Analysis of the thermodynamic disequilibrium loss of the two-phase expansion process and its influence on the performance of a reciprocating expander

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Abstract. The screw expander and the reciprocating expander were considered as the suitable technologies to be employed as two-phase expanders of the Trilateral Flash Cycle (TFC) systems. The two-phase expansion process in the screw expander could be approximately viewed as the thermodynamic equilibrium process due to the strong disturbing and mixing between the two-phase working fluids in the chamber volume. However, for the reciprocating expander, the situation was different. The flashed vapor in the reciprocating expander cylinder was separated from liquid in the flash chamber, which would result in a significant thermodynamic disequilibrium loss. To assess this loss, the thermodynamic equilibrium model (TEM) and the flash vaporization model (FVM) were simultaneously proposed in this paper. Water was used as the two-phase working fluid. The two-phase expansion processes, the isentropic efficiencies and the output expansion works of the two models were obtained. Predicted results were analyzed and compared under different rotation speeds and injection temperatures. The results showed that when rotation speed of the expander was increased from 50 r/min to 1000 r/min, the thermodynamic disequilibrium loss was approximately increased from 5.3% to 6.3%, and when injection temperature was decreased from 523.15 K to 473.15 K, the thermodynamic disequilibrium loss was approximately increased from 4.5% to 6.2%.

1. Introduction

Converting the heat energy into mechanical energy or electrical energy through several thermodynamic cycles is an effective method to recover the waste thermal energy. There are two conventional thermodynamic cycles used for recovery of the moderate-to-low grade waste thermal energy: the Organic Rankine Cycle (ORC) and the Kalina Cycle (KC).

In recent years, the Trilateral Flash Cycle (TFC) has been widely investigated due to its higher exergy recovery efficiency [1,2]. Fischer [1] made a comparison between the optimized TFC systems using water as working fluid and the optimized ORC systems with organic working fluids. The results showed that exergy efficiency of the TFC was larger by 14%-29% than that of the ORC. Yari et al. [2] analyzed the performance of the TFC and compared it with that of the ORC and the KC. It was concluded that compared with the ORC system and the KC system, the net power output of the TFC system was larger.

Although the exergy efficiency and the performance of TFC system were theoretically higher than that of the ORC system or the KC system, the TFC system was still in a state of technical development. The reasons might be related to the following design constraints and operation limitations: (i) design
and manufacturing difficulties of the efficient and reliable two-phase expanders; (ii) selection of the suitable working fluids; (iii) constraint of the capital and operational cost. The screw expander [3-7] and the reciprocating expander [8-11] were considered as the suitable technologies to be employed as two-phase expanders of the TFC systems. And their efficiencies were affected by the two-phase expansion process (i.e. flash vaporization process). However, the two-phase expansion process is so complicated that its mechanism has not been clarified so far.

Thankfully, some experiments and numerical simulations were conducted to analyze the two-phase expansion process and improve isentropic efficiency of the expander. An analytical and experimental investigation of two-phase flow screw expanders for power generation was performed by Taniguchi et al. [3], and the effects of non-equilibrium flow in screw expanders were first considered by them. Some researchers assumed a uniform temperature distribution existing between the liquid phase and the vapor phase, and proposed the thermodynamic equilibrium model (TEM) to investigate the two-phase expansion process in the screw expanders [4-6]. For instance, Bianchi et al. [4] proposed a TEM to analyze the two-phase expansion process and calculate the volumetric and adiabatic efficiencies of the twin-screw expander. Pressure, temperature and quality evolutions of the two-phase expansion process were obtained, and parametric analysis including inlet pressure, inlet quality and rotation speed was carried out to assess their effects on the expander performance. Vasuthevan et al. [5] also proposed a TEM of the screw expander with injection and flash vaporization process. And a theoretical investigation of flash vaporization in a screw expander was further conducted by them [6]. It was shown that the pressure evolutions obtained by the FVM (flash vaporization model) and the TEM were almost the same.

For the two-phase reciprocating expander, the situation was different. Kanno et al. [8-10] carried out a series of experiments of the two-phase expansion in a cylinder with moving piston. A significant temperature difference between liquid and vapor was measured [8], and a large thermodynamic disequilibrium loss of the two-phase expansion process was presented [9]. Experimental results showed that the insufficient heat transfer between the liquid bulk and the vapor-liquid interface resulted in an uneven temperature distribution in the liquid phase inside the cylinder [10]. To analyze the insufficient heat transfer, an experimental correlation was finally developed, and was verified to be able to predict adiabatic efficiency of the expansion process within about 5% accuracy [10].

