ANALYSIS AND SIMULATION OF A SOLAR-POWERED REFRIGERATION CYCLE

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Rapidly escalating gasoline and heating oil costs in the Spring of 2000 represented the first major energy consumption crisis in this country since 1973. Nonetheless, this experience served once again to demonstrate the vulnerability of the nations of the Western world to production and marketing policies in the various oil-producing nations. One of the many energy conservation efforts in the 1970s after that earlier crisis was associated with a search for alternative methods of cooling, air-conditioning, and refrigeration.[1] One of these methods, using low-grade thermal energy (*e.g.*, solar or waste heat) to power the cooling cycle, forms the subject of this article.

The most expensive step, corresponding to the greatest amount of energy consumption, in conventional refrigeration cycles is the mechanical compression step, wherein a refrigerant vapor is compressed from a low pressure to a higher pressure. It is then condensed to liquid form in, typically, an air-cooled heat exchanger before expansion back to the same low pressure in an expansion valve, followed by vaporization—whereby the refrigeration effect occurs. This refrigerant vapor is then recompressed to the higher pressure, and the cycle is complete. These mechanical compressors are typically driven by electric motors or internal combustion engines, the energy sources for which can generally be traced back to fossil fuels or nuclear power.

For more than twenty years now,[2] the widespread use of vapor-compression refrigeration for commercial and household air conditioning has caused a shift in the seasonal peak for electric power production from mid-winter to mid-summer. This trend naturally suggests the possibility of matching demand with availability, *i.e.*, the use of solar thermal energy in the neighborhood of 200°F to power the cooling cycle rather than mechanical work, at least in certain climes. Related to this possibility is the suggestion of heat-driven mobile refrigeration cycles,[3] as in an automobile, wherein waste heat from engine cooling water could serve as the driving medium. All of these developments were naturally spurred by various energy tax credits,[4] also inaugurated in the 1970s.

PROCESS DESCRIPTION

By contrast with the conventional vapor-compression refrigeration cycle with its two refrigerant pressure levels, in this solar-powered refrigeration cycle there are three different pressure levels: low, medium, and high. Here, the high-pressure is achieved by pumping a portion of the liquid refrigerant stream, and *not* a vapor stream. This high-pressure liquid refrigerant stream is vaporized in a solar-collector heat exchanger. The high-pressure vaporized stream then serves as the motive stream to a thermal (or jet) compressor, wherein this stream sucks up the low-pressure stream from the refrigeration coil, thereby creating a medium-pressure stream (much like a laboratory aspirator). This latter stream is then totally condensed, as in a conventional refrigeration cycle, typically by heat exchange with ambient air. Part of this condensed stream feeds through an expansion valve in the low-pressure loop and then evaporates in the refrigeration coil, again just as in conventional refrigeration. The remainder of the condensed mixed stream is pumped in the high-pressure loop and to the solar collector, thus completing the refrigeration cycle.

It is clear that the crucial piece of equipment in the above cycle is the thermal compressor. With its lack of moving parts, it is certainly an attractive alternative to the conventional mechanical compressor. Rigorous mathematical modeling of thermal compressors is a rather formidable task,

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however. Earlier one-dimensional models of such devices[5,6] proceeded from first principles in fluid mechanics and thermodynamics. The model of DeFrate and Hoerl[6] was later coded in a FORTRAN routine,[7] suitable for incorporation into the early FLOWTRAN® system[8] for CAPD (but never implemented therein). Indeed, most of today's state-of-the-art CAPD systems in chemical engineering still do not have building blocks or modules for thermal compressors. This lack is due to, among other things, the complexity of the models themselves, difficulties in generalization, and the need for detailed specifications.

