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Research Article

Numerical analysis of a two-phase injection refrigeration cycle using R32

Praveen ALOK[1], Debjyoti SAHU[2*]

[1]*Former M. Tech Student, Amrita Vishwa Vidyapeetham, Bengaluru, 560035, India*
[2]*School of Mechanical Engineering, Kalinga Institute of Industrial Technology, Bhubaneswar, 751024, India*

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ABSTRACT

The present paper reports the performance of a popular refrigerant R32 (Difluoromethane, CF_2H_2) experiencing the two phase injection process. Two phase injection process may lower the discharge temperature of a multistage compressor. In order to investigate the role and impact of two-phase injection on a compressor, a Scroll compressor is selected because scroll compressor has high tolerance for liquid refrigerant. A reputed compressor is chosen where all the operating conditions and specifications are available in public domain. The modelling and analysis of refrigeration system is carried out using a simple MATLAB code. Around 200 iterations were performed for four different condensing and evaporating temperatures. The maximum reduction in discharge temperature is found to be 44°C when compared to R410A used in the same system.

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INTRODUCTION

Since the first vapor compression refrigeration system was made by Jacob Parking in 1834, a large number of chemical substances have been tried as working fluid like ammonia, methyl chloride, carbon dioxide etc. [1]. After the invention of dichloromethane in 1930 by Thomas Midgely and Albert Henna chlorofluorocarbons (CFCs) and hydrofluorocarbons (HCFCs) became the dominant types of refrigerants. R11 and R12 were the most widely used refrigerants [1]. When HVAC industry started to grow at a faster rate, the production of these refrigerants exceeded many million tons, the high production of these refrigerants was seen as a cause of concern as these refrigerants were found responsible for ozone depletion and global warming [2]. Due to environmental concerns the production of R11 and R12 has been stopped. This led to their replacement with R22. But as R22 is also responsible for ozone depletion many R22 alternatives were introduced into the market such as R410A, R407C etc. which are non-ozone depleting refrigerants but have very high global warming potential (GWP). For example, R410A has a GWP of 2088 on a 100

*__Corresponding author.__
*E-mail address: debjyothi@gmail.com (corresponding author)

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year time frame (IPCC, 2007) [3], also needs to be phased out. Now, the best alternative for R410A is R32 which is a non-ozone depleting refrigerant with global warming potential one third of R410A (GWP = 675 on a 100 year time frame) [4]. The pressure ratio of R32 similar to R410A [5]. Taira *et al.,* [5] suggested that among various refrigerants R32 can be retrofitted immediately. In order to gauge the performance potential of R32, Lemmon *et al.,* [6] did the thermodynamic assessment of R32 and R410A in ideal vapor compression cycle using REFROP. They found that R32 has 10% more volumetric efficiency than R410A and refrigeration capacity per mass is 52% greater than R410A. They also found that discharge temperature is 13°C higher than R410A. Compressor discharge temperature of R32 seems to be high in regular compressor so the present work proposes the use of two-phase injection in a scroll compressor. Numerical simulation is performed using NIST REFROP data to calculate discharge temperature. A simple MATLAB code links simulation with NIST data [7].

In order to decrease the discharge temperature of refrigerants various method can be used such as vapor injection, liquid injection and two-phase injection etc. [8]. Vapour injection Cycle has been studied by many researchers. Wang *et al.,* tested the flash tank at -18°C and found that maximum COP improvement to 23% [9]. Currently popular refrigerants R22 and R410A are going to be phased out in few years from now. Recently Xu *et al.,* did performance comparison of R32 and R410A in vapour injection cycles and concluded that the capacity and coefficient of performance (COP) improvements using R32 can reach up to 10% and 9%, respectively and R32 is the best alternative to replace R410A in terms of performance [10]. System performance for R32 can be further enhanced by component optimization using an identical cycle that uses R410A.

Refrigeration cycle working with evaporator temperature 1 to 8°C and a condenser temperature ~65°C should be given importance as they are practical values of summer air conditioning. Simulation work related to two-phase injection cycle suitable for air conditioning application is rare because there are multiple mathematical equations involved [11]. Simulation of Two-phase injection refrigeration cycle employing refrigerant R32 is worth investigating as R32 is expected to dominate the Air conditioning market in this decade. The main aim of the present work is to analyze the effect of two-phase injection in decreasing the discharge temperature of R32 and its effect on the COP of the refrigeration system with a scroll compressor.

THEORETICAL CALCULATION

Performance of Refrigeration System

Before simulating two-phase injection thermodynamic assessment of R32 is done in ideal refrigeration cycle using actual operating conditions obtained from the compressor manufacturer (Model YP292K1T-TND**, R32 COPELAND SCROLL™3-50)** [12]. A sample calculation can be interesting. A popular scroll compressor was chosen for doing this calculation (Table 1–3) which is used for air-conditioning. The operating conditions are taken from the manufactures webpage. However the operating conditions which

Table 1: Operating conditions of YP292K1T-TND COPELAND™ SCROLL-3-50 [12]

Sl. No.	Parameter	value	Value in SI unit
1	Evaporator temp. T_{evap}	35˚F	274.66 K
2	Condenser temp. T_{cond}	140˚F	333 K
3	Refrigerant flow rate m	1949 lbm/hr	0.25kg/s
4	Isentropic efficiency η_{ise}	70.5%	–
5	Degree of superheat ΔT_{sup}	18˚F	11.11 K
6	Degree of sub-cooling ΔT_{sub}	0˚F	0 K
7	Compressor cooling capacity	1,72,500 Btu/hr	50.55 kW
8	Energy efficiency ratio, EER	7.3	–

Table 2: Refrigerant R32 flow calculation for compressor

Compressor Inlet		Compressor Outlet	
Temperature of the refrigerant	12.66°C	Pressure	39.332 bar at T_{cond} = 60°C
Pressure	8.572 bar	Enthalpy	625.22 kJ/kg (S_1 = 1.1001 kJ/kgK)
Enthalpy	528.73 kJ/kg	h_{2s}	596.76 kJ/kg
Entropy	2.1952 kJ/kg-K	T_2	137.26°C