Based on these findings, it could be concluded that the TEM was suitable for the two-phase expansion process in the screw expander, while the FVM was fit for that in the reciprocating expander. The reasons might be that the two-phase working fluids in the screw expander were strongly disturbed and mixed, but in the reciprocating expander, the flashed vapor was separated from the liquid. Insufficient heat exchange between the two-phase working fluids in the reciprocating expander cylinder resulted in a large thermodynamic disequilibrium loss that could not be ignored.

However, rare literature could be found reporting this disequilibrium loss of the two-phase expansion process, particularly, there was lack of information about the effect of this irreversible loss on the expander performance. An in-depth analysis of the thermodynamic disequilibrium loss was important for developing the TFC systems with two-phase reciprocating expanders.

2. Description of the conventional TFC and expander concept design

2.1. Description of the conventional TFC

The schematic cycle configuration and the corresponding T-ς diagram of the conventional TFC are shown in figure 1. As is shown in figure 1, the conventional TFC employs the same components as the basic ORC, but unlike the basic ORC, the subcooled liquid working fluid in the heater is heated up to the saturation temperature corresponding to its pressure without undergoing a phase change process. Therefore, a better temperature matching between the heat source and the working fluid is achieved. In addition, unlike the superheated vapor expansion of the basic ORC, the saturated liquid in the conventional TFC is continuously flashed and generates the vapor during the expansion process, which is the so-called flash vaporization expansion.
Therefore, isentropic efficiency of the two-phase expansion process can be described as

\[
\eta_{ise} = \frac{\Delta h_{act}}{\Delta h_{ide}} = \frac{h_3 - h_4}{h_3' - h_4'}
\]

where \(\Delta h_{act}\) and \(\Delta h_{ide}\) are referred to as the actual specific enthalpy drop and the ideal specific enthalpy drop, respectively.

2.2. Expander concept design

A schematic of the concept design is shown in Figure 2. In particular, the piston is moved from the top dead center (TDC) to the bottom dead center (BDC) during the two-phase expansion process. The interspace between the bottom plate of cylinder and the TDC is the flash chamber which is considered as the intake and discharge cavities that could avoid the hydraulic shocks in the cylinder [11]. However, the two-phase expansion in the flash chamber would not create the effective expansion work. Therefore, the flash chamber resulted in the dead volume. In addition, it was believed that after the exhaust process, few working fluids would be left in the flash chamber as the residual mass appearing in the next cycle [11].
The residual mass \( m_{res} \) and the dead volume \( V_d \) could be given by equation (2) and equation (3), respectively.

\[
m_{res} = a \cdot m
\]  
\[
V_d = f \cdot \frac{m_{res} + m}{\rho(T_{inj})}
\]

where \( a \) \((0 < a < 1)\) and \( f \) \((f \geq 1)\) are the adjustable coefficients, \( m \) is the injection mass of an expansion process and \( \rho(T_{inj}) \) is the liquid density corresponding to the injection temperature \( T_{inj} \).

Considering that the size of a practical two-phase reciprocating expander is unknown, selection of the main expander parameters used in the present simulation is based on the Reciprocating Compressor Design Manual (RCDM) and the published literature, as listed in table 1.

| Parameter (Unit) | Value | Reference |
|------------------|-------|-----------|
| Piston Diameter \( d_{p_{is}} \) (m) | 0.25 | the RCDM |
| Crank Radius \( r_c \) (m) | 0.05 | the RCDM |
| Link Length \( l_{link} \) (m) | 0.225 | the RCDM |
| Piston Stroke \( l_{str}, l_{str} = 2r_c \) (m) | 0.1 | the RCDM |
| Stroke Volume \( V_{str} \) (m³) | 4.9×10⁻³ | the RCDM |
| Coefficient \( a \) (-) | 0.05 | [11] |
| Coefficient \( f \) (-) | 3 | [11] |

3. Thermodynamic modelling and solution procedure

3.1. Thermodynamic modelling

3.1.1. Conservation of mass. During the two-phase expansion process, the mass conservation equations can be expressed as:

\[
\frac{dm_l}{dt} = \frac{dm_{inj}}{dt} - \frac{dm_{fla}}{dt}
\]

\[
\frac{dm_v}{dt} = \frac{dm_{fla}}{dt}
\]

where \( m_{inj} \) and \( m_{fla} \) are the injected mass and the flashed mass, respectively.