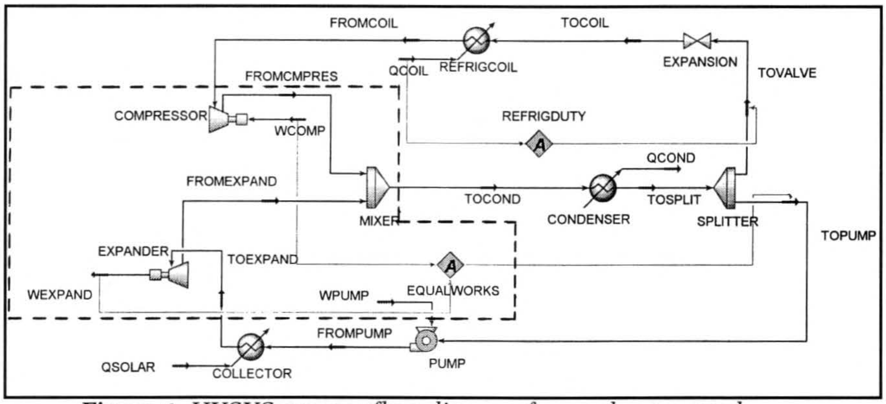

Figure 1. *HYSYS process flow diagram for a solar-powered refrigeration cycle.*

TABLE 1
Streams, Unit Operations, and Their Functions in the HYSYS Simulation of a Solar-Powered Refrigeration Cycle

Unit Operation Name	*Operation Function*
MIXER	Mixes the vapor streams (both at the same medium pressure) from the high-pressure (FROMCMPRES) and the low-pressure (FROMEXPAND) loops
CONDENSER	Rejects heat from the mixed vapor stream (TOCOND) to the ambient air (duty = QCOND)
SPLITTER	Splits the condensed stream (TOSPLIT) into the two parts feeding the low-pressure (TOVALVE) and high-pressure (TOPUMP) loops
EXPANSION	Reduces the pressure of the liquid stream in the low-pressure loop (TOCOIL)
REFRIGCOIL	Extracts heat from the environment to vaporize the stream in the low-pressure loop (FROM COIL)
REFRIGDUTY	Varies the refrigerant flow rate in the low-pressure loop to achieve the desired refrigeration duty in the coil (QCOIL)
COMPRESSOR	Compresses (work = WCOMP) the low-pressure vapor stream to the cycle's medium pressure (FROMCMPRESS)
PUMP	Increases the pressure of the liquid stream (TOPUMP) in the high-pressure loop (work = WPUMP)
COLLECTOR	Vaporizes the liquid stream (FROMPUMP) in the high-pressure loop with solar or waste heat (duty = QSOLAR)
EXPANDER	Reduces the pressure (work = WEXPAND) of the vapor stream in the high-pressure loop (TOEXPAND) to the cycle's medium pressure (FROMEXPAND)
EQUALWORKS	Equates the work of compression (WCOMP) in the COMPRESSOR with the work of expansion (WEXPAND) in the EXPANDER

There was an early attempt at modeling a solar-powered refrigeration cycle using the seminal FLOWTRAN system.[9] The module used for the thermal compressor was an adiabatic flash block, in which the high-pressure stream was simply mixed adiabatically with the low-pressure stream to yield the medium-pressure stream. The same effect could similarly have been achieved with a mixer module also operating in adiabatic fashion. Thus, these modules in the FLOWTRAN system would allow a pressure rise across them, with no concern as to how this pressure increase was to be achieved. The same, admittedly unrealistic, capability existed in the PRO/II® system.[10]

HYSYS® SIMULATION

Current CAPD systems in chemical engineering have a graphical user interface (GUI) as their input/output medium, and operate on a personal computer (PC) platform. One such modern system is HYSYS—the precursor of which was HYSIM,[11] both of which were developed by Hyprotech, Ltd., in Canada. This HYSYS system is the one presently used in the chemical engineering instructional program at Georgia Tech, and thus it is the one employed in this study.

The process flow diagram (PFD) for the solar-powered refrigeration cycle, as constructed by the HYSYS system, is shown in Figure 1. The various streams and unit operations in this PFD, as well as their functions, are described in Table 1. This HYSYS system is more realistic in the sense that it will not allow a pressure rise across either an adiabatic flashing or mixing operation. That is, the outlet stream pressure from the unit cannot exceed the pressure of any one of the incoming process streams. And, as with most present CAPD systems in chemical engineering, there is no formal thermal compressor module in the HYSYS system. Thus, an alternative method must be developed to simulate this device in the refrigeration cycle, the description of which follows.

A logical place to begin the description of this three-pressure-level refrigeration cycle is with the mixed vapor stream exiting the MIXER unit. This stream is at the medium pressure of the loop and feeds the air-cooled condenser (duty of QCOND), as in a conventional air conditioning cycle. It was assumed here that the exiting stream from this condenser was saturated liquid refrigerant at 125°F. This assumption thus determined the medium-pressure level in this process for a given (pure component) refrigerant. A summary of the various assumed operating conditions for all of these simulations is given in Table 2.