Table 3: Refrigerant R32 flow calculation for condenser and capillary tube

Condenser outlet		Capillary tube outlet	
Temperature $T_3 = T_{cond} - \Delta T_{sub}$	60°C – 0°C = 60°C	Enthalpy $h_4 = h_3$	321.93 kJ/kg
Enthalpy h_3	321.93 kJ/kg (since $P_3 = P_2$ at h_3)	Pressure $P_4 = P_1$	8.572 bar
Pressure P_3	39.332 bar	Temp. T_4	1.66°C

Table 4: Theoretically calculated isentropic efficiency and energy efficiency ratio [12]

Sl No	Operating Conditions (COPELAND SCROLL™3-50)	Cooling Capacity (Btu/hr.)	Power (watts)	Mass flow Rate (lbm/hr.)	EER	Isentropic efficiency (%)
1	T_{cond} = 130°F, T_{eva} = 35°F	1,87,000	21,100	1981	8.9	72.9
2	T_{cond} = 130°F, T_{eva} = 45°F	2,26,000	21,500	2,374	10.5	74
3	T_{cond} = 130°F, T_{eva} = 50°F	2,47,000	21,600	2,593	11.4	74.1
4	T_{cond} = 140°F, T_{eva} = 35°F	1,72,500	23,600	1949	7.3	70.5
5	T_{cond} = 140°F, T_{eva} = 45°F	2,08,000	24,000	2,333	8.7	72.2
6	T_{cond} = 140°F, T_{eva} = 50°F	2,27,000	24,200	2,548	9.4	72.7
7	T_{cond} = 140°F, T_{eva} = 60°F	2,72,000	24,600	3.029	11.1	73

the manufacturers of the compressors use are under standard test conditions. These are common to test any Air-conditioning system. The manufacturer has also provided the mass flow rate of R32 and cooling capacity of the system at various operating conditions.

Isentropic efficiency (ratio of work input to isentropic process and actual process) [11] is given in Eq. 1.

$$\eta_{isen} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{1}$$

Cooling capacity is the system's ability to remove heat [11] and given in Eq. 2.

$$q = m \times (h_1 - h_4) \tag{2}$$

Coefficient of performance and energy efficiency ratio are important parameters [11] and given in Eq. 3 and Eq. 4.

$$COP = \frac{Refrigeration\ effect}{Work\ done} \tag{3}$$

EER is the ratio of cooling capacity in BTU per hour and electrical energy input in watt,

$$EER = 3.415 \times COP \tag{4}$$

Therefore referring to Table 2 and 3, η_{isen} = 0.705;

Refrigeration effect = $h_1 - h_4$ = 206.8 kJ/kg;

Cooling capacity = 0.24557 × 206.8 = 50.7838 kW = 173281.53 Btu/hr.

There is 0.4% difference between cooling capacity given by the manufacturer and the cooling capacity calculated here.

$COP = \frac{137.06}{70{,}342} = 2.1431$ and energy efficiency ratio, EER= 7.31. EER rating of the compressor is given as 7.3 (Table 1) which is similar to calculated value of 7.31. The value for isentropic efficiency is also available for different operating conditions. The discharge temperature of R32 is 137.6°C for given working condition therefore steps should be taken to lower the discharge temperature of R32 for the safe working of the compressor. High discharge temperature often leads to the compressor failure due to the degradation of the compressor oil [13, 14]. Volumetric efficiency and isentropic efficiency of the compressor affect the performance of any refrigerator or air conditioner. Volumetric efficiency controls the refrigerant flow rate that will pass through the compressor. Low volumetric efficacy eventually decreases the mass flow through the evaporator thus decreases the cooling capacity of the system.

Volumetric efficiency is inversely proportional to the pressure ratio. Due to low volumetric efficiency at high pressure ratio, the cooling capacity decreases as the pressure ratio increases further. However isentropic efficiency affects the COP of the refrigeration systems by affecting the work done by the compressor. Table 4 comprises of change in cooling capacity, power requirement, mass flow rate, EER and isentropic efficiency of the same compressor at

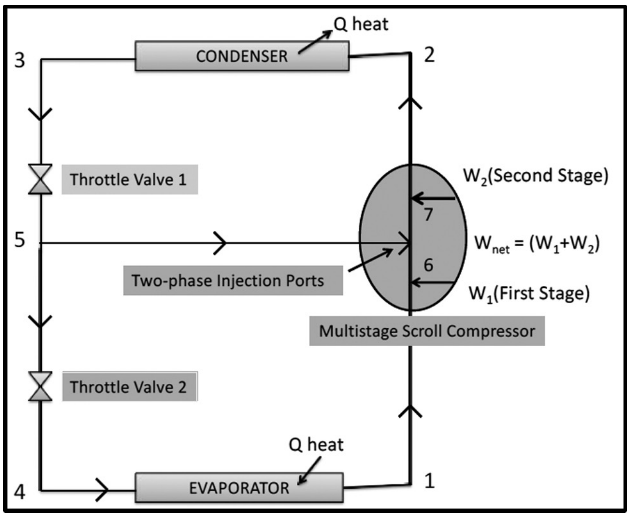

Figure 1. Schematic diagram for multi stage refrigeration system.

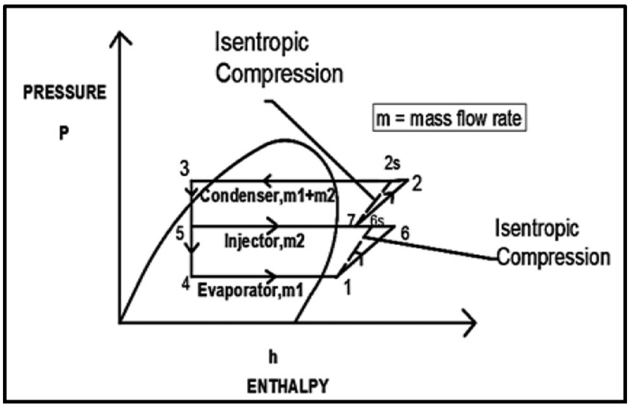

Figure 2. Pressure Enthalpy Chart for two-phase injection [19].

different operating conditions as mentioned in the manufacturer's data sheet.

