Thermodynamic Analysis of a Combined Single Effect Vapour Absorption System and tc-CO$_2$ Compression Refrigeration System

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Abstract
Transcritical CO$_2$ refrigeration system is coupled with the single effect vapour absorption with LiBr-water as a working pair having an objective to enhance the performance of low temperature transcritical refrigeration system while using natural working pair and to reduce the electricity consumption to produce low temperature refrigeration. The high grade waste heat rejected in the gas cooler of tc-CO$_2$ compression refrigeration system (TCRS) is utilized to run the single effect vapour absorption system (SEVAR) to enhance the energy efficiency of the system. The gas cooler in the transcritical CO$_2$ system is having heat energy at high temperature and pressure, which is utilized to run the vapour absorption system, while the other refrigerant heat exchanger provides subcooling to further enhance the performance. The combined cycle can provide refrigeration temperature at different levels, to use it for different applications. Energetic and exergetic analysis have been done to analyze the combined system to compute the performance parameters and the irreversibilities occurring in different components to further increase the performance. The combined system is optimized for various heat rejection and refrigeration temperatures. The COP of the combined system has been enhanced by to 24.88% while the enhancement in exergetic efficiency ($\eta_{ex}$) is observed as 10.14% respectively over tradition transcritical CO$_2$ compression refrigeration system, with -10°C as an evaporation (TCRS cooling) temperature and exit temperature of gas cooler $T_4$ being 40°C.

Keywords: Exergy; Vapour Absorption; Carbon Dioxide; Waste Heat; Transcritical.

1. Introduction
Refrigeration and air conditioning plays vital role in almost every sector of the society. Global warming and depletion of ozone layer have become the key issue for the conventional refrigeration systems. In 1987, Montreal Protocol [1], gave the time limit for the usage of chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) refrigerants, as they are responsible for ozone depletion, but still the use of hydrofluorocarbons (HFC)s was being the major concern as its effects are hazardous for the environment and climate change. In 1997, Kyoto Protocol [2], limited the use of HFCs with large GWP. In 1998, Robinson and Groll [3], suggested the use of naturally occurring refrigerants which do have low GWP and environment friendly. CO$_2$ as a naturally occurring refrigerant having good thermophysical properties [4] finds favour across almost all sectors of refrigeration to be used as the refrigerant [5]. It is having critical temperature of 30.85°C [3]. Transcritical CO$_2$ (R744) has been adopted worldwide for supermarkekt, food storage and industrial appliactions etc., even to the locations having high ambient temperatures [6]. In CO$_2$ refrigeration system subcooling plays an important role in upgrading the performance of the system, dedicated sub cooling methods improve COP by 30%, thermoelectric systems by 25.6%, internal heat exchangers improves COP by 12 while 22% with economizers [7]. In CO$_2$ refrigeration system, Bellos and Tzivanidis [8], reported the performance of the system upto 75% by the use of mechanical subcooling system, over the basic configuration. Use of ejector and

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thermoelectic subcooling enhance the performance upto 40% [9]. Internal heat exchanger for subcooling improves the COP by 12 and 25% by using thermolectric subcooling for specific operating conditions [10] Hojjat Mohammadi [11] studied various configurations of the CO₂ refrigeration system coupled with different absorption chillers to produce refrigeration at different temperature levels from -80°C to -30°C with an increase of COP upto 200% in few cases. The mean COP improvement for CO₂ refrigeration system has been reported by 23% with subcooling by absorption chiller [12]. The net impact of GWP is 1 for CO₂ refrigeration systems [15]. Basso et al. [13], integrated the transcritical CO₂ heat pump to reduce the load of the external heat source, for hybrid system using dynamic simulation. The working pair used in the absorption refrigeration system is ecofriendly, non-flammable and non-toxic in nature, having low operating pressures [14]. Presently, the emphasis is on the energy efficient system to fulfill the demand of refrigeration and air conditioning (RAC) of the society. To improve the energy efficiency of the system, exergy analysis is the favorable tool to be used as it is defined as the potential of a stream to cause change as well as it tells the quality of the system, as an effective portion of the potential of the system to have an impact on environment [17]. It also quantifies the irreversibilities occurring in components of the system.

