides nine groups of property functions that estimate properties of the substances that are mostly used in energy systems. The functions included in each group have a common three-letter "prefix" that indicates their relevant subject(s) as shown in Table A1. For example, the names of all the functions in the group that deals with properties of ideal gases start with "Gas" and those of all the functions that deal with the properties of saturated water and steam start with "Wat". The functions in the "Ref" group deal with refrigerants. Both the "Gas" and the "Ref" groups require the name of the intended fluid as an input argument, e.g. "Air", "R134a", etc.

Table A1. Substances supported by Thermax functions

#	Group subject	Group prefix
1	Ideal gases (29 gases)	Gas
2	Water and superheated steam	Wat
3	Refrigerants (27 refrigerants)	Ref
4	Water-lithium-bromide solution	Lib
5	Ammonia-water solution	Nh3
6	Psychrometry	Psy
7	Humidified air at high temperatures	Hat
8	Chemical reaction and combustion analysis	Chm
9	Atmospheric air	Air

Two add-in groups deal with the properties of vapour-absorption refrigeration (VAR) solutions, which are the "Lib" group for water-lithium-bromide solution and the "Nh3" group for ammonia-water solution. Also, two groups deal with the properties of humidified air, which are the "Psy" group and the "Hat" group. While the "Psy" group determines properties of humidified air at normal atmospheric temperatures, the "Hat" group deals with humidified air at high temperatures as needed for power cycle's analyses. Group 8 provides thermo-chemical properties of fuels and reacting substances, such as the heating value, molar mass, etc. The functions in this group also require the name of the intended substance as an input argument. Group 9 determines the thermo-physical properties of atmospheric air at different temperatures as needed in fluid-flow and heat-transfer analyses.

In addition to its nine groups of property functions listed in Table A1, Thermax provides linear and quadratic interpolation functions that can be used to interpolate tabulated fluid properties not provided by the add-in functions. It also provides a Newton-Raphson solver for nonlinear equations like the following Soave-Redlich-Kwong (SRK) equation of state used by Thermax to determine the molar specific volume ($\tilde{v}$) of a superheated refrigerant [13]:

$$P = \frac{R_u T}{\tilde{v} - b} - \frac{a\alpha}{\tilde{v}(\tilde{v} + b)} \tag{A}$$

where, R_u is the universal gas constant, P is the absolute pressure, and T is the absolute temperature. The constants a, b and α depend on the refrigerant's pressure and temperature at the critical point and its acentric factor. Since the SRK equation is implicit in $\tilde{v}$, it is solved by using the internal Newton-Raphson solver. Figure A.1 compares the densities determined by the SRK equation for refrigerant R134a with those given by ASHRAE at pressures of 0.1 MPa, 1.4 MPa, and 3.0 MPa. The figure confirms the accuracy of the estimations obtained by the SRK equation at all three

pressure levels. The Newton-Raphson solver can also be used to solve other non-linear equations met in the analyses of energy systems like the Colebrook-White equation.
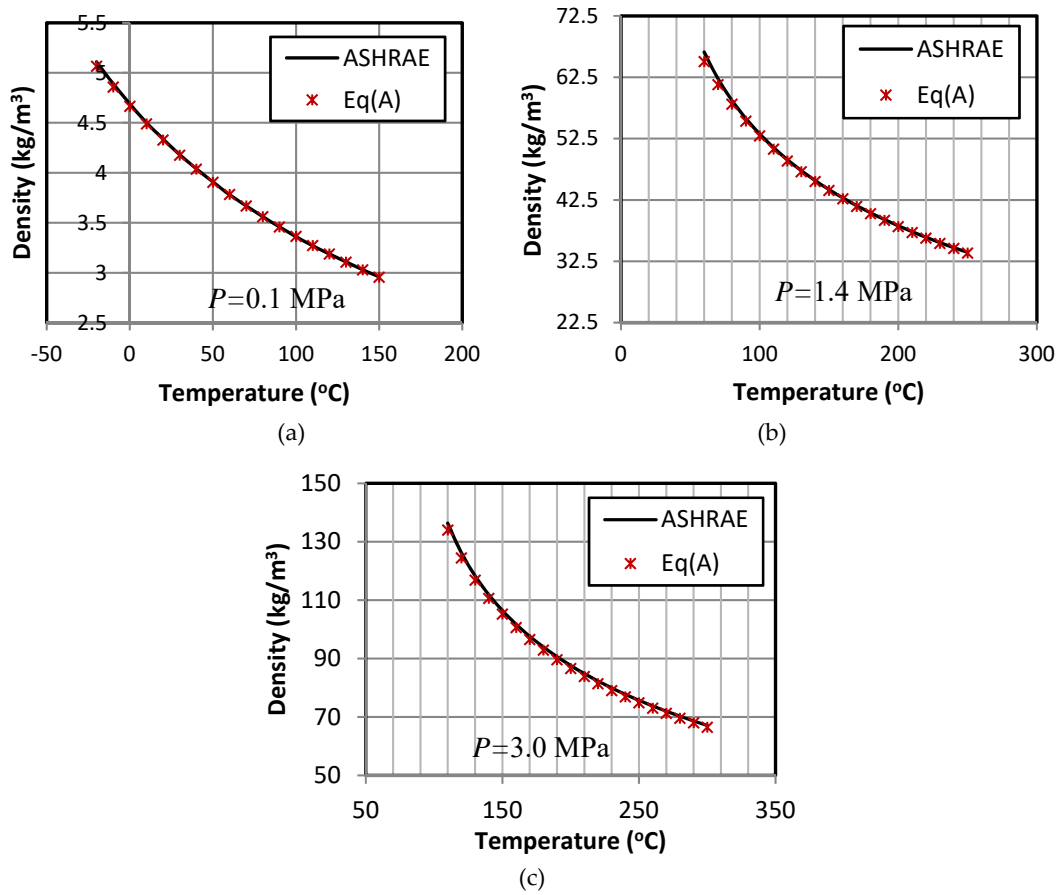

(a)

(b)

(c)

Figure A.1: Accuracy of the computed densities for R134a at: (a) 0.1 MPa, (b) 1.4 MPa, and (c) 3.0 MPa