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Ashwath Bhat
Indian Institute of Technology Madras, India, ashwath.m.bhat@smail.iitm.ac.in

Anandu Surendran

Satyanarayanan Seshadri

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Ashwath BHAT[1], Anandu SURENDRAN[1], Satyanarayanan SESHADRI[1*]

[1] Indian Institute of Technology Madras, Chennai-600036,
Tamil Nadu, India
Tel.: +91 44 2257 4072, E-mail: satya@iitm.ac.in

* Corresponding Author

ABSTRACT

Increasing air conditioning and domestic refrigeration demands pose a massive burden on grid electricity calling for efficient cooling technologies. In this paper, a novel refrigeration architecture - the Dual Evaporator Dual Ejector Cycle (DEDEC) is proposed. The DEDEC utilizes separate high and low-temperature (HT and LT) evaporators and two ejectors, allowing cooling at two different temperatures levels. Compared to the conventional Dual Evaporator Ejector Cycle, the proposed architecture maintains a higher temperature difference between the two evaporators while reducing the throttling losses. This feature finds application for domestic refrigeration where the temperature difference between the fresh food and freezer section is to be maintained. Additionally, the thermal compression provided by the dual ejectors decreases the compressor work, leading to higher COPs than the dual evaporator vapour compression systems.

In this study, a steady-state thermodynamic model for the proposed DEDEC is tested for a domestic refrigerator application. A sensitivity study on various cycle parameters is presented. Results based on R134a show that for fixed condensing temperature and LT evaporator temperatures of -15°C and -20°C, the system is able to maintain the HT evaporator at a temperature of -4.3°C and -7.8°C, respectively. Corresponding minimum load ratios (fresh food : freezer) are found to be 1.67 and 1.75, respectively. Compared to a standard vapour compression cycle (VCC) operating in the same conditions, the DEDEC delivers a COP improvement of 31% and 35% respectively.

Lastly, the potential of integrating a low-grade heat source such as solar power with the system is analyzed. The ejector motive fluid is preheated, enabling the system to achieve a higher temperature difference between the HT and LT evaporators. The higher temperature difference reduces the dehumidification of air and build-up of frost typically associated with the conventional VCC refrigeration systems. Results indicate that for an LT evaporator temperature of -20°C, a 10°C preheating delivers an additional 4.6°C temperature rise in the HT evaporator, bringing it to -3.2 °C. Compared to the case without heat integration, a 13.6% decrease in COP is noted with heat integration. The additional heat requirement is 28% of the compressor power to be supplied at a temperature greater than 57°C, which is typical of residential solar heater systems. Therefore, DEDEC refrigerators can be readily deployed in residential buildings with solar heating.

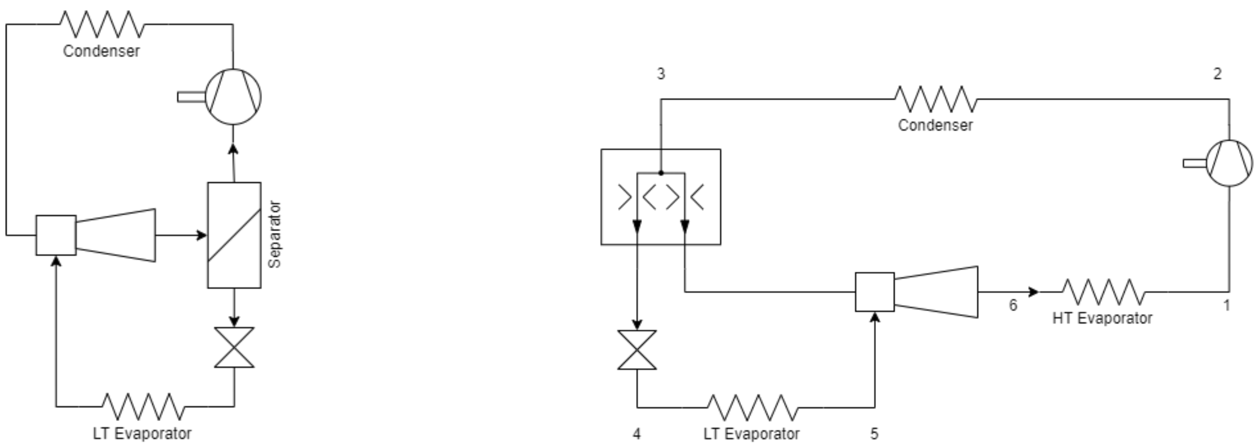
1. INTRODUCTION

Refrigerators, used to retain the quality of various edibles, consume roughly 14.5% of the energy consumed by an average Indian household as of 2008. A typical refrigerator consists of two compartments, the fresh food and the freezer compartments, which are maintained at two separate temperatures and have distinct associated cooling loads. In most domestic refrigerators, the cooling for the fresh food and freezer compartments are provided by a single vapour-compression cycle operating at the freezer's temperature. Here, the evaporator is used to cool the freezer compartment directly, and the fresh food compartment's refrigeration needs are met by circulating the freezer's air.