The reported literature review suggests that subcooling the tc-CO₂ system has been showing promising enhancement in the performance of the system. Thus to couple the absorption system with tc-CO₂ system improves the overall system performance and provide refrigeration with different temperature levels. The present study focus on the use of waste heat to run the single effect vapour absorption system (SEVAR). The waste heat being rejected in the gas cooler (GC) of tc-CO₂ compression refrigeration system (TCRS) [18] is being utilized to run SEVARS, to improve the energy efficiency of the combined system. The exergy analysis and parametric study of the combined system is being presented.

2. System Description

The SEVARS is coupled with TCRS. The superheated refrigerant (CO₂) is compressed in a compact compressor to a high temperature and pressure CO₂. The heat rejection of high pressure and temperature CO₂ occurred in gas cooler 1 (GC1) and gas cooler 2 (GC₂) is utilized by circulating the water. In GC₁, water at 1 atmospheric pressure and 100°C enters and changes its phase from liquid water to saturated steam at constant temperature of 100°C, as the temperature of CO₂ is well above 100°C. The generated saturated steam is utilized as the heat input to the SEVARS, where generator is kept at 90°C [10]. The temperature of the refrigerant (CO₂) is still higher (more than 100°C) after rejecting heat in GC₁, therefore it is required to further reject heat in GC₂ to an optimum gas cooler temperature of 40°C, by circulating the water at 25°C, 1 atm pressure. The hot water obtained from GC₂ can be utilized for various applications, such as desalination, distillation, industrial and domestic uses [17].

![Figure 1. TCRS coupled with SEVARS](image-url)
Further the refrigerant is passed through refrigerant heat exchanger (RHX\textsubscript{TC}), to improve the efficiency of TCRS and expanded through the refrigerant throttling valve RTV\textsubscript{TC}. The refrigerant CO\textsubscript{2} is expanded upto a low temperature of -10°C. The saturated refrigerant vapour at the exodus of evaporator gets superheated vapour while exchanging heat in RHX\textsubscript{TC} which reduces the compressor work considerably. The working of SEVAR Libr-H\textsubscript{2}O based system has reported by various authors [20-23].

3. Thermodynamic Analysis

3.1. System Model

The analysis of the combined cycle includes mass balance, concentration balance, energy balance and exergy balance for individual component and is presented as [16, 22]:

\[
\sum \dot{m}_i - \sum \dot{m}_e = 0 \quad (1)
\]

\[
\sum \dot{m}_i \dot{x}_i - \sum \dot{m}_e \dot{x}_e = 0 \quad (2)
\]

\[
\sum \dot{Q} - \sum \dot{W} = \sum \dot{m}_e \dot{h}_e - \sum \dot{m}_i \dot{h}_i \quad (3)
\]

Form 1\textsuperscript{st} law of thermodynamics (FLT), the (COP) of TCRS is given as:

\[
COP_{TC} = \frac{\dot{Q}_{Etc}}{\dot{W}_C} \quad (4)
\]

Form 1\textsuperscript{st} law of thermodynamics (FLT), the (COP) of VARS is given as:

\[
COP_{VA} = \frac{\dot{Q}_{Eva}}{\dot{Q}_G + \dot{W}_P} \quad (5)
\]

Form 1\textsuperscript{st} law of thermodynamics (FLT), the (COP) of combined is given as:

\[
COP_{NET} = \frac{\dot{Q}_{Etc} + \dot{Q}_{Eva}}{\dot{W}_C} \quad (6)
\]

Exergy flow rate for a stream on each state is defined as:

\[
\dot{E} = \dot{m}(h - h_0) - T_0(s - s_0) \quad (7)
\]

Considering a steady state process, the exergy destruction rate (\(\dot{E}D\)) to a component is specified as [16, 24]:

\[
\dot{E}D = \sum \dot{E}_i - \sum \dot{E}_e + \sum \dot{Q} \left(1 - \frac{T_0}{T}\right) - \sum \dot{W} \quad (8)
\]