The liquid stream exiting the condenser is fed to a tee module (named SPLITTER); here, the stream is divided into the two parts, feeding the low-pressure and high-pressure loops, respectively. The stream for the low-pressure loop is fed through a conventional expansion valve and then to the refrigeration coil. The duty (QCOIL) of this coil (and hence of the refrigeration cycle) was set at

$$4 \text{ tons} = 48{,}000 \text{ BTU/hr} = 0.48 \text{ therm/hr}$$

in all of these simulations. This duty is a typical value for a modern residential dwelling of moderate size. This condition was achieved with the aid of an adjust module (named REFRIGDUTY), which varied the flow rate of refrigerant in the low-pressure or cold loop, much like a proportional controller. The thermodynamic condition of the refrigerant leaving the coil was assumed to be saturated vapor at 40°F; this condition then specified the operating pressure level in the cold loop. The refrigerant vapor was next supplied to a conventional mechanical compressor module (nonexistent in the actual process itself, of course), which recompressed this stream to the medium pressure of the process before entering the mixer module. The power requirement of this compressor is denoted as WCOMP.

TABLE 2
Operating Conditions and Parameters in the Simulations of a Solar-Powered Refrigeration Cycle

Condition/Parameter	*Value*
Temperature of the saturated vapor refrigerant leaving the refrigerant coil (T_R)	40°F
Temperature of the saturated liquid refrigerant leaving the air condenser (T_C)	125°F
Temperature of the saturated vapor refrigerant leaving the solar collector (T_S)	200°F
Refrigeration duty of the coil and cycle (Q_R)	4 tons

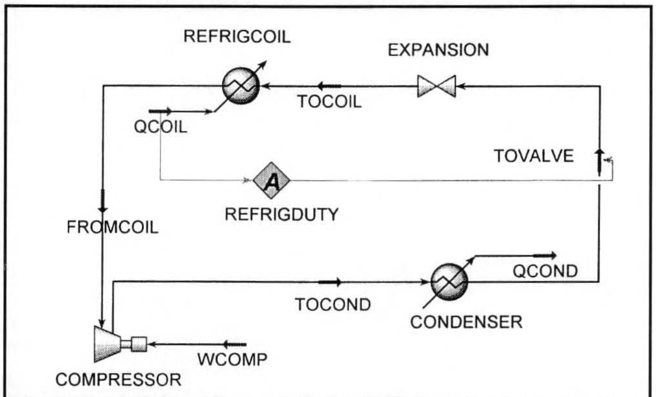

Figure 2. *HYSYS process flow diagram for a mechanical vapor-compression refrigeration cycle.*

TABLE 3
Refrigerants and Operating Pressures and Flow Rates (Refrigeration Duty (Q_R) = 4 tons = 0.48 therm/hr)

Refrigerant	*Low Pressure (P_R), psia*	*Medium Pressure (P_C), psia*	*High Pressure (P_S), psia*	*Cold Loop Flow Rate (M_c), lbs/hr*
R-113	2.906	17.34	55.03	1014
R-134a	49.7	200	506.1	881
Propane	78.49	258.8	581.8	477
Iso-propanol (IPA)	0.25	3.96	22.12	179.5

Returning to the tee module (SPLITTER) following the air condenser, the remainder of the saturated liquid refrigerant was pumped (power requirement = WPUMP) to the high-pressure level of the process. This latter value, for a given refrigerant, was dictated by the condition that the refrigerant exiting the downstream solar collector (duty of QSOLAR) was saturated vapor at 200°F—a not unreasonable value for modern solar collection systems. This collector, as well as the air condenser and the refrigeration coil, were all modeled by simple process-utility heat exchangers in these simulations. Also, for simplicity, any process fluid pressure drops in these exchangers were neglected.

Lastly, the saturated refrigerant vapor from the solar collector is fed to a conventional mechanical expander (again, non-existent in the actual process). This expander module reduces the vapor refrigerant pressure down to the medium-pressure level in the process and generates a work stream denoted as WEXPAND. Another adjust module (named EQUALWORKS) equates the power required by the mechanical compressor with that generated by the expander, by varying the refrigerant flow rate in the high-pressure loop. The process output stream from this expander, at the same medium-pressure level as the vapor stream from the compressor is then mixed with this latter stream to form the input to the air condenser, thus closing the loops.