Dual evaporator variants have also been developed and is a major focus of many manufacturers, which provide two major advantages. By ensuring each compartment is cooled by an evaporator with matching temperature, particularly the fresh food compartment, the irreversibility associated with the heat transfer is reduced. Furthermore, better humidity control is observed as the humid air from the fresh food compartment is no longer coming in contact with the freezer's evaporator, thus preventing frost growth and unnecessary desiccation of the food in the fresh food compartment. These dual evaporator variants can be a single compressor or dual compressor system. In the former, the condenser outlet is typically throttled to two separate pressures providing cooling at two distinct temperatures. In the latter case, two separate cycles are used to provide refrigeration to the two separate compartments.

The throttling process in a refrigeration cycle, from a high-pressure saturated or subcooled liquid to a low-pressure two-phase mixture, is an irreversible process that limits the potential performance of the cycle. Using two-phase ejectors for recovering this expansion work recovery was first proposed by Gay (1931) and has seen numerous applications since then in making refrigeration systems more energy efficient. The cycle proposed by Gay, Ejector Expansion Refrigeration Cycle (EERC), involves an ejector discharging into a liquid-vapour separator. The separator's vapour is compressed and enters the condenser, and the condenser outlet is connected with the ejector's motive nozzle. The liquid from the separator is throttled and enters the evaporator, and the evaporator outlet is connected to the ejector's suction nozzle, thus completing the cycle.

Expanding the idea to a Dual-Evaporator Ejector Cycle (DEEC) was proposed by Oshitani et al. (2005). In this, the discharge from the two-phase ejector is sent through a second evaporator, following which the vapour is compressed. The condenser outlet in this cycle is split between two streams. One is throttled and enters the first/low-temperature (LT) evaporator, while the other enters the ejector's motive nozzle. The discharge from the ejector enters the second/high-temperature (HT) evaporator. Due to the ejector's thermal compression, the HT evaporator's saturation temperature is higher than that of the LT evaporator. However, the temperature difference between the two evaporators is low for use in domestic refrigerators.



(a) EERC

(b) DEEC

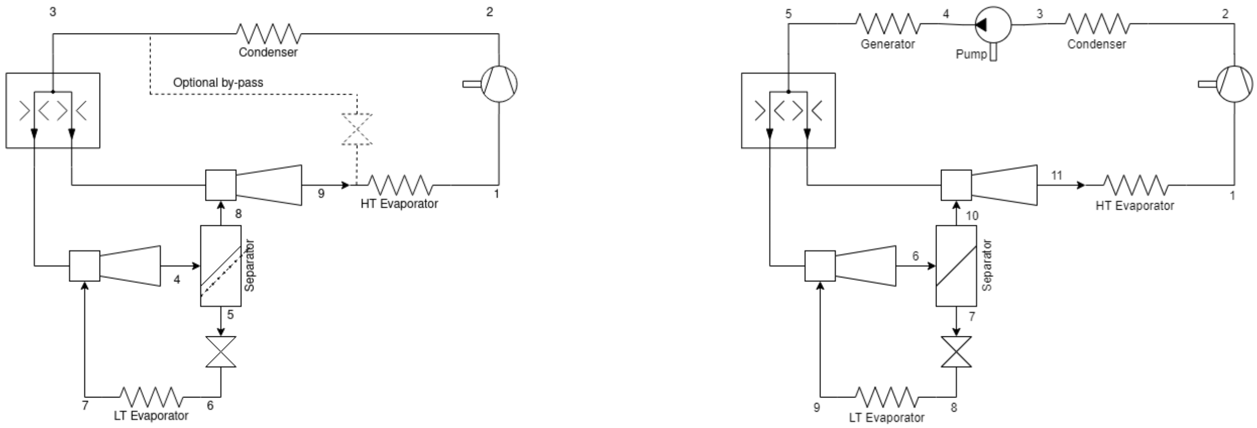
Figure 1: Layouts of the EERC and DEEC ejector refrigeration cycles

This paper proposes a dual ejector cycle with a single compressor and two evaporators and compares its COP with the typical single compressor dual evaporator cycle.