Based on 2\textsuperscript{nd} law of thermodynamics (SLT), the performance parameter (exergetic efficiency) for the system given as [16, 24]:

\[
\eta_{exTC} = \frac{\dot{Q}_{Etc} \left|1 - \frac{T_0}{T_{TC}}\right|}{\dot{W}_C} \quad (9)
\]

\[
\eta_{exVA} = \frac{\dot{Q}_{Eva} \left|1 - \frac{T_0}{T_{VA}}\right|}{\dot{Q}_G \left|1 - \frac{T_0}{T_G}\right| + \dot{W}_P} \quad (10)
\]

\[
\eta_{exNET} = \frac{\left(\dot{Q}_{Etc} \left|1 - \frac{T_0}{T_{TC}}\right| + \dot{Q}_{Eva} \left|1 - \frac{T_0}{T_{VA}}\right|\right)}{\dot{W}_C + \dot{W}_P} \quad (11)
\]

3.2. Assumptions

The following have been made to analyze the combined system:

- Entirely, individual components of the combined system are considered as control volume.
- The combined system follows steady state conditions.
- The pressure drop in connecting lines and components is neglected.
- The temperature of the refrigerated space required is supposed to be 5°C higher to the respective evaporator temperature.

- The refrigerant (water) leaving condenser of SEVAR is considered to be saturated liquid.

- The exodus of evaporator is considered to be saturated vapour.

- Pumping in SEVAR is considered to be isentropic [19].

- Entropy change through the solution throttling valve (STV) is derelict and the temperature is expected to be constant [19].

- The SEVARS is well away from crystallization.

- Water at 25°C, 1 atm is used to cool the GC₂ TCRS and condenser & absorber of SEVARS.

3.3. Research Methodology

The research methodology has been explained in the flowchart for the combined analysis of the coupled cycle.

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**Figure 2. Flowchart for the analysis of the combined cycle**
3.4. Input parameters

- Isentropic efficiency of compressor is [3]:
  \[ \eta_c = 0.815 + 0.022r_p - 0.0041(r_p)^2 + 0.0001(r_p)^3 \]  
  \[ (12) \]

- Generator temperature, \( T_g \) = 90°C;
- Evaporator temperature in SEVARS, \( T_{eva} \) = 7°C;
- Mass flow rate of refrigerant (CO\(_2\)) in TCRS, \( m_{rc} \) = 1 kg/s;
- Effectiveness of gas cooler 1, \( (\epsilon_{gc1}) \) = 0.8;
- Effectiveness of gas cooler 2, \( (\epsilon_{gc2}) \) = 0.8;
- Effectiveness of solution heat exchanger (SHX), \( \epsilon_{shx} \) = 0.7;
- Condenser and absorber temperatures, \( T_{cond} = T_a = 35°C \).

4. Results and Discussion

4.1. Simulation Validation

The validation of the combined system, has been done by validating the two cycles separately as there is no literature available for this coupled cycle in this manner. The analysis of single effect VARS cycle is compared with the results presented by Kaushik and Arora [20]. The input parameters considered for the validation of cycle are: \( T_E = 7.2°C \), \( T_G = 87.8°C \), \( T_A = T_C = 37.8 °C \), solution heat exchanger effectiveness = 0.7, refrigerant mass flow rate = 1 kg/s. Table shows the comparison of the results obtained by the Kaushik and Arora [20] and the present study for energy transfer involved in various components and the COP of the system. It is seen that there is a good agreement among the results obtained in the present study and those available in the literature. Also, Figures 3 and 4 show that the variation of COP with generator temperature ant various absorber temperature shows similar trends for the present study and those reported in the literature. Thus, the present validation of the system is reliable.

- Table 1. Energy analysis comparison of present work with numerical values given in Kaushik & Arora [20] for SEVARS

| S. No. | Component                  | Kaushik & Arora [20] | Present study | Difference |
|-------|----------------------------|----------------------|--------------|------------|
| 1.    | Generator                  | 3095.7               | 3096         | -0.00969   |
| 2.    | Absorber                   | 2945.27              | 2946         | -0.02479   |
| 3.    | Condenser                  | 2505.91              | 2506         | -0.00359   |
| 4.    | Evaporator                 | 2355.45              | 2355         | 0.019105   |
| 5.    | Solution Heat Exchanger    | 518.72               | 519.5        | -0.15037   |
| 6.    | Solution throttle valve    | 0                    | 0            | -          |
| 7.    | Refrigerant throttle valve | 0                    | 0            | -          |
| 8.    | Pump                       | 0.0314               | 0.03093      | 1.496815   |
| 9.    | Energy Input               | 5451                 | 5451         | 0          |
| 10.   | COP (No Dimensions)        | 0.7609               | 0.7608       | -          |