That part of the process flow diagram in Figure 1 representing the thermal compressor is enclosed within the dashed-line envelope of that figure. The units enclosed therein include the mechanical compressor, mixer, expander, and the adjust block to equate the work streams for the two mechanical units. Also, for comparison purposes, the HYSYS process flow diagram for a comparable and conventional vapor-compression refrigeration cycle is shown in Figure 2. It is obviously considerably simpler, in that the high-pressure loop from Figure 1 is no longer present and there is no need to construct an artificial representation of a thermal compression unit here.

Four different pure-component refrigerants were investigated in this simulation study. These are summarized in Table 3. The first one of these, R-113, has been a popular refrigerant for many home air conditioning systems and was the refrigerant employed in earlier analyses[2,3] of refrigeration cycles using a jet ejector. This particular refrigerant is rapidly being replaced, however, with R-134a—a more environmentally friendly species. The remaining two prospective refrigerants considered (propane[1] and isopropanol[9]) were similarly studied by earlier investigators of these cycles. Once a refrigerant had been chosen and given the operating conditions specified in Table 2, all of the remaining process conditions followed. These latter conditions, such as the three operating pressures in the cycle and the refrigerant flow rate in the cold or low-pressure loop, are also given in Table 3. Lastly, the Peng-Robinson thermodynamic system for computing physical properties as implemented in the HYSYS system was employed throughout this work.

SIMULATION RESULTS

The use of each of the above four refrigerants in a solar-powered air conditioning cycle rated at 4 tons of refrigeration was investigated. Specifically, the effect of the adiabatic efficiencies of the compressor/expander combination on the performance of the cycle was determined. Five different values of this efficiency were chosen: 100, 90, 75, 60, and 50%. The same value of the efficiency (*e.g.*, 75%) was applied to both the compressor and the expander in a given simulation. These efficiency values bracket the compression ratio efficiencies determined experimentally (56 to 74%) in an earlier study of jet ejectors using butane and hexane as the process fluids.[5]

The simulation results obtained for the four refrigerants are summarized in Tables 4-7, respectively. Some general observations from these tables may be made first. Thus, all of the dependent parameters shown in the tables, save for the coefficient of performance (COP), increase monotonically with decreasing compressor/expander efficiency. For all practical purposes, the condenser duty (Q_C) exceeds the solar collector duty (Q_S) by the assumed refrigeration duty (Q_R = 0.48 therm/hr). This is readily apparent from the specific curves in Figure 3 for the condenser and collector duties in the case of refrigerant R-134a. The small amount of thermal energy contributed by the pump work (W_p) is also rejected by the condenser. This latter power stream is the only mechanical energy contribution to this loop, and varies from fractions of a horsepower up to 4+ hp in the worst case of propane as the refrigerant at 50% adiabatic efficiencies (Table 6). The magnitude of the pump work stream is obviously directly related to the magnitude of the hot loop circulation rate (M_h in these tables).

The coefficient of performance (COP) for a refrigeration cycle or heat pump is generally computed as the ratio of the refrigeration duty or the amount of heat pumped to the thermal energy or mechanical work supplied to the cycle.[12] Thus, a COP value in this study was computed as the quotient of the refrigeration effect (Q_R) divided by the solar collector duty (Q_S), or COP = Q_R/Q_S. Note that the small amount of mechanical work contributed to the cycle by the pump (W_p) was ignored. This COP quantity then varies in the range of 0.45-0.55 down to 0.1+ as the adiabatic efficiencies decrease. The best values are observed in the case of isopropanol (Table 7). Among other deficiencies, however, this refrigerant suffers from the rather large compression ratio (>10) required in the compression step (see Table 3). Isopropanol is followed in performance by R-113, as shown in Table 4. Propane and R-134a (Table 5) are virtually identical in displaying the poorest performance of the four refrigerants; they also require the highest operating pressures in the loops.

The COP values calculated above may be compared with maximum theoretical values. Thus, Chen[3] also begins his analysis of this cycle by ignoring the negligible amount of work contributed to the cycle by the pump (W_p). The maximum attainable coefficient of performance for the ejector-operated refrigeration cycle is then equal

. . . this study of a solar-powered refrigeration cycle, exploring different refrigerants, efficiencies, operating conditions, etc., could represent an excellent computer-aided design project in an introductory engineering thermodynamics course. It is in this spirit that this study was formulated.