3.1.2. Conservation of energy. The energy change in the liquid phase depends on the energy of the injected liquid, the energy of the flashed vapor and the heat loss of the liquid phase. The energy change in the vapor phase depends on the energy of the flashed vapor, the expansion work and the heat loss of the vapor phase. By neglecting variations in the kinetic and potential energies of working fluid \([5,8,10]\), the energy balance equations for the liquid and the vapor in the cylinder can be expressed as:

\[
\frac{d(m_lu_l)}{dt} = \frac{dm_{inj}}{dt} h_{inj} - \frac{dm_{fla}}{dt} h_{fla} - \frac{dQ_l}{dt}
\]

\[
\frac{d(m_vu_v)}{dt} = \frac{dm_{fla}}{dt} h_{fla} - p_v \frac{dV_{cyl}}{dt} - \frac{dQ_v}{dt}
\]
where $u$ is the specific internal energy, $p_v$ is the vapor pressure and $Q$ is the heat loss term which is caused by the heat exchange between the two-phase working fluids and the cylinder wall. In the actual operation, in order to reduce the heat loss of the working fluids as much as possible, it was necessary to implement an insulation device (e.g. insulation materials, vacuum insulation etc.) out of the cylinder block, especially when the rotation speed of the expander was low. Kanno et al. [8,9] conducted the calibration experiments to assess the performance of vacuum insulation, and the obtained experimental measurement results were adopted to calculate the heat loss term $Q$ in the present study.

3.1.3. The output expansion work. During the two-phase expansion process, the output expansion works of the two models could be expressed as follows:

$$dW_{\exp} = (p_v - p_a)dV_{\text{cyl}} \quad (8)$$

$$W_{\exp} = \int dW_{\exp} \quad (9)$$

where $p_a$ is the back pressure of the expander which was set as the atmospheric pressure in the present study. It should be noted that the present study only takes into account the expansion work output during the two-phase expansion process. For the exhaust process, it is assumed that there is almost no expansion work outputted or consumed since a proper injection mass, $m$, is selected to ensure that the exhaust pressure of the working fluid is close to the condensation pressure (i.e. the back pressure).

3.1.4. Solution equations. According to the analysis of Kanno et al. [10], the heat transfer between the liquid bulk and the vapor-liquid interface was considered as the predominant mechanism of the two-phase flash vaporization expansion. Based on the conservation of energy, latent heat of vaporization should come from the heat transfer from the liquid bulk to the vapor-liquid interface, as shown in equation (10).

$$\frac{dm}{dt} = L = \frac{dm_{\text{fl}}}{dt} = L = A_{\text{int}} \alpha_{\text{int}} (T_i - T_{\text{int}}) = A_{\text{pis}} \alpha_{\text{int}} (T_i - T_v) \quad (10)$$

where $L$ is the latent heat of vaporization, the vapor-liquid interface area $A_{\text{int}}$ is replaced by the piston area $A_{\text{pis}}$, and the vapor temperature $T_v$ is viewed as the interface temperature $T_{\text{int}}$, as suggested by Kanno et al. [10]. $\alpha_{\text{int}}$ is the heat transfer coefficient at the interface and can be calculated by employing the obtained experimental correlation of the Nusselt number [10].

The TEM was proposed to analyse the thermodynamic disequilibrium loss of the two-phase expansion process. According to the thermal equilibrium assumption, a sufficient heat exchange exists between the two-phase working fluids, which implies that the temperatures of the liquid and the vapor are consistent, as shown in equation (11).

$$T_i = T_v \quad (11)$$

It should be noted that the thermophysical properties appearing in the above equations could be retrieved from the NIST REFPROP database.

3.2. Solution procedure MATLAB&SIMULINK software associated with the above-mentioned NIST REFPROP database was used as the simulation tool, and the time iteration method was adopted in the present study.