- Figure 3. COP variation with generator temperature in single effect systems, (Kaushik and Arora [20])
- Figure 4. COP variation with generator temperature in single effect systems (Present study)
4.2. COP and Exergetic Efficiency

The trend of COP with gas cooler pressure $P_{GC}$ and exergetic efficiency ($\eta_{ex}$) with $P_{GC}$ is shown in Figure 5, having $GC_{2}$ outlet temperature to be 40°C. The COP and exergetic efficiency ($\eta_{ex}$) of the system increases as the $P_{GC}$ increases up to an optimum gas cooler pressure and then started decreasing gradually. At constant gas cooler outlet temperature, the $P_{GC}$ increases the refrigerating capacity and the compressor work, while initially the rate of increase in refrigerating effect is more than the exergetic efficiency ($\eta_{ex}$) and COP and increases up to an optimum $P_{GC}$ and further decreases gradually beyond this optimum $P_{GC}$ as shown in Table 2. For an evaporation (cooling) temperature of -10°C, in TCRS, outlet temperature of gas cooler $T_{4}$ being 40°C, the optimum $P_{GC}$ is found to be approx. 10 MPa. The thermodynamic state points, energy transfer, exergy destruction are computed in Tables 5-7. The performance parameters of the combined system is compared with the base case of TCRS and accessible in Table 6.

![Figure 5. Influence of $P_{GC}$ on COP and $\eta_{ex}$ of TCRS](image)

Table 2. The deviation of power input (W), refrigerating effect (Q_{Etc}), COP & exergetic efficiency ($\eta_{ex}$) of TCRS with pressure of gas cooler for ($T_{c} = 40^\circ C$) and ($T_{Etc} = -10^\circ C$)

| $P_{GC}$ (MPa) | W (kW) | $Q_{Etc}$ (kW) | COP_{TCRS} | $\eta_{ex_{TCRS}}$ |
|---------------|--------|---------------|------------|-------------------|
| 8.0           | 72.72  | 81.46         | 1.12       | 12.53             |
| 8.2           | 74.56  | 89.89         | 1.206      | 13.49             |
| 8.4           | 76.36  | 99.97         | 1.309      | 14.65             |
| 8.6           | 78.14  | 112.2         | 1.436      | 16.07             |
| 8.8           | 79.89  | 126.7         | 1.586      | 17.74             |
| 9.0           | 81.61  | 140.6         | 1.723      | 19.27             |
| 9.2           | 83.3   | 151.1         | 1.814      | 20.3              |
| 9.4           | 84.98  | 158.5         | 1.866      | 20.87             |
| 9.6           | 86.63  | 163.9         | 1.892      | 21.17             |
| 9.8           | 88.26  | 168           | 1.903      | 21.3              |
| 10.0          | 89.87  | 171.3         | 1.906      | 21.33             |
| 10.2          | 91.46  | 174.1         | 1.903      | 21.29             |
| 10.4          | 93.04  | 176.4         | 1.896      | 21.22             |
| 10.6          | 94.59  | 178.5         | 1.887      | 21.11             |
| 10.8          | 96.14  | 180.3         | 1.876      | 20.99             |
| 11.0          | 97.66  | 182           | 1.863      | 20.85             |
| 11.2          | 99.18  | 183.5         | 1.85       | 20.7              |
| 11.4          | 100.7  | 184.8         | 1.836      | 20.54             |
| 11.6          | 102.2  | 186.1         | 1.821      | 20.38             |
| 11.8          | 103.6  | 187.2         | 1.807      | 20.21             |
| 12.0          | 105.1  | 188.3         | 1.792      | 20.04             |
Table 3. State points obtained by thermodynamic analysis