TABLE 4
Effects of Compressor/Expander Efficiencies with R-113
(Refrigeration Duty (Q_R) = 4 tons = 0.48 therm/hr)

Compressor/ Expander Efficiency (E), %	*Collector Duty (Q_S), therms/hr*	*Condenser Duty (Q_C), therms/hr*	*COP (=Q_R/Q_S)*	*Compressor/ Expander Work (W), hp*	*Pump Work (W_p), hp*	*Hot Loop Flow Rate (M_h), lbs/hr*
100	0.982	1.463	0.489	3.881	0.053	1363
90	1.212	1.694	0.396	4.312	0.066	1683
75	1.746	2.228	0.275	5.175	0.095	2423
60	2.728	3.211	0.176	6.469	0.148	3787
50	3.928	4.413	0.122	7.762	0.213	5453

TABLE 5
Effects of Compressor/Expander Efficiencies with R-134a
(Refrigeration Duty (Q_R) = 4 tons = 0.48 therm/hr)

Compressor/ Expander Efficiency (E), %	*Collector Duty (Q_S), therms/hr*	*Condenser Duty (Q_C), therms/hr*	*COP (=Q_R/Q_S)*	*Compressor/ Expander Work (W), hp*	*Pump Work (W_p), hp*	*Hot Loop Flow Rate (M_h), lbs/hr*
100	1.081	1.579	0.444	4.338	0.723	1667
90	1.334	1.837	0.360	4.821	0.893	2058
75	1.922	2.434	0.250	5.785	1.286	2964
60	3.002	3.534	0.160	7.231	2.010	4631
50	4.323	4.877	0.111	8.677	2.894	6669

TABLE 6
Effects of Compressor/Expander Efficiencies with Propane
(Refrigeration Duty (Q_R) = 4 tons = 0.48 term/hr)

Compressor/ Expander Efficiency (E), %	*Collector Duty (Q_S), therms/hr*	*Condenser Duty (Q_C), therms/hr*	*COP (=Q_R/Q_S)*	*Compressor/ Expander Work (W), hp*	*Pump Work (W_p), hp*	*Hot Loop Flow Rate (M_h), lbs/hr*
100	1.076	1.584	0.446	4.455	1.082	961
90	1.328	1.842	0.361	4.950	1.336	1187
75	1.913	2.442	0.251	5.940	1.924	1709
60	2.989	3.545	0.161	7.425	3.006	2671
50	4.304	4.894	0.112	8.910	4.329	3846

TABLE 7
Effects of Compressor/Expander Efficiencies with iso-Propanol
(Refrigeration Duty (Q_R) = 4 tons = 0.48 therm/hr)

Compressor/ Expander Efficiency (E), %	*Collector Duty (Q_S), therms/hr*	*Condenser Duty (Q_C), therms/hr*	*COP (=Q_R/Q_S)*	*Compressor/ Expander Work (W), hp*	*Pump Work (W_p), hp*	*Hot Loop Flow Rate (M_h), lbs/hr*
100	0.884	1.364	0.543	3.644	0.010	271
90	1.092	1.572	0.440	4.049	0.013	335
75	1.572	2.052	0.305	4.859	0.018	482
60	2.456	2.937	0.195	6.073	0.029	753
50	3.537	4.018	0.136	7.288	0.041	1084

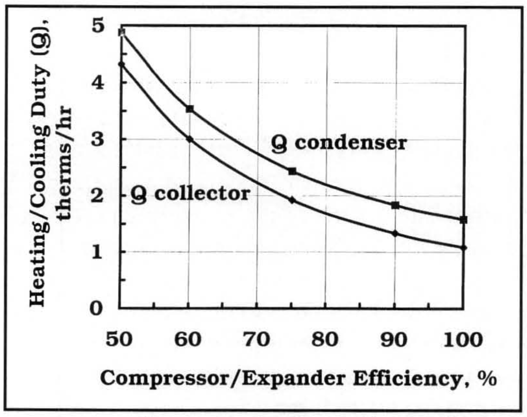

Figure 3. *Air condenser and solar collector duties for a 4-ton solar-powered air-conditioning cycle, with R-134a as the refrigerant, as functions of the adiabatic efficiencies of the compressor and expander.*