2. CYCLE DESCRIPTION

The refrigeration cycle proposed in this paper, the Dual-Evaporator Dual-Ejector Cycle (DEDEC), can be considered a modification of DEEC and EERC. Like the EERC, the DEDEC has a separator serving as the outlet of the first ejector. Liquid from the separator is throttled into the LT evaporator, from where it exits into the first ejector's suction nozzle. The vapour from the separator is directed to the suction nozzle of the second ejector, and the discharge from this ejector is sent to the HT evaporator - which exits to the compressor. The condenser outlet is split into two streams directed to the two ejectors' primary nozzles, respectively. Since the refrigerant undergoes thermal compression by two ejectors, the increase in the HT evaporator's saturation temperature is higher for DEDEC than DEEC. Additionally, by adding the separator, the throttling process in DEDEC happens at a lower pressure drop than the DEEC.

The load ratio (HT evaporator : LT evaporator) for the DEDEC is constant under steady-state conditions. For load ratio flexibility, a by-pass can thus be introduced at the condenser outlet. For this, a third stream from the condenser outlet is introduced, which is throttled and enters the HT evaporator - leading to an increase in the load ratio. This



(a) DEDEC **(b)** SI-DEDEC

Figure 2: Layouts of the proposed dual evaporator dual ejector cycles

by-pass in the DEDEC can be viewed as the superimposition of the DEDEC with a vapour compression cycle (VCC) operating at HT evaporator and condenser saturation temperatures.

The COP of a refrigeration cycle is calculated as:

$$COP = \frac{Q_{ev}}{W_{cp}} = \frac{Q_{cn}}{W_{cp}} - 1 \tag{1}$$

Even with the by-pass, the modified DEDEC operates with the same condenser and compressor inlet and outlet state points. Thus the ratio Q_{cn}/W_{cp} remains the same. This would mean that a modified DEDEC with by-pass will theoretically have the same COP as one without a by-pass, irrespective of the new load ratio.

For comparing cycle parameters, a dual evaporator single compressor refrigeration cycle (DESCC) has been considered. In this modification of the VCC, the condenser outlet is split into two steams, which are throttled into the two evaporators providing cooling at two distinct saturation temperatures. The HT evaporator outlet is then throttled to the LT evaporator's pressure, where the two streams are mixed before proceeding to the compressor. Evident through a state-point analysis, COP of the VCC and DESCC are equivalent irrespective of the HT evaporator, with minor deviations accounting for the subtle changes in the degree of super-heat at the compressor inlet.

To allow a further increase in the temperature rise for the second evaporator in the DEDEC, the motive nozzle inlet's enthalpy can be increased for the two ejectors. For this, the condenser discharge is pumped to a higher pressure and heated, following which it enters the motive nozzle. Here, though heated, the refrigerant remains in the sub-cooled region and the motive fluid remains liquid. The heater is referred to as the generator, and this modification is referred to as Solar Integration (SI) for the remaining of the paper. As discussed later, this modification comes with a trade-off of reduced COP and a higher steady-state load ratio.