When all the known structural and operating parameters are imported into the edited program, the time step $dt$ needs to be set. For the above FVM, there is one level of iterative calculation that needs to be completed. When the calculation of one time step is completed, the latest cylinder volume could be obtained according to the well-known equation of piston motion. Comparing the calculated cylinder
volume with the maximum cylinder volume (i.e. the stroke volume $V_{str}$), if the former is smaller than the latter, the iterative calculation continues. On the contrary, the iterative calculation is completed and the calculation results are exported from the program.

However, for the proposed TEM in this paper, there is another level of iterative calculation that needs to be finished. Under the given time step, guessing a flashed mass $dm_{fla}$, temperatures of the liquid and vapor could be obtained based on the previous conservation equations of mass and energy. Comparing the liquid temperature with the vapor temperature, if the difference between the two temperatures exceeds the preset error tolerance range, the guessed flashed mass needs to be adjusted to recalculate the liquid and vapor temperatures until the difference meets the permission condition. Step by step, the two levels of iterative calculation are finally completed.

The flowcharts of the proposed solution procedures of the FVM and the TEM are shown in figure 3.

![Flowcharts](image)

**Figure 3.** Flowcharts of the proposed solution procedures of the FVM and the TEM.

### 4. Results and discussion

#### 4.1. Temperature evolution and pressure evolution of the two-phase expansion process

Table 2 shows the main operating parameters used in the simulation. The injection mass, $m$, should be determined based on the operating temperature range, the stroke volume of the expander and the working fluids [11]. The proper injection mass under the present operating conditions is found to be 0.017 kg in order to make sure that the exhaust pressure of the working fluid is close to the condensation pressure (i.e. the back pressure). Temperature evolution curves and pressure evolution curves of the two-phase expansion process are shown in figure 4.
In figure 4(a), $\theta$ is the crank corner, and when the corner $\theta$ is increased from 0° to 180°, the two-phase expansion process is finished. $T_{eq}$ is the working fluids temperature of the TEM, $T_{l,\text{neq}}$ and $T_{v,\text{neq}}$ represent the liquid temperature and the vapor temperature of the FVM, respectively. It can be seen from figure 4(a) that during the two-phase expansion process, the temperature evolution processes of the proposed two models are almost the same, and all of them are rapidly increased to the maximums first of all and then are continuously decreased until the expansion process is completed. Meanwhile, it can also be seen that the insufficient heat transfer considered by the FVM mainly appears at the beginning of the expansion process where the difference between $T_{l,\text{neq}}$ and $T_{v,\text{neq}}$ is significant. And at the middle and late stages of the expansion process, the predicted temperatures of the two models are basically the same. It was considered that this result was related to the continuous injection of the saturated liquid. In addition, because the sufficient heat exchange considered by the TEM will decrease the liquid temperature $T_{l,\text{neq}}$ and increase the vapor temperature $T_{v,\text{neq}}$, the simulated temperature $T_{eq}$ appears between $T_{l,\text{neq}}$ and $T_{v,\text{neq}}$, which is in accordance with the law of conservation of energy.

| Parameter (Unit)                        | Value     |
|----------------------------------------|-----------|
| Working Fluid (-)                      | Water     |
| Injection Temperature $T_3$ or $T_{\text{inj}}$ (K) | 473.15    |
| Condensing Temperature $T_1$ or $T_{\text{con}}$ (K) | 373.15    |
| Back Pressure $p_a$ (kPa)              | 101.325   |
| Rotation Speed $n$ (r/min)             | 500       |
| Injection Mass of an Expansion Process $m$ (kg) | 0.017     |

**Table 2.** Main operating parameters used in the simulation.

![Figure 4](image.png)

**Figure 4.** (a) Temperature evolution curves and (b) Pressure evolution curves of the expansion process.

Similarly, both the pressure evolution processes of the two models are also rapidly increased to the maximums first of all and then are continuously decreased until the expansion process is completed, as shown in figure 4(b). Meanwhile, the difference between the pressures of the FVM and the TEM also appears at the beginning of the two-phase expansion process. Due to the thermodynamic disequilibrium loss existing in the FVM, the simulated pressure of the TEM is higher than that of the FVM, and the area between the pressure evolution curves of the two models represents this irreversible loss.