to the coefficient of performance for a Carnot refrigeration cycle (COP_C) working between the temperatures of the refrigeration coil (T_R) and the heat rejection temperature (T_0), multiplied by the efficiency of a Carnot heat engine (E_C) operating between the solar collector temperature (T_S) and the rejection temperature of T_0. The above Carnot refrigeration cycle can also be viewed as a heat pump operating in the cooling mode between the two temperatures of T_R and T_0. If one selects the condenser temperature (T_C) as the heat rejection temperature, then

$$COP_c = \frac{T_R}{T_C - T_R} = \frac{500}{585 - 500} = 5.882 \tag{1}$$

and

$$E_C = \frac{T_S - T_C}{T_S} = \frac{660 - 585}{660} = 0.1136 \tag{2}$$

from which COP = (COP_C)(E_C) = 0.6684. The best COP values (at adiabatic efficiency values = 100%) in Tables 4-7 are seen to approach this value.

The selection of the heat rejection temperature (T_0) is clearly somewhat arbitrary. The selection of the condenser temperature (T_C) of 125°F as this rejection temperature is admittedly a very conservative choice, leading to the poorest or lowest values for the theoretical COP. As this temperature is reduced, the computed theoretical COP value improves, as summarized in Table 8, wherein these values are calculated for heat rejection temperatures of T_0 = 125, 110, 100, 90, and 77°F. The value of T_0 = 100°F, for example, was chosen by Chen[3] in his analysis. Of course, a common value for this latter quantity is 77°F, particularly in thermodynamic availability or exergy analyses.[12]

In his more thorough analysis of this cycle, Hamner[2] computes various other theoretical coefficient of performance values, which are generally less than those from the Carnot analysis and thus somewhat more realistic. An analysis assuming an isentropic turbine-compressor combination with no mixing losses yields COP values not much less than the Carnot values. Once mixing losses are allowed to affect the results, using either an ideal gas model or a real gas model, the COP values drop markedly due to the internal irreversibilities or lost work. Hamner also reports experimental data on such an ejector-operated refrigeration cycle, rated at approximately one ton of refrigeration and employing R-11 as the refrigerant. Experimental COP values of about 0.10 to 0.25 were obtained for pressure ratios (P_S/P_C) of 5.0 to 7.5.

CONCLUSIONS

This article has demonstrated the applicability of the HYSYS computer-aided process design system to the simulation and analysis of a solar-powered refrigeration cycle. While such a cycle consists of a number of standard chemical process equipment items such as heat exchangers, a pump, and an expansion valve, the key hardware element in this cycle is a thermal compressor or jet ejector. Models of the latter item, while a relatively common piece of processing equipment in the chemical and allied industries, are not that extant in computer-aided process design systems such as HYSYS or comparable software packages. The employment of an adjust or control module to balance the work of a compressor and an expander in a cycle was illustrated in this work.

The coefficient of performance (COP) values for refrigeration cycles driven by a solar collector and jet ejector are admittedly much smaller than those of conventional cycles employing mechanical compressors. As numerous authors[1-3] have pointed out, however, applications of the former may be economical in cases wherein the required input heat is very inexpensive (*e.g.,* solar energy) or it would be otherwise wasted, as from the cooling system of an automobile engine. And there are certainly more than just technological factors operative in this arena.[4] Lastly, it should be remembered that the energy input to a mechanical vapor-compression refrigeration cycle generally originates from an electrical power plant. This power often derives from the combustion of a fuel with a process efficiency of about 33%. Thus, the ultimate amount of energy required in such a mechanical cycle is roughly three times the amount actually supplied to the compressor.

TABLE 8
Influence of Heat Rejection Temperature (T_0) on COP and Efficiency Values ($T_R = 40°F$, $T_S = 200°F$)

Rejection Temperature (T_0), °F	Refrigeration Cycle $(COP)_C$	Efficiency of heat engine (E_C)	Overall cycle COP $[=(COP)_C(E_C)]$
125	5.882	0.1136	0.6684
110	7.143	0.1364	0.9740
100	8.333	0.1515	1.2626
90	10.000	0.1667	1.6667
77	13.514	0.1864	2.5184

NOTE FROM OCTAVE LEVENSPIEL

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Understanding Engineering Thermo

and it uses a radically different teaching approach. Students like it.

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Perhaps the major contribution of this work is of a pedagogical nature. Thus, this study of a solar-powered refrigeration cycle, exploring different refrigerants, efficiencies, operating conditions, etc., could represent an excellent computer-aided design project in an introductory engineering thermodynamics course. It is in this spirit that this study was formulated.

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