3. MODELING METHODOLOGY

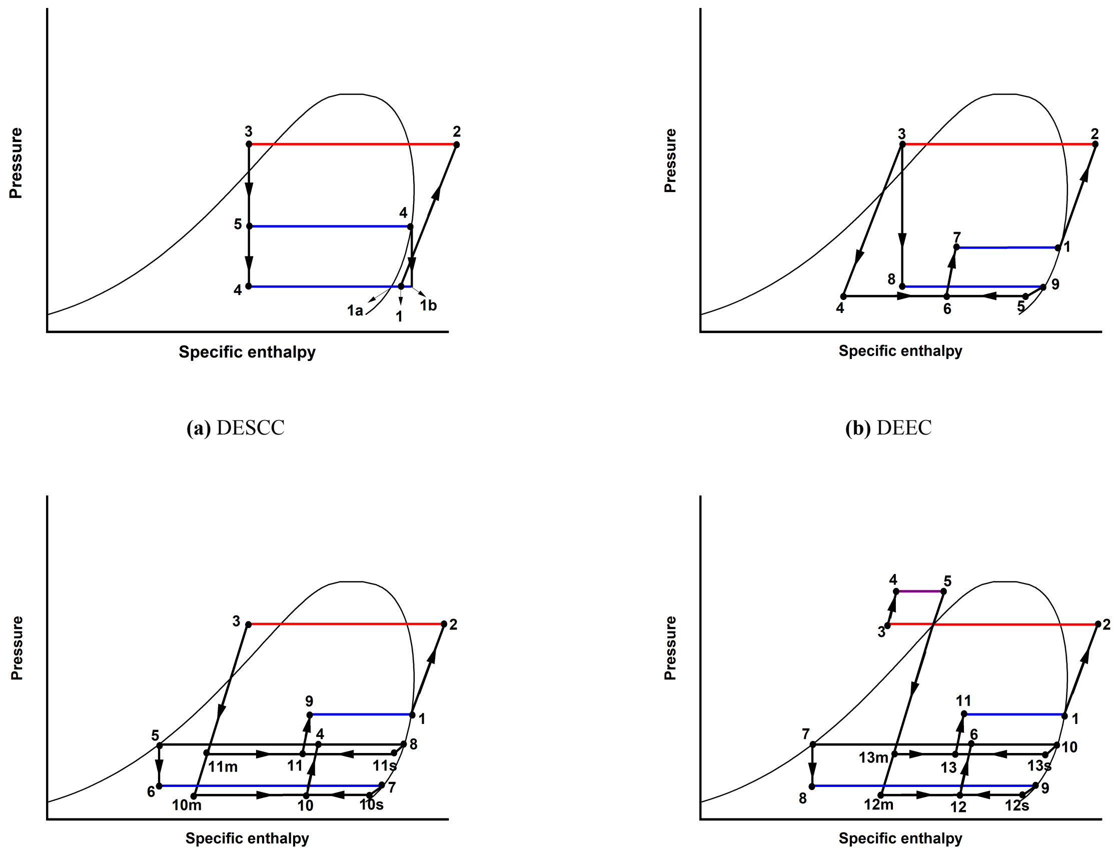
In this study, steady-state thermodynamic models were developed with state point analysis for R134a. Models have been developed on MATLAB using thermodynamic properties of fluids from REFPROP 10.

3.1 Ejector Modeling

For modelling two-phase ejector, the ejector efficiency definition provided by Elbel and Hrnjak (2008) is put to use. The entrainment ratio is related to the overall ejector efficiency as:

$$\omega = \eta_{ej} \frac{h_{mn,in} - h_{mn,out,s}}{h_{sn,out,s} - h_{sn,in}} = \eta_{ej} \frac{h_{mn,in} - h(P_{dif,out}, s_{mn,in})}{h(P_{dif,out}, s_{sn,in}) - h_{sn,in}} \tag{2}$$

For an adiabatic ejector, we can solve for the diffuser outlet state through an iterative method. By assuming the ejector



(a) DESCC **(b)** DEEC **(c)** DEDEC **(d)** SI-DEDEC

Figure 3: Representation of the P-h diagram for the four dual evaporator refrigeration cycles

to provide identical outlet conditions as it would should it be installed in an EERC, for a steady-state, the diffuser outlet's vapour quality determines the entrainment ratio. Thus the methodology described in Table 1 can be used to determine the diffuser outlet state, given the motive and suction inlet condition.

Table 1: Ejector model using overall ejector efficiency

Parameter	Equation
Iterated: Exit Temperature ($T_{dif,out}$)	$T_{dif,out}$ is assumed and corrected every loop
Exit Pressure ($P_{dif,out}$)	$P_{dif,out}$ is the corresponding saturation Pressure
State A: Isenthalpic compression of Motive Fluid to $P_{dif,out}$	$h_A = h_{mn,in}$; $P_A = P_{dif,out}$
State B: Isentropic compression of Motive Fluid to $P_{dif,out}$	$s_B = s_{mn,in}$; $P_B = P_{dif,out}$
State C: Isentropic expansion of Suction Fluid to $P_{dif,out}$	$s_C = s_{sn,in}$; $P_C = P_{dif,out}$
Entrainment Ratio, ω	$\omega = \frac{\dot{m}_{sn,in}}{\dot{m}_{mn,in}}$; $\omega = \eta_{ej} \frac{h_A - h_B}{h_C - h_{sn,in}}$

Enthalpy of exiting mixture, $h_{dif,out}$	$h_{dif,out} = (h_{mn,in} + \omega h_{sn,in})/(1+\omega)$, assuming adiabatic mixing process in the ejector
Quality of the exiting mixture, x_{mix}	Quality corresponding to $P_{dif,out}$ and $h_{dif,out}$
Exit Criteria for simulation:	$x_{mix} = \frac{1}{1+\omega}$, to ensure the steady state conditions