The predicted exhaust parameters and calculated results of the two models are shown in table 3. It can be seen from table 3 that both the isentropic efficiency and the output expansion work of the TEM are larger than that of the FVM, and the thermodynamic disequilibrium loss (i.e. $\eta_{\text{ISE,eq}} - \eta_{\text{ISE,neq}}$) under this operating condition is about 6.19%.
### Table 3. Predicted exhaust parameters and calculated results of the two models.

| Parameter (Unit)                  | Value (the FVM) | Value (the TEM) |
|-----------------------------------|-----------------|-----------------|
| Exhaust Temperature of Liquid $T_{4,l}$ (K) | 373.89          | 373.41          |
| Exhaust Temperature of Vapor $T_{4,v}$ (K)   | 373.35          | 373.41          |
| Exhaust Pressure $p_{4}$ (kPa)       | 102.160         | 101.768         |
| Isentropic Efficiency $\eta_{ise}$ (%)   | 76.30           | 82.49           |
| Output Expansion Work $W_{exp}$ (J)    | 634.8           | 680.6           |

#### 4.2. Influence of the rotation speed of the expander

To have a more comprehensive analysis of the thermodynamic disequilibrium loss, the influence of the rotation speed on the expander performance was discussed in this section.

The rotation speed of the expander is increased from 50 r/min to 1000 r/min and other parameters remain unchanged, the isentropic efficiencies and the output expansion works of the two models are calculated, as shown in table 4. As is shown in table 4, both the isentropic efficiencies and the output expansion works of the two models are decreased with the increase in the rotation speed of the expander. At the same time, the isentropic efficiency and the expansion work of the TEM are always significantly larger than that of the FVM.

### Table 4. The calculated results of the TEM and the FVM under different rotation speeds.

| Rotation Speed (r/min) | FVM         | TEM         | $\eta_{ise,eq}$ (%) | $W_{exp,eq}$ (J) | $\eta_{ise,eq}$ (%) | $W_{exp,eq}$ (J) | $\frac{W_{exp,eq} - W_{exp,neq}}{W_{exp,neq}} \times 100$ (%) |
|------------------------|-------------|-------------|---------------------|-----------------|---------------------|-----------------|---------------------------------------------------------------|
| 50                     | 81.88       | 687.7       | 87.15               | 728.9           | 5.27                | 5.99            |                                                               |
| 100                    | 81.60       | 686.7       | 87.03               | 728.7           | 5.43                | 6.11            |                                                               |
| 200                    | 80.09       | 672.9       | 85.76               | 716.9           | 5.67                | 6.53            |                                                               |
| 300                    | 78.85       | 661.5       | 84.76               | 706.6           | 5.91                | 6.81            |                                                               |
| 400                    | 77.57       | 648.8       | 83.65               | 694.5           | 6.08                | 7.04            |                                                               |
| 500                    | 76.30       | 634.8       | 82.49               | 680.6           | 6.19                | 7.22            |                                                               |
| 600                    | 75.01       | 619.6       | 81.29               | 665.2           | 6.28                | 7.37            |                                                               |
| 700                    | 73.72       | 603.2       | 80.04               | 648.3           | 6.32                | 7.49            |                                                               |
| 800                    | 72.40       | 585.7       | 78.74               | 630.2           | 6.34                | 7.61            |                                                               |
| 900                    | 71.06       | 567.4       | 77.36               | 611.1           | 6.30                | 7.71            |                                                               |
| 1000                   | 69.68       | 548.4       | 75.85               | 590.9           | 6.17                | 7.74            |                                                               |

In addition, it can be seen from table 4 that when the rotation speed of the expander is increased from 50 r/min to 1000 r/min, isentropic efficiency ($\eta_{ise,eq}$) of the FVM is approximately decreased from 81.88% to 69.68%, isentropic efficiency ($\eta_{ise,eq}$) of the TEM is approximately decreased from 87.15% to 75.85%, output expansion work ($W_{exp,eq}$) of the FVM is approximately decreased from 687.7 J to 548.4 J, and output expansion work ($W_{exp,eq}$) of the TEM is approximately decreased from 728.9 J to 590.9 J. Meanwhile, it is found that isentropic efficiency of the TEM is approximately larger by 5.3%-6.3% than that of the FVM, and output expansion work of the TEM is approximately larger by 6.0%-7.7% than that of the FVM. It was concluded that the effect of the thermodynamic disequilibrium loss on the performance of the two-phase reciprocating expander would be enhanced with the increase of the rotation speed.