The cooling capacity, isentropic efficiency, mass flow rate at operating condition 1 and at operating condition 4 are low though the power rating is continuously increasing through conditions 1 to 7. This shows that as the pressure ratio increases system performance decreases. The decrease in mass flow rate is due to the decrease in volumetric efficiency as the pressure ratio increases [15]. The decrease in EER of the system is due to decrease in isentropic efficiency of the compressor at higher pressure ratio [8]. The power required by the compressor increases as the pressure ratio increases. The sample calculation was done on operating condition 4.

Two-Phase Injection System

In a two-phase injection refrigeration system a two stage compressor is used (Figure 1). The refrigerant (mass flow rate = m_1) after getting compressor at first stage gets mixed with the low-temperature two-phase refrigerant (mass flow rate = m_2) coming out from first expansion device. The refrigerant mass flow rate after getting mixed (mass flow rate = $m_1 + m_2$) gets into the second stage of compressor and is compressed till condenser pressure [16–18]. The refrigerant after getting condensed enters the first expansion device where the refrigerant is depressurized till the discharge pressure of first stage of compression (Figure 2). Refrigerant passing through the first expansion device splits into two-parts. One part (m_2) gets into the injector and other part gets into the second expansion device where the refrigerant is further depressurized till evaporator pressure. A two-phase injector can be used to inject the refrigerant coming out of the first expansion device to the compressor at the end of first stage. The mixing of refrigerant decreases the temperature of the refrigerant vapor. A mathematical model can be developed to predict the discharge temperature, COP of the system. This mathematical model of vapor compression refrigeration system with two-phase injection system can be coded in MATLAB to recall the parameters of REFPROP.

MATHEMATICALMODELLING OF TWO-PHASE INJECTION SYSTEM

Compression pressure ratio (P_r) is an important factor in design of the compressor. The present work involves two-stage compression, so there will be two compression ratios for two stages [11]. However, the pressure ratio of both the stages can be same or different. The pressure ratio of first compressor is given as P_{r1} and pressure ratio of second stage is given as P_{r2} ranging from 0 to 6.

$$P_{r1} = P_{inj}/P_1 \tag{5}$$

$$P_{r2} = P_2/P_{inj} \tag{6}$$

In addition the pressure which is important in calculation of pressure ratio of 1st stage of compression is called injection pressure ratio [11] which is defined as shown in Eq. 7.

$$r_p = \frac{P_{inj} - P_1}{P_2 - P_1} \tag{7}$$

r_p can vary from 0.001 to 0.999. If $r_p = 1$, $P_{inj} = P_2$ and two-stage cycle will then become single stage and $0 < r_p < 1$. Optimization of r_p is another objectives of this work. It decides the pressure at which injection will take place vis-a-vis the discharge pressure of first stage of compression.

Table 5: Inlet conditions at first stage:

Sl. No.	Parameter	Symbol
1	Pressure	P_1 ($T = T_1$°C, x = 1)
2	Enthalpy	h_1 ($P = P_1$, $T = T_1$)
3	Entropy	S_1 ($P = P_1$, $T = T_1$)

Another important parameter with two-phase injection system is the mass flow ratio of injection and it is given in Eq. 8 [11].

$$m_r = \frac{mass\ of\ injection}{total\ mass\ flow\ rate} \tag{8}$$

Here m_r can vary from 0.001 to 0.999. If m_r will become zero then in that case no refrigerant will be injected to the compressor at an intermediate stage and the cycle will become single stage system instead of two-stage. If the value of m_r becomes 1 then no refrigerant will get into the second expansion device and no-refrigeration effect will be produced at the evaporator. Optimization of m_r is also an objective of this work.

First Stage of Compression

Inlet condition to the first stage of compressor is represented in Table 5. The injection pressure P_{inj} can be calculated after selecting certain value of pressure ratio, r_p. Pressure P_2 for the refrigerant must be found at $T = T_{cond}$ and x = 0. Using Eq. 7 injection pressure can be found; $P_{inj} = r_p \times (P_2 - P_1) + P_1$ and $P_6 = P_{inj}$. Enthalpy h_{6s} must be found at $P = P_{inj}$; $S_{6s} = S_1$.

First stage compressor efficiency as shown in Eq. 9 [19].

$$\eta_1 = \frac{h_{6s} - h_1}{h_6 - h_1} \tag{9}$$

Enthalpy after compression process as shown in Eq. 10 [19].

$$h_6 = \frac{h_{6s} - h_1}{\eta_1} + h_1 \tag{10}$$

Temperature T_6 can be found at $P = P_{inj}$ and enthalpy $h = h_6$;

Work input at first stage compressor [19] = $m_1 \times (h_6 - h_1)$

$$\text{Work input at first stage compressor [19]} = m_1 \times (h_6 - h_1) \tag{11}$$

Mixing of Fluids

Injection mass flow ratio m_r varies from 0.001 to 0.999, m_{inj} can be calculated after specifying the value of m_r. Therefore refrigerant injected $m_{inj} = m_2 = m_r \times m_{total}$ that follows the mass balance equation, $m_{total} = m_{inj} + m_1$

Energy Balance

From energy balance concept if m_t is the total enthalpy then, $m_1 \times h_6 + m_{inj} \times h_5 = m_{total} \times h_7$

Therefore enthalpy after first stage (Figure 2),

$$h_7 = \frac{m_1 * h_6 + m_{inj} * h_5}{m_{total}} \tag{12}$$

Saturation temperature T_5 can be be found at $P = P_{inj}$ and x = 1; whereas T_7 to be found using the relation $h = h_7$. As per COPELAND compressor requirement minimum degree of superheat at the entry of compressor should be 11.11°C; $T_{sup} = (T_7 - T_5) \geq 11.11$°C.

Second Stage of Compression

Pressure ratio r_{p2} can be obtained using Eq. 6; now, isentropic efficiency of second stage compression can be found using Eq. 13 [19].

$$\eta_2 = \frac{h_{2s} - h_7}{h_2 - h_7}. \tag{13}$$

Where second stage compressor temperature T_2 corresponds to $P = P_2$, $h = h_2$ and work done at second stage = $m_{total} \times (h_2 - h_7)$

Condenser

Sub cooling happens in the condenser and the temperature [19],

$$T_3 = T_{cond} - \Delta T_{sub} \tag{14}$$

Where pressure $P_3 = P_2$ and h_3 = Enthalpy corresponds to $P = P_3$, $T = T_3$.