Compared to Kornhauser's (1990) model, this model provides the advantage of being computationally faster and requiring only the overall ejector efficiency, allowing a simpler sensitivity analysis. Lawrence and Elbel (2013) used the Konhauser model for the two-phase ejector to analyse EERC and DEEC cycles' performance. Thus, to arrive at an acceptable value for the overall ejector efficiency, the methodology in Table 1 was used to model an EERC and DEEC cycle, and results were compared with Lawrence and Elbel's (2013) results. For identical values of condenser and evaporator temperatures, degrees of subcooling and super-heat, and compressor efficiency, the overall efficiency of $\eta_{ej} = 0.365$ gave values of COP, ω and $T_{Evap,high}$ within sufficient accuracy. A comparison between the results of the two ejectors models has been presented in Table 2.

Table 2: Comparing results of Kornhauser model and the model presented in Table 1

Parameter	**EERC**		**DEEC**	
	Kornhauser	Elbel & Hrnjak	Kornhauser	Elbel & Hrnjak
$T_{ev,LT}$	5	5	5	5
ΔT_{SH}	5	5	5	5
T_{cn}	45	45	45	45
ΔT_{SC}	3	3	3	3
η_{cp}	0.75	0.75	0.75	0.75
η_{mn}	0.8	-	0.8	-
η_{sn}	0.8	-	0.8	-
η_{dif}	0.75	-	0.75	-
η_{ej}	-	0.365	-	0.365
$T_{ev,HT}$	7.4	7.40	7.4	7.40
COP	4.6	4.63	4.6	4.66
Load Ratio	-	-	1.38	1.39
ω	0.7	0.73	0.7	0.73

3.2 Refrigeration Cycle Modeling

A state point analysis has been used for all the cycles considered. The ejector is modeled as described in the section before, calculating the outlet state using the two inlet states. The following assumptions were made for the refrigeration system:

- Pressure drop and heat transfer are neglected in the pipeline.
- Changes in the kinetic and potential energy of the working fluid is negligible.
- The throttling process is entirely isenthalpic.
- All heat exchanges have negligible pressure drop and have a constant exit state.
- A constant compressor and ejector total efficiency is assumed.
- Steady state of the ejector outlet is dependent on the suction and motive inlet states and independent of the cycle it is implemented in (ie- EERC, DEEC or DEDEC).

The cycle parameters for analysing all the cycles are presented in Table 3.

Table 3: Values for different cycle parameters

Cycle Parameter	Value
Isentropic Compressor Efficiency, η_{cp}	75%
Overall Ejector Efficiency, η_{ej}	35%, but varied for sensitivity analysis.
Evaporator Degree of Super-heat, ΔT_{SH}	$5°C$
Condenser Degree of Sub-cool, ΔT_{SC}	$3°C$
Lower Evaporator Temperature, T_{ev}	$-20°C$ and $-15°C$, case-wise
Condenser Temperature, T_{cn}	$50°C$

The model to analyse the steady state performance of the DEDEC without the by-pass uses the definitions and equations described in Table 4. Ejector 1 ($ej1$) and Ejector 2 ($ej2$)) are the ejectors in direct connection with LT and HT evaporators respectively.

Table 4: Models of the Cycle Components for DEDEC

Component and Parameter	Equation
Compressor Output State	$h_{out} = h_{in} + (h_{in} - h(P_{out}, s_{in}))/\eta_{cp}$
Compressor Power Consumption	$W_{cp} = \dot{m}_{cp}(h_{out} - h_{in})$
LT Evaporator Heat Absorption	$Q_{ev,LT} = \dot{m}_{ev,LT}(h_{out} - h_{in})$
HT Evaporator Heat Absorption	$Q_{ev,HT} = \dot{m}_{ev,HT}(h_{out} - h_{in})$
Condenser Heat Rejection	$Q_{cn} = \dot{m}_{cn}(h_{in} - h_{out})$
Coefficient of Performance	$COP = (Q_{ev,LT} + Q_{ev,HT})/W_{cp}$
Mass Balance	$\frac{1}{\omega_{ej1}}\dot{m}_{ev,LT} = \dot{m}_{sn,ej2} = \frac{\omega_{ej2}}{1+\omega_{ej2}}\dot{m}_{ev,HT}$; $\dot{m}_{ev,HT} = \dot{m}_{cn} = \dot{m}_{cp}$
(Minimum) Load Ratio	$LR = Q_{ev,HT}/Q_{ev,LT}$

The same equations are used to study the other cycles as well. For DESCC, however, the Load Ratio is also an input for the thermodynamic model. This procedure is analogous to that in Lawrence and Elbel (2013), apart from the modeling ejector discharge state.