| State | T (°C) | s (kJ/kgK) | h (kJ/kg) | x   | m (kg/s) | P (kPa) |
|-------|--------|------------|-----------|-----|----------|---------|
| 1     | 30     | -0.6658    | -22.42    | -   | 1        | 2649    |
| 2     | 151.9  | -0.6325    | 67.48     | -   | 1        | 10004   |
| 3     | 110.4  | -0.7685    | 12.61     | -   | 1        | 10004   |
| 4     | 40     | -1.383     | -193.8    | -   | 1        | 10004   |
| 5     | 27.51  | -1.543     | -243      | -   | 1        | 2649    |
| 6     | -10    | -1.492     | -243      | -   | 1        | 2649    |
| 7     | -10    | -0.8405    | -71.64    | -   | 1        | 2649    |
| 8     | 25     | 0.3669     | 104.8     | -   | 0.7217   | 101.3   |
| 9     | 93.3   | 1.231      | 390.8     | -   | 0.7217   | 101.3   |
| 10    | 100    | 1.307      | 419.1     | -   | 0.02432  | 101.3   |
| 11    | 100    | 7.354      | 2676      | -   | 0.02432  | 101.3   |
| 12    | 35     | 0.2184     | 81.15     | 0.5408 | 0.1091  | 1.002   |
| 13    | 35     | 0.2184     | 81.16     | 0.5408 | 0.1091  | 5.627   |
| 14    | 62.29  | 0.3945     | 137.7     | 0.5408 | 0.1091  | 5.627   |
| 15    | 90     | 0.4751     | 239.6     | 0.6477 | 0.09108 | 5.627   |
| 16    | 51.93  | 0.2775     | 171.9     | 0.6477 | 0.09108 | 5.627   |
| 17    | 51.93  | 0.2775     | 171.9     | 0.6477 | 0.09108 | 1.002   |
| 18    | 90     | 8.662      | 2669      | 0    | 0.01802  | 5.627   |
| 19    | 35     | 0.505      | 146.6     | 0    | 0.01802  | 5.627   |
| 20    | 7      | 0.5246     | 146.6     | 0    | 0.01802  | 1.002   |
| 21    | 7      | 8.973      | 2513      | 0    | 0.01802  | 1.002   |
| 22    | 25     | 0.3669     | 104.8     | -   | 1        | 101.3   |
| 23    | 37.45  | 0.5381     | 156.9     | -   | 1        | 101.3   |
| 24    | 25     | 0.3669     | 104.8     | -   | 1        | 101.3   |
| 25    | 35.86  | 0.5166     | 150.3     | -   | 1        | 101.3   |

Table 4. Energy transfer in various components

| Component            | Q (kW) | W (kW) |
|----------------------|--------|--------|
| **TCRS**             |        |        |
| Evaporator           | 171.4  | -      |
| Compressor           | -      | 134.7  |
| **Gas Cooler (GC₁)** | 54.87  | -      |
| **Gas Cooler (GC₂)** | 206.4  | -      |
| **RHX**              | 54.9   | -      |
| **VARS**             |        |        |
| Evaporator           | 42.64  | -      |
| Condenser            | 45.43  | -      |
| Absorber             | 52.08  | -      |
| Generator            | 54.87  | -      |
Table 5. Exergy destructed rate in various components

| Component             | Exergy destructed (kW) |
|-----------------------|-------------------------|
| TCRS             |                         |
| Compressor          | 9.906                   |
| Evaporator          | 3.621                   |
| Gas Cooler (GC₁)    | 3.293                   |
| Gas Cooler (GC₂)    | 2.65                    |
| RHX                 | 4.38                    |
| RTV₉₂               | 15.28                   |
| Total₉₂₃            | 39.13                   |
| VARS             |                         |
| Evaporator          | 0.80                    |
| Condenser           | 0.81                    |
| Absorber            | 2.39                    |
| Generator           | 2.75                    |
| SHX                 | 0.36                    |
| STV                 | 0.01                    |
| RTV₉₄               | 0.12                    |
| Total₉₄₃           | 7.219                   |

Net Exergy Destruction 46.35

Table 6. Performance parameters (COP and ηex) comparison with the base TCRS

|                  | TCRS | SEVARS | Combined system | % increase |
|------------------|------|--------|-----------------|------------|
| COP              | 1.91 | 0.78   | 2.38            | 24.88      |
| ηex (%)          | 21.33| 17.62  | 23.49           | 10.14      |

4.3. Exergy Destruction

Figure 6 presents the exergy destruction in various components of TCRS & SEVARS. It is found that the exergy destruction in SEVAR is maximum in generator followed by absorber and condenser, while the exergy destruction in TCRS is maximum in RTV₉₂ followed by compressor, RHX, evaporator, GC₁ and GC₂ respectively. The exergy destruction of the components implies us to use the components with higher energy efficiency. Therefore to improve the performance of SEVARS, the design of generator & absorber should be focused. In TCRS, the energy can be recovered by replacing the throttling valve with other expansion devices such as expander, ejector, work recovery turbine etc. to further progress the performance of the combined system.