First Expansion Tube

Use the energy balance concept to refrigerant flow [19],

$$m_{total} \times h_3 = m_{total} \times h_5 \tag{15}$$

Where pressure $P_5 = P_{inj}$ and T_5 = Temperature corresponds to $P=P_{inj}$, $h=h_5$.

Second Expansion Tube

Similarly use the energy balance concept to refrigerant flow [19],

$$m_1 h_5 = m_1 h_4 \tag{16}$$

Where pressure $P_4 = P_1$ and Temperature = T_4 corresponds to $P = P_1$, $h = h_4$.

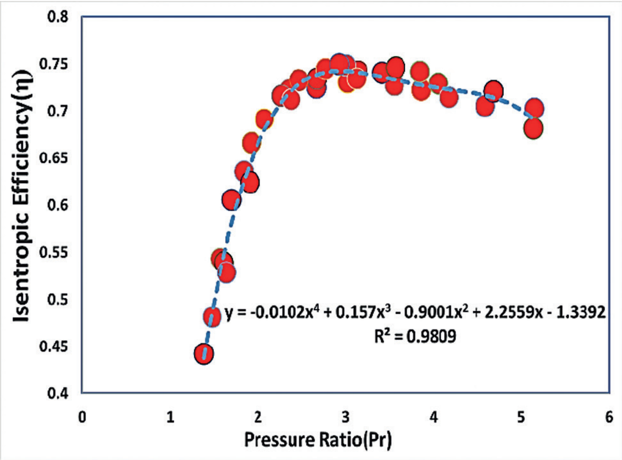

Figure 3. Variation of Isentropic efficiency with Pressure Ratio.

Evaporator

Use Eq. 2 to find refrigeration effect and use Eq.3 and Eq.4 to get COP and EER respectively.

Pre-validation

Isentropic efficiency is a function of pressure ratio so relationship between isentropic efficiency and pressure ratio should be established for the compressor. Isentropic efficiency of the compressor is provided by the manufacturer according to various pressure ratios and operating condition [12]. The data of isentropic efficiency of the compressor is plotted against the pressure ratio to obtain a relationship between the isentropic efficiency and pressure ratios. Figure 3 shows the relationship between the pressure ratios of the Copeland compressor. Isentropic efficiency is taken along y axis and pressure ratio along x axis. Now the isentropic efficiency of the two stages of the compressor can be expressed as polynomials given in Eq. 17 and Eq. 18 respectively.

$$\eta_1 = -0.0102P_{r1}^{\ 4} + 0.157P_{r1}^{\ 3} + 0.9001P_{r1}^{\ 2} - 2.2559P_{r1} + 1.3392 \tag{17}$$

$$\eta_2 = -0.0102P_{r1}^{\ 4} + 0.157P_{r1}^{\ 3} + 0.9001P_{r1}^{\ 2} - 2.2559P_{r1} + 1.3392 \tag{18}$$

Interestingly, experimental isentropic efficiency is 0.705 whereas isentropic efficiency with best fit graph is 0.7069; thus it pre-validates this model (Eq.1 to Eq. 18). The mathematical model is coded in MATLAB. In order to test the accuracy, the EER of experimental cycle is compared with the EER obtained from the code. Values are as shown in table 6.

Table 6: EER and discharge temperature values calculated

Experimental	Ideal Cycle value (calculated using Eq.4)	MATLAB code value
EER = 7.3	EER = 7.31	EER = 7.33
Discharge Temp. not given	Discharge Temp. 137.26°C	Discharge Temp. 137.06°C

In order to convert two-phase injection cycle to single stage cycle r_p is considered as 0.9999 and m_r as 0.0001 respectively. The percentage difference between the experimental EER and EER obtained from MATLAB code is ~0.4%. The difference is due to the fact that isentropic efficiency used in the MATLAB code is obtained from best fit equation (Eq. 17 and 18) which predicts the isentropic efficiency value with ~98.1% accuracy.

RESULTS AND DISCUSSIONS

After validating the COP/EER value with COP obtained from MATLAB code, various values of r_p and m_r are selected ranging from 0.001 to 0.999 to simulate four different cases with varying condenser and evaporator temperature.

Case 1: T_{cond} = 60°C and T_{eva} = 1.66°C

The variation of COP with respect to the pressure ratio for mass injection ratio m_r = 0.05, 0.1, 0.125 and 0.15 fraction of total mass flow rate flowing through the injector (at intermediate pressure ratio r_p varying from 0.001 to 0.999 in 50 steps) is shown in Figure 4. Fifty values of COP are obtained by the MATLAB code for 50 different intermediate pressure ratios. The maximum COP obtained for m_r= 0.05, 0.1, 0.125 and 0.15 are 2.0402, 2.0548, 2.0729 and 2.0917 respectively (at intermediate pressure ratio of 0.32). The value of COP increases as mass injection ratio increases [11]. The orange line in Figure 4 represents the COP of single stage cycle (COP = 2.1491) without two-phase injection. The COP obtained in said condition are 5%, 4.38%, 3.54% and 2.67% lesser than COP of refrigeration system of same capacity but using a single stage compressor.

Figure 5 shows the variation of compressor discharge temperature with respect to 50 values of intermediate pressure ranging from 0.001 to 0.999 for m_r= 0.05, 0.1, 0.125 and 0.15. The minimum discharge temperature obtained at the r_p = 0.32 and m_r = 0.05, 0.1, 0.125 and 0.15 are 128.59°C, 114.80°C, 107.55°C and 100.12°C respectively. This clearly shows that discharge temperature decreases when mass injection ratio increases at intermediate pressure ratio of 0.32. Red circles at the top of Figure 5 represents the discharge temperature without two-phase injection. The discharge temperature without two phase injection is as high

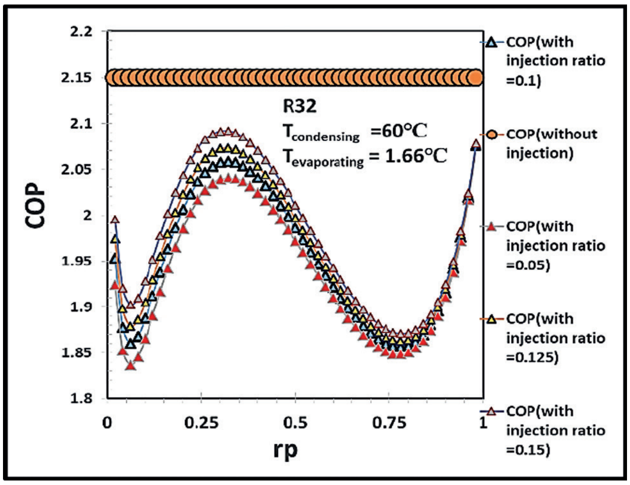

Figure 4. Variation of COP with respect to Intermediate Pressure Ratio.