For analysing the SI-DEDEC, the the pump and generator's addition must be accounted for in the model. The paper assumes that the process of pumping and heating is such that the same ΔT_{SC} is retained for the ejector motive fluid.

Table 5: Additional and Modified Component Models for SI-DEDEC

Component and Parameter	Equation or Description
Pump Outlet Pressure	$P_{out} = P_{sat}(T = T_{gen} + \Delta T_{SC})$
Pump Outlet State	$h_{out} = h_{in} + (h_{in} - h(P_{out}, s_{in}))/\eta_{pmp}$
Pump Power Consumption	$W_{pu} = \dot{m}_{pu}(h_{out} - h_{in})$
Generator Heat Addition	$Q_{gen} = \dot{m}_{gen}(h_{out} - h_{in})$

Mass Balance	$\dot{m}_{pu} = \dot{m}_{cn} = \dot{m}_{gen}$
Net Work	$W_{net} = W_{pu} + W_{cp}$
Coefficient of Performance	$COP = (Q_{ev,LT} + Q_{ev,HT})/(W_{cp} + W_{pu})$

The SI-DEDEC can be further modified by introducing a by-pass at the condenser outlet. However, unlike the case for DEDEC, the COP for the SI-DEDEC will change due to the by-pass introduction, as described in Table 6.

Table 6: Calculation of COP for an SI-DEDEC with by-pass

Cycle Parameter	Equation or Description
COP_1	COP of the SI-DEDEC without the by-pass
COP_2	COP of a VCC operating at the HT evaporator
LR and LR_{min}	Load ratios of the modified and unmodified SI-DEDEC
Total COP, COP_t	$\frac{1+LR}{COP_t} = \frac{1+LR_{min}}{COP_1} + \frac{LR-LR_{min}}{COP_2}$

4. RESULTS

Using the modeling methodology described in the previous section, the state properties and different cycle performance parameters have been calculated for the DEDEC. Results for the SI-DEDEC has also been calculated with the same cycle parameters keeping $T_{gen} = 57°C$ and $\Delta T_{SC} = 3°C$. These values have been calculated for T_{LTEvap} of $-20°C$ and $-15°C$ and compared with equivalent DESCC's in Table 7.

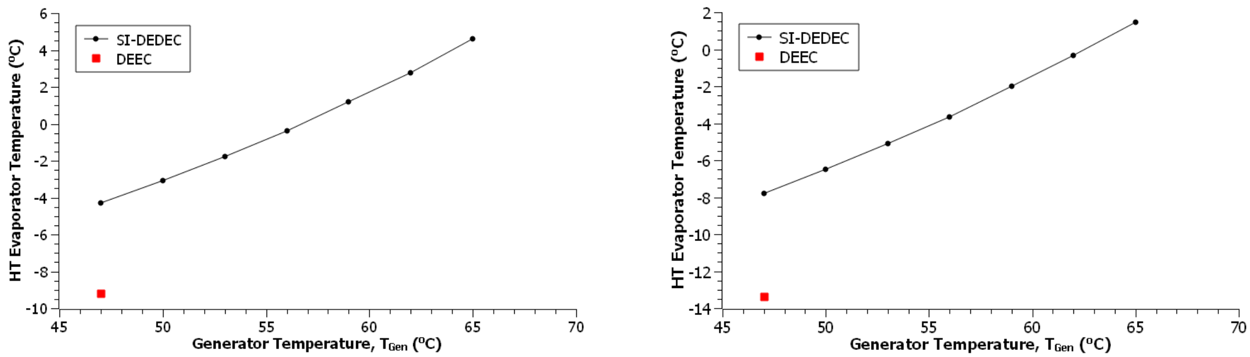
Table 7: Comparison of results of DEDEC and SI-DEDEC against equivalent DESCC's

Parameter	DEDEC	DESCC	SI-DEDEC	DESCC
$T_{LTEvap} = -20°C$				
T_{HTEvap}	$-7.78°C$		$-3.16°C$	
COP	2.56	1.90	2.22	1.91
LR or LR_{min}	1.75		1.92	
Q_{gen}/W_{net}	-	-	0.28	-
$T_{LTEvap} = -15°C$				
T_{HTEvap}	$-4.28°C$		$0.20°C$	
COP	2.82	2.14	2.41	2.15
LR or LR_{min}	1.67		1.89	
Q_{gen}/W_{net}	-	-	0.30	-

Table 7 shows that the DEDEC provides over a 30% increase in COP than the DESCC for either of the temperatures. In a typical domestic refrigerator, cooling at $-18°C$ and $4°C$ are required for the freezer and fresh-food compartments, respectively. Considering these temperature ranges, it is evident that the DEDEC has a strong potential in this domain. However, the applications are limited by the lower bound on the load ratio, LR_{min}.