Figure 6. Exergy destructed in various components
4.4. Gas Cooler Pressure

Figure 7 presents the variation of $P_{GC}$ with TCRS evaporation temperature $T_{ETC}$ with different outlet temperature of gas cooler. The $P_{GC}$ is an important parameter in tc- CO$_2$ compression refrigeration system. The $P_{GC}$ depends upon the exit temperature of gas cooler and the temperature of evaporator [17]. The $P_{GC}$ declines as the evaporator temperature increases, while it increases as the outlet temperature of gas cooler ($T_i$) increases. The gas cooler pressure is the deciding factor in sizing the components of the transcritical refrigeration system and has an influence on the COP of the system. To maximize the performance parameters of TCRS, the optimum $P_{GC}$ is approximated to be 10 MPa for an evaporator temperature at -10°C and outlet temperature of gas cooler ($T_i$) being 40°C.

![Figure 7. Influence of evaporator temperature of TCRS on $P_{GC}$ at different $T_i$.]

4.5. COP vs Evaporator Temperature

Figure 8 presents the trend of COP$_{TCRS}$ and COP$_{NET}$ (combined system) with temperature of evaporator. Both the COP trends increases with the increase in evaporating temperature. This is due to the fact, as the temperature of evaporator increases, the work of compressor decreases and hence the COP. The utilization of waste heat from the gas cooler has increased the cooling capacity of TCRS, also provides additional cooling capacity is observed through SEVARS.

![Figure 8. Influence of evaporator temperature of TCRS on COP at different $T_i$.]
4.6. Exergetic Efficiency ($\eta_{ex}$) vs. Evaporating Temperature

Figure 9 presents the trends of exergetic efficiency ($\eta_{ex}$) of TCRS and the combined system with the TCRS evaporating temperature with changed gas cooler outlet temperature. It is observed that the refrigerating temperature has a dominant effect on the exergetic efficiency of the combined system. Both the exergetic efficiencies get the optimum peak and decreases gradually with the increase in evaporating temperature, as the maximum work potential to brought to the system to the environmental conditions decreases as the temperature of evaporator increases and hence the exergetic efficiency.

![Figure 9. Influence of evaporator temperature of TCRS on ($\eta_{ex}$) at different $T_i$.](image)

5. Conclusions

The subsequent conclusions have been made from the results:

- Coupling the two cycles increases the COP of the system interestingly. The COP of the combined system increases by 24.88% while the exergetic efficiency is increased by 10.14% over the modified TCRS having RHX$_{TC}$.
- There occurs an optimum gas cooler pressure for an evaporator temperature and the gas cooler outlet temperature $T_i$ for TCRS. At an evaporation temperature of -10°C and $T_i$ being 40°C, $P_{GC}$ is found to be 10 MPa.
- The COP of the system increase as the evaporation temperature increases, on the contrary it decreases it decreases with the rise in outlet temperature of gas cooler.
- The exergetic efficiency shows a peak with the evaporation temperature, which further decreases with the increase in the evaporation temperature however it increases with $T_i$.
- Efficient compressor and heat exchangers would result in increase in performance of the combined system.
- The exergy destructed in RTV$_{TC}$ is considerably high, therefore replacement of throttling (expansion) valve by an expander, ejector, and work-recovery turbine will contribute to increase in performance of the system.

6. Declarations

6.1. Author Contributions

A.V.: Conceptualization, Writing - Original Draft preparation, Formal analysis, Methodology; S.K.T.: Formal analysis, Writing- Reviewing and Editing, Supervision; S.C.K.: Conceptualization, Reviewing and Editing, Supervision.

6.2. Data Availability Statement

The data presented in this study are available in article.
6.3. Funding and Acknowledgements

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6.4. Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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