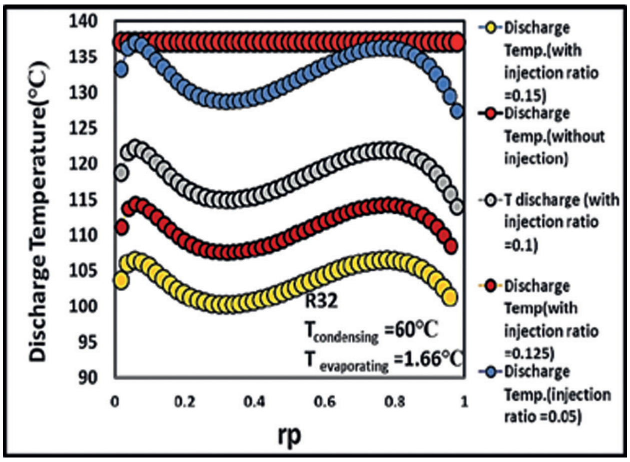

Figure 5. Variation of Discharge Temp with Intermediate Pressure Ratio.

as 137.02°C. It's safe to operate R32 at low discharge temperature as R32 is slightly inflammable [20, 21]. Vali *et al.*, reported a discharge temperature of 110.18°C with COP value of 4.245 where the condenser and evaporator temperature assumed to be 54.4°C and 7.2°C respectively (superheat temperature = 10°C, Sub cooling 5°C) [22]. As per scroll compressor specification, R32 will not work in this condition [11]. Pramudantoro *et al.*, tested and reported maximum COP for R32 is 2.11 for a 0.75 TR air conditioner [23].

Variation of degree of superheat at the inlet of second stage of compressor with respect to 50 different values of r_p ranging from 0.001 to 0.999 can be plotted in a graph. The degree of superheat should be near to 11.11°C (~18 °F) at the inlet of each stage. Figure 6 shows the variation in degree of superheat at the inlet of second stage of compression with respect to intermediate pressure. Therefore degree of superheat needed theoretically are 33.5683°C, 21.6453°C, 15.5282°C and 9.3973°C respectively for given mass injection ratio, m_r. Here the only value corresponding to the degree of superheat <11.11°C is for m_r = 0.15. Thus the degree of superheat required for safer working of compressor limits the mass injection ratio to 0.15 or less than that.

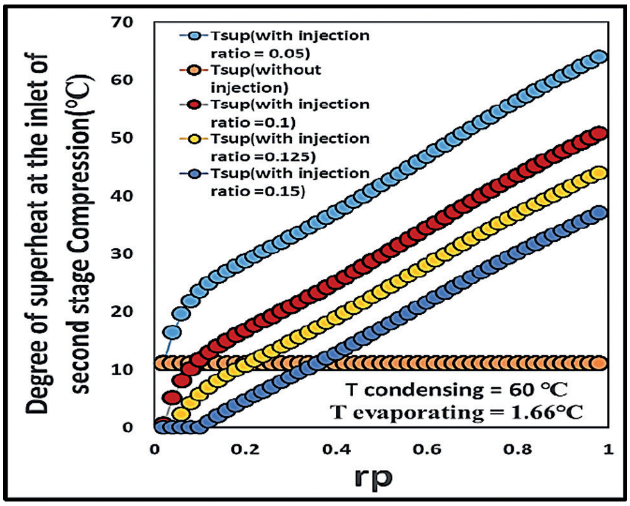

Figure 6. Variation of T_{sup} with Intermediate rp Ratio.

Case 2: T_{cond} = 60°C and T_{eva} = 7.22°C

Now, the evaporator temperature is increased to 7.22°C keeping condenser temperature constant at 60°C. Interestingly, the experimental isentropic efficiency = 0.722 and isentropic efficiency obtained from best fit graph = 0.7225. The difference between the experimented isentropic efficiency and isentropic efficiency is negligible. Various values of COP of the cycle with respect to fifty values of intermediate pressure ratio ranging from 0.001 to 0.999 for m_r = 0.05, 0.1, 0125 and 0.15 is shown in Figure7. The maximum COP obtained are 2.2343, 2.2255, 2.2713 and 2.2916 respectively at an intermediate pressure ratio of 0.34; much higher than values correspond to evaporator temp. of 1.66°C. Yildirim *et al.*, did a theoretical study to retrofit R410A with R32 and reported a COP >5 when the evaporator temperature is >10°C which is an irrelevant temperature for any practical use [24]. Refrigerant in any air conditioner evaporator will be at <7.22°C.

The minimum discharge temperature obtained at the r_p 0.34 for said values of m_r are 123.68°C, 110.37°C, 103.42°C and 96.31°C respectively as shown in Figure 8. The degree of superheat should be maintained for mass injection ratio

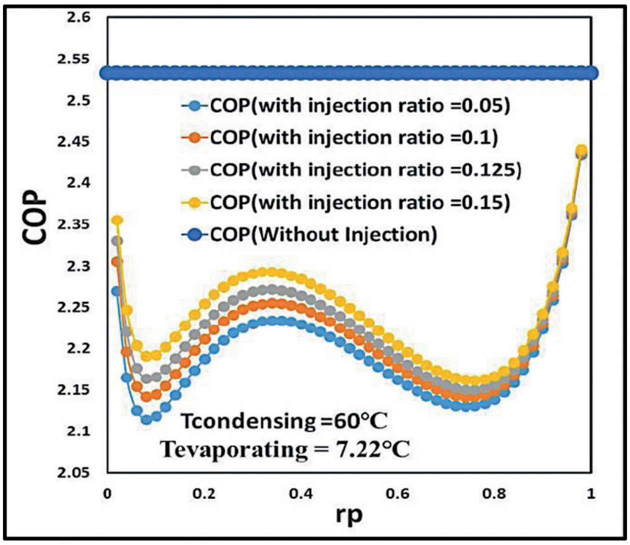

Figure 7. Variation of Degree of Superheat with respect to intermediate pressure ratio.