Table 7 also shows that by integrating the generator, the HT evaporator temperature gets a boost of $4 - 4.5°C$ in either case. This could provide subtle advantages, such as avoiding going below the freezing point. This, however, comes with a compromise on the system's COP and the minimum limit on the load ratio. The increase in enthalpy of the motive fluid leads to an increased entrainment ratio. This would mean an increase in refrigerant flow through the

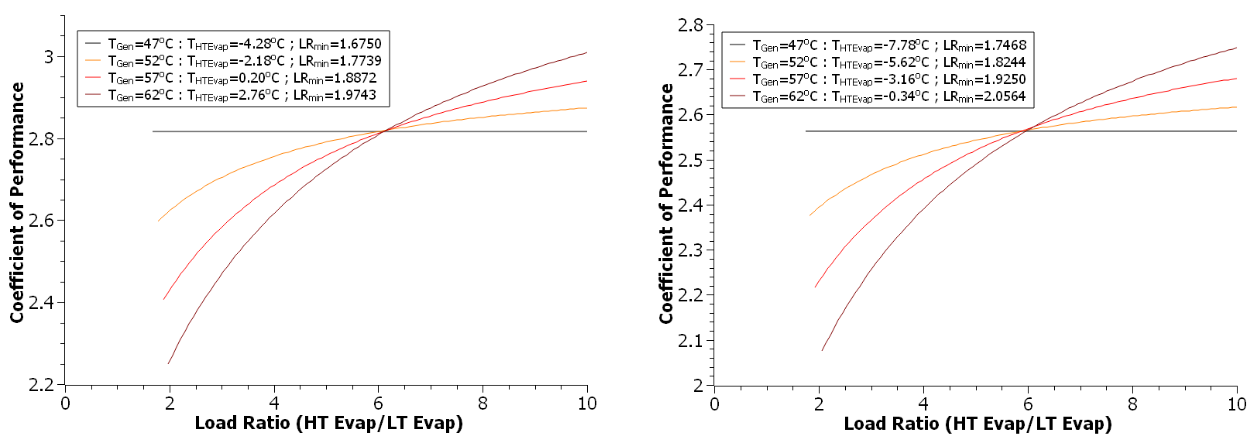
HT evaporator, leading to an increased steady-state load ratio. The heat required by the generator is calculated as 28-30% of the grid electricity requirement for the SI-DEDEC. Additionally, since the heat is required at a temperature of less than $60°C$, this integration is a practical process. At the minimum load ratios, i.e., without the by-pass, the COP improvement compared to equivalent DESCC's is brought down to around 12-16%. However, as discussed later on, the SI-DEDEC's will be a lot more advantageous for higher load ratios.



(a) $T_{LTEvap} = -15°C$ **(b)** $T_{LTEvap} = -20°C$

Figure 4: HT evaporator temperature variation for different generator temperature

The effect of the generator temperature on the SI-DEDEC's HT evaporator temperature has been presented in Figure 4. It must be noted that $T_{gen} = 47°C$ represents the case of no generator, which is the DEDEC. The additional boost of 4-$5°C$ in the HT Evaporator temperature for DEDEC against the DEEC is also evident in Figure 4.