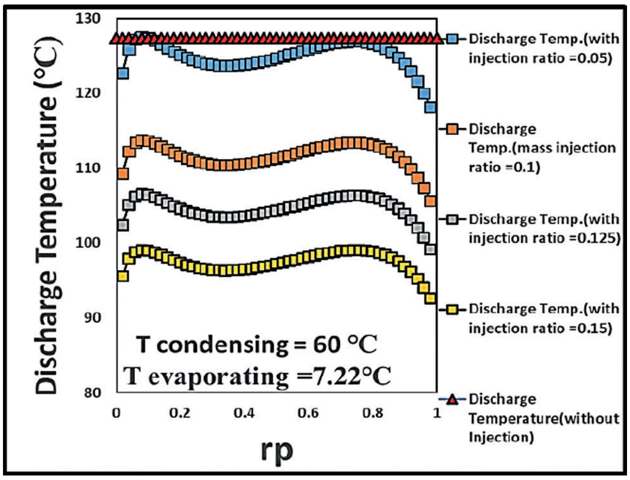

Figure 8. Variation of Discharge Temp with intermediate pressure ratio.

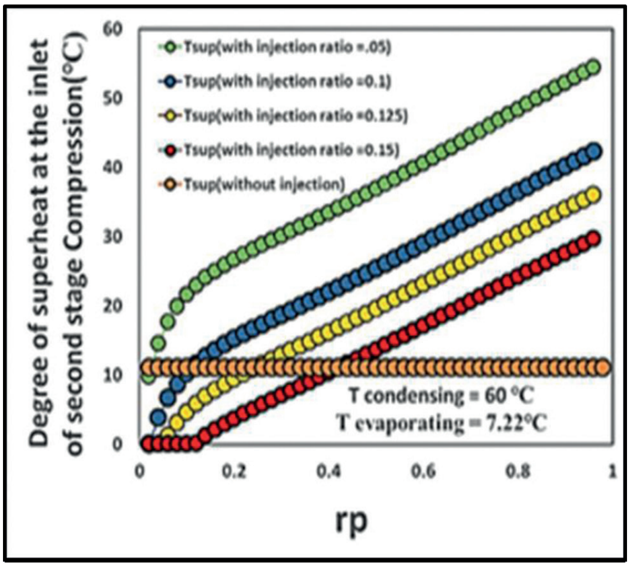

Figure 9. Variation of superheated temperature with intermediate pressure ratio.

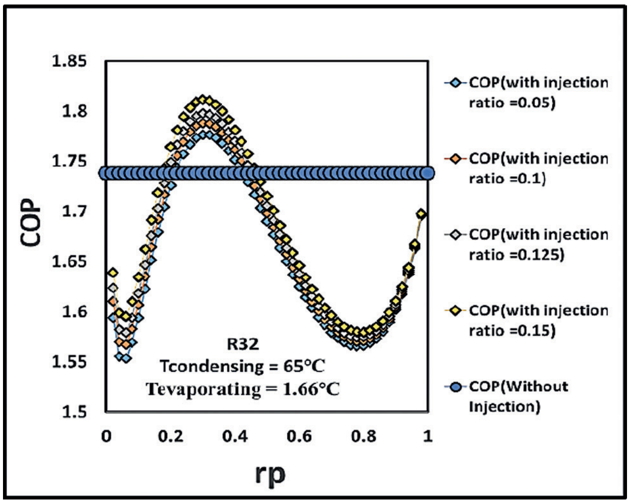

Figure 10. Variation of COP with respect to Pressure ratio.

0.05, 0.1, 0.125, 0.15 are 31.38°C, 19.96°C, 14.14°C and 8.33°C respectively as shown in Figure 9. The degree of superheat for mass injection ratio 0.15 is 8.33 which suggests that the maximum value of mass injection ratio should be less than 0.15 for the safer working of the compressor.

Case 3: T_{cond} = 65°C and T_{eva} = 1.66°C

Now the condenser temperature and evaporator temperatures are changed to 65°C and 1.66°C respectively (data for this condition is not available in Copeland webpage). To predict the COP, discharge temperature and degree of superheat, similar method can be used which is used to predict the performance for above 2 cases. The value of isentropic efficiency can be calculated by a MATLAB code from the polynomials given in Eq. 16 and Eq.17. So 50 values of COP are obtained by the code for 50 different intermediate pressure ratios. The maximum COP obtained for said values of m_r are 1.7764, 1.7877, 1.7977 and 1.8111 respectively (at intermediate pressure ratio of 0.30) as shown in figure 10. However, the COP of refrigeration system of same capacity but using a single stage compressor is 1.7379. Cheng *et al.*, analyzed non-azeotropic mixture R32/R1234ze (E) in a heat pump and predicted COP of ~3 for mass injection ratio of 0.15 for a vapor injection system [25]. Then for a refrigeration system the COP would be ~2 and our value is near to it.

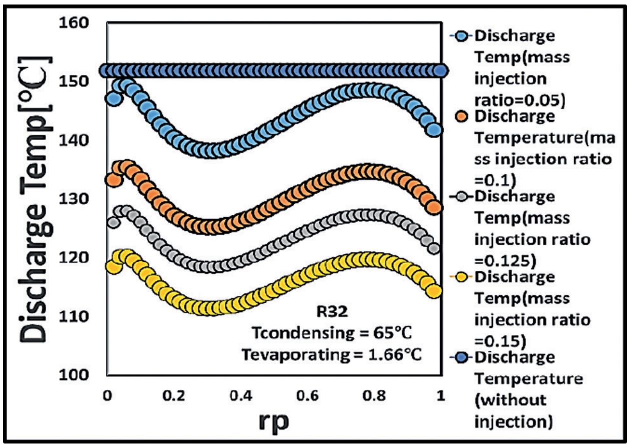

Figure 11. Variation of discharge Temp. with Intermediate pressure ratio.

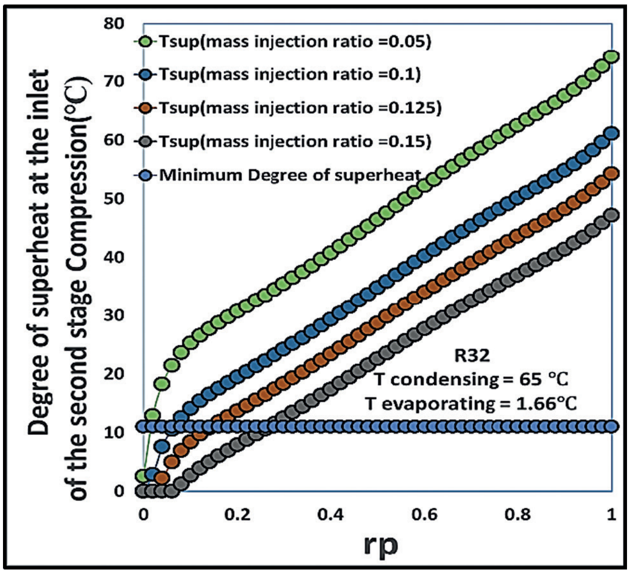

Figure 12. Variation of Degree of Superheat with respect to Intermediate Pressure Ratio.

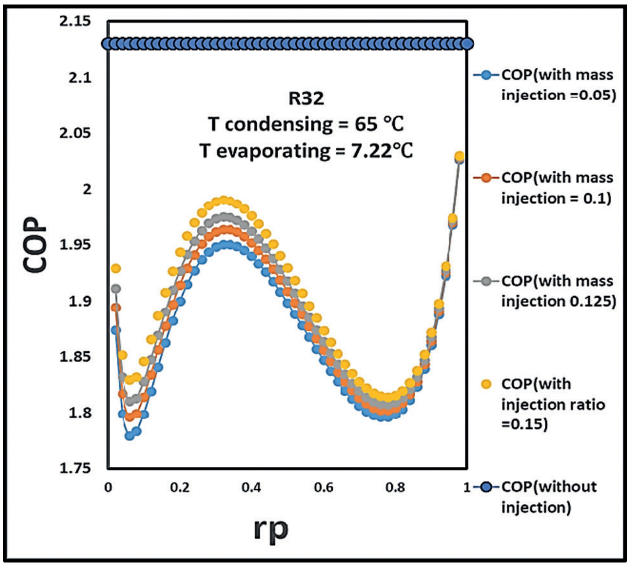

Figure 13. Variation of COP with respect to Pressure ratio.

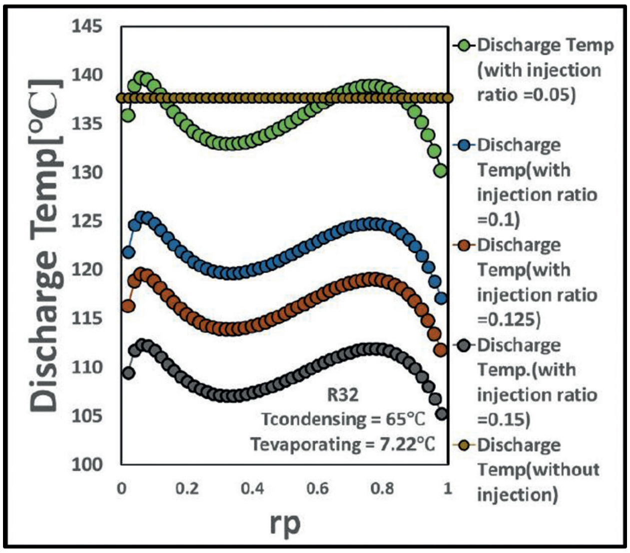

Figure 14. Variation of Discharge Temp with respect to Pr.

Figure 10 shows the variation in COP of the cycle with respect to the 50 values of intermediate pressure ratio ranging from 0.001 to 0.999 for mass injection ratio of 0.05, 0.1, 0125 and 0.15. The COP obtained at $r_p = 0.30$ and said values of m_r are 2.21%, 2.86%, 3.44%, 3.54%, 4.21% higher than COP of refrigeration system of same capacity but using a single stage compressor. Figure 11 shows the variation of compressor discharge temperature with respect to 50 values of intermediate pressure ranging from 0.001 to 0.999 for $m_r = 0.05$, 0.1, 0.125 and 0.15. The minimum discharge temperature obtained at the $r_p = 0.32$ and for said values of m_r are 138.18°C, 125.2°C, 118.35°C and 111.3°C respectively.

Figure 12 shows the variation in degree of superheat at the inlet of second stage of compression with respect to 50 values of intermediate pressure ranging from 0.001 to 0.999. The degree of superheat obtained for said values of m_r are 35.48°C, 24.20°C, 18.37°C and 12.49°C respectively.

The obtained value of degree of superheat is slightly higher than 11.11°C for mass injection ratio 0.15. The results shows that even in a very hot tropical climate when the condensing temperature becomes very high due to increase in the ambient temperature the two-phase injection system can helps prevent the compressor discharge temperature to not reach beyond 120°C.

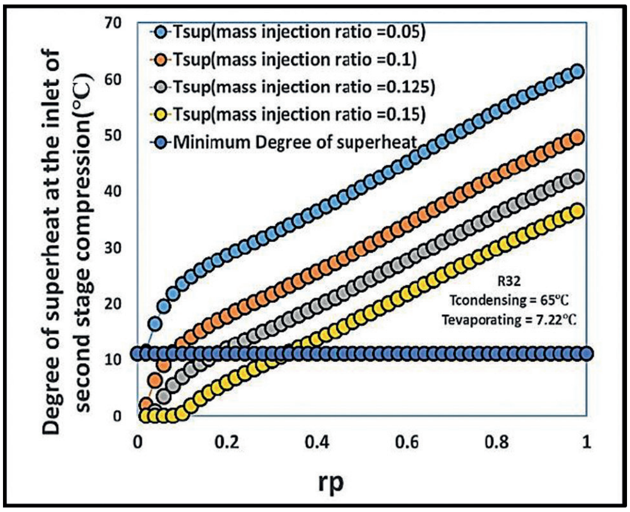

Figure 15. Variation of Degree of Superheat with respect to Pr.