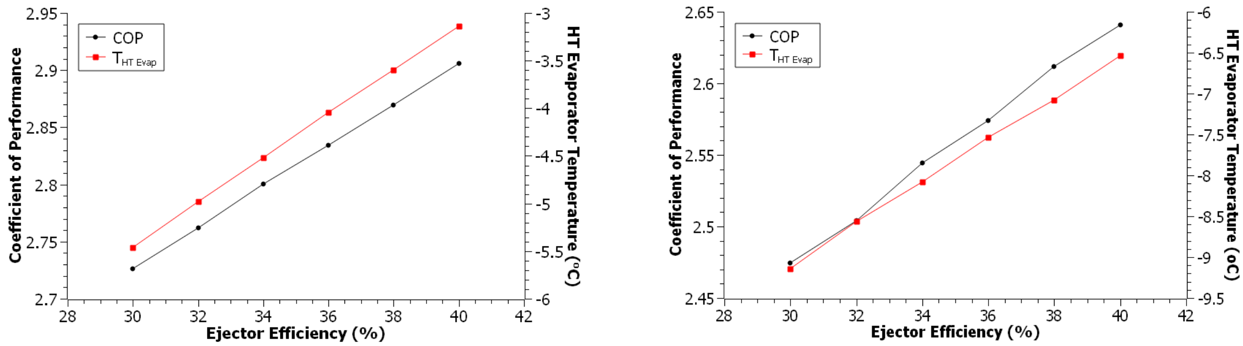


(a) $T_{LTEvap} = -15°C$ **(b)** $T_{LTEvap} = -20°C$

Figure 5: COP variation for different load ratio at different generator temperature

As described before, when the application requires a higher load ratio, we can modify the DEDEC and the SI-DEDEC by introducing a by-pass at the condenser outlet. While the modification does not alter the DEDEC's COP, SI-DEDEC's COP is bound to vary. Figure 5 presents the variation in the cycle's performance. It must be noted that higher generator temperature leads to a higher HT evaporator temperature, due to which the compressor will be acting across a smaller pressure difference. This leads to the superimposed VCC having a higher COP for a higher T_{gen} in an SI-DEDEC. Thus beyond a critical load ratio, the SI-DEDEC will be the favoured system than the DEDEC with identical LT evaporator temperature.

The ejectors being the key components of the DEDEC, it is vital to understand its influence on the system's perfor-



(a) $T_{LTEvap} = -15°C$ **(b)** $T_{LTEvap} = -20°C$

Figure 6: COP and HT evaporator temperature variation for varying ejector efficiency

mance. The paper assumes an over-all ejector efficiency of 35% for the calculations. However, the HT evaporator and total COP variation for the system have been plotted for the ejector efficiency varying from 30% to 40% in figure 6. Both $T_{evap,HT}$ and COP, are found to increase with the ejector efficiency consistently.

5. CONCLUSIONS

A dual evaporator dual ejector refrigeration cycle is proposed, with both ejectors operating as two-phase ejectors. Integration of low-grade heat (generator) into the cycle has also been explored. The paper also looks into adding a by-pass in the cycle, allowing flexibility on the ratio between the two evaporator loads. The influence of the ejector's efficiency, low-grade heat supplied and load ratio on the proposed cycle have been evaluated for R134a as the working fluid.

From the results summarized in Table 7, we can conclude:

- COP of the proposed cycle is significantly higher than the conventional single compressor dual evaporator cycle.
- The proposed cycle provides a greater temperature difference between the two evaporators than the conventional dual-evaporator ejector cycle.
- Upon integrating a generator, the mentioned temperature difference can be further increased. However, this comes with a decrease in COP for the cycle without by-pass.
- For load ratios beyond a critical value, integrating the generator and adding by-pass results in higher COP.
- For domestic refrigerators with compartment temperatures typically around -18°C and 4°C respectively, the proposed cycle proves to be an excellent alternative.

NOMENCLATURE

h	Specific Enthalpy	(kJ/kg)
m	Mass Flow Rate	(kg/s)
P	Pressure	(MPa)
Q	Heat Transfer Rate	(kW)
s	Specific Entropy	(kJ/kg-K)
T	Temperature	(°C)
W	Power Input	(kW)
x	Vapour Quality	(-)
η	Efficiency	(–)
ω	Entrainment Ratio	(–)
COP	Coefficient of Performance	(–)
LR	Load Ratio (= Q_{HT}/Q_{LT})	(–)

Abbreviations

DEDEC	Dual Evaporator Dual Ejector Cycle
DEEC	Dual Evaporator Ejector Cycle
DESCC	Dual Evaporator Single Compressor Cycle
EERC	Ejector Expansion Refrigeration Cycle
HT	High Temperature
LT	Low Temperature
SC	Subcool
SH	Superheat
SI-DEDEC	Solar Integrated - DEDEC
VCC	Vapour Compression Cycle

Subscript

cn	Condenser
Cond	Condenser
cp	Compressor
dif	Diffuser
ej	Ejector
ev	Evaporator
Evap	Evaporator
gen	Generator
in	Inlet State
min	Minimum
mix	Mixture
mn	Motive Nozzle
out	Outlet State
pu	Pump
s	Isentropic Process
sn	Suction Nozzle
sat	Saturation
t	Total

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