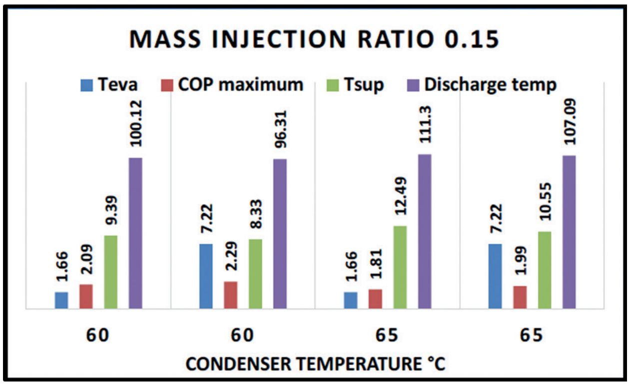

Figure 16. Varying values of COP, degree of superheat and discharge temperature with respect to condenser temperature.

Case 4: T_{cond} = 65°C and T_{eva} = 7.22°C

The evaporator temperature is considered to be 7.22°C and condensing temperature as 65°C, 50 iterations were performed. Figure 13 shows the variation of COP with respect to 50 values of intermediate pressure ratio ranging from 0.001 to 0.999 for m_r = 0.05, 0.1, 0125 and 0.15. The maximum COP obtained are 1.9506, 1.9638, 1.9751 and 1.9903 respectively at intermediate pressure ratio of 0.32. The COP of refrigeration cycle using single stage compressor working within same temperature limit is 2.1304 which is even higher than the reported COP by Yang *et al.* [19]. The COP obtained at r_p = 0.3 and said values of m_r are 8.43%, 7.82%, 7.28% and 6.57% lower than the COP of refrigeration system of identical capacity but using a single stage compressor.

Figure 14 shows the variation of compressor discharge temperature with respect to 50 values of intermediate pressure ranging from 0.001 to 0.999. For said values of m_r compressor discharge temperatures are 132.85°C, 119.64°C, 113.81°C and 107.09°C respectively. Figure 15 shows the variation of degree of superheat with respect to 50 values of intermediate pressure ranging from 0.001 to 0.999. For said values of m_r, the degree of superheat are 33.19°C, 22.41°C, 16.41°C and 10.55°C respectively.

In summary, the best mass injection ratio is 0.15 as per the simulation results of this work. Consolidated values of maximum COP, minimum degree of superheat and minimum discharge temperature for each cases are plotted together at mass injection ratio of 0.15 (Figure 16). Following the trend of graph in Figure 16 it is assuring that increasing evaporator temperature and lowering condenser temperature increases COP [26]. Moreover, low condenser temperature is also related to lesser degree of superheat and discharge temperature. It is evident that the maximum discharge temperature is less than 120°C even for extreme operating temperature, say, 65°C condenser temperature and 1.66°C evaporator temperature (figure 15). Thus the refrigerant R32 is safe at high atmospheric temperature. The degree of superheat at the inlet of second stage of compressor is also greater than 8°C which ensures no liquid refrigerant enters the scroll compressor. Xu *et al.* performed experiment on vapor injection system using R32 and R410a for various mass injection ratio, for extreme cooling at ambient temperature of 46°C [27]. However, condenser temperature will be ~15 to 20°C higher than the atmosphere temperature (in summer it can go up to 65°C) [28, 29]. Xu *et al.* reported ~15 to 20% decrease in cooling COP of the system when employing vapor injection system [27]. The maximum decrease in COP observed for two phase injection system was ~9%. The minimum discharge temperature obtained for R32 using vapor injection system was ~122°C for vapor injection ratio of 0.4. In case of two-phase injection system, the discharge temperature for mass injection ratio of 0.15 is ~110°C. Thus two-phase injection system is effective in decreasing the compressor discharge temperature. *Shuxe et al.* reported the experimental results for various ambient (condenser) temperature enhanced vapor injection system [30]. The enhanced vapor injection seems to increase the COP of the system depending on the vapor mass injection ratio. Between ambient temperatures 45°C and 40°C ~15% decrease in COP was observed. When compared to enhanced vapor injection system two phase injection system (proposed here) seems advantageous because the maximum decrease in COP for present system is ~9%. Guo *et al.* did experiment on R32 for air-conditioning and refrigeration application using the enhanced vapor injection approach [31]. The maximum decrease in discharge temperature reported is of 20-30°C. The maximum discharge temperature decrease obtained in this work is ~40°C for 65°C condenser temperature (1.66°C evaporator temperature). Therefore, the use of two-phase injection process can decrease the discharge temperature of R32 and may keep the COP of the refrigeration system competitive [32]. The optimum value of mass injection ratio depends on the operating conditions. The minimum degree of superheat required at the inlet of each stage of the compressor also affects the maximum value of mass injection ratio.

CONCLUSIONS

The modelling and analysis of refrigeration systems with two-phase injection is carried out using a MATLAB code. In order to establish the model, the operating conditions has been taken from a commercial scroll compressor data sheet. Hot atmospheric temperature has been considered for the COPELAND™ Scroll compressor for analyzing R32 performance. It is evident from the results that two-phase injection system at mass injection 0.15 decreases the discharge temperature and also ensures that no liquid refrigerant enter the scroll compressor. When compared to a conventional single stage system, a two phase system would operate at low discharge temperature (~120°C) without compromising the system coefficient of performance and energy efficiency ratio.

NOMENCLATURE

EER	Energy efficiency ratio
P_{inj}	Injection pressure, bar
η_{ise}	Isentropic efficiency, %
m	Mass flow rate, kg/s
T	Temperature, °C
Pr	Pressure ratio
r_p	Injection Pressure ratio
m_r	Mass injection ratio
T_{eva}	Evaporator temperature, °C
T_{sup}	Superheated temperature, °C
T_{sub}	Sub-cooling temperature, °C
T_{cond}	Condenser temperature, °C
m_{total}	Total mass flow rate lbm/hr
m_{inj}	Injection Mass flow rate lbm/hr
S	Entropy, kJ/kgK
Subscript:	
isen	Isentropic
inj	Injection

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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