#### 4.3. Influence of the injection temperature of the saturated liquid

The effects of the injection temperature of the saturated liquid ($T_{3}$ or $T_{in}$) on the thermodynamic disequilibrium loss and the expander performance were also analyzed in the present study.

The injection temperature is increased from 473.15 K to 523.15 K and other parameters remain unchanged, the isentropic efficiencies and the output expansion works of the two models are obtained,
as shown in table 5. As is shown in table 5, the isentropic efficiencies of the two models are decreased with the increase in the injection temperature, while the output expansion works are increased. Meanwhile, the isentropic efficiency and the expansion work of the TEM are always significantly larger than that of the FVM.

In addition, it can be seen from table 5 that when the injection temperature is increased from 473.15 K to 523.15 K, isentropic efficiency ($\eta_{\text{ise,eq}}$) of the FVM is approximately decreased from 76.30% to 66.64%, isentropic efficiency ($\eta_{\text{ise,eq}}$) of the TEM is approximately decreased from 82.49% to 71.16%, output expansion work ($W_{\text{exp,eq}}$) of the FVM is approximately increased from 634.8 J to 1378.3 J, and output expansion work ($W_{\text{exp,eq}}$) of the TEM is approximately increased from 680.6 J to 1429.4 J. Meanwhile, it is found that isentropic efficiency of the TEM is approximately larger by 4.5%-6.2% than that of the FVM, and output expansion work of the TEM is approximately larger by 3.7%-7.2% than that of the FVM. It was concluded that the effect of the thermodynamic disequilibrium loss on the performance of the two-phase reciprocating expander would be enhanced with the decrease of the injection temperature.

Table 5. The calculated results of the TEM and the FVM under different injection temperatures.

| Injection Temperature (K) | FVM | TEM | $\frac{\eta_{\text{ise,eq}} - \eta_{\text{ise,neq}}}{\eta_{\text{ise,neq}}} \times 100$ (%) |
|---------------------------|-----|-----|----------------------------------|
| 473.15                    | 76.30 | 82.49 | 6.19 | 7.22 |
| 478.15                    | 74.59 | 80.51 | 5.93 | 6.71 |
| 483.15                    | 73.11 | 78.80 | 5.69 | 6.26 |
| 488.15                    | 71.83 | 77.31 | 5.48 | 5.86 |
| 493.15                    | 70.73 | 76.02 | 5.29 | 5.48 |
| 498.15                    | 69.77 | 74.89 | 5.12 | 5.14 |
| 503.15                    | 68.94 | 73.91 | 4.97 | 4.82 |
| 508.15                    | 68.23 | 73.06 | 4.83 | 4.52 |
| 513.15                    | 67.61 | 72.32 | 4.71 | 4.24 |
| 518.15                    | 67.09 | 71.69 | 4.60 | 3.97 |
| 523.15                    | 66.64 | 71.16 | 4.52 | 3.71 |

5. Conclusions

In this paper, the FVM and the TEM were simultaneously proposed to analyze the two-phase expansion process and the thermodynamic disequilibrium loss of the reciprocating expander. Temperature and pressure evolution curves of the two models were obtained, and the effect of the thermodynamic disequilibrium loss on the performance of the reciprocating expander was analyzed under different rotation speeds and injection temperatures. Following conclusions could be obtained:

(a) Under the present simulation condition, the temperature difference between the liquid and the vapor mainly appeared at the initial stage of the two-phase expansion process. It was considered that this result was related to the continuous injection of the saturated liquid.

(b) The other parameters remained unchanged, when the rotation speed of the expander was increased, both the calculated isentropic efficiency and the output expansion work of the two-phase reciprocating expander were decreased. And when the injection temperature of the saturated liquid was increased, the output expansion work was increased, while the isentropic efficiency was decreased.

(c) Under the same operating conditions, the calculated isentropic efficiency and the obtained output expansion work of the TEM were always significantly higher than that of the FVM. For instance, when the rotation speed of the expander was increased from 50 r/min to 1000 r/min, isentropic efficiency of the TEM was approximately larger by 5.3%-6.3% than that of the FVM, and output expansion work of the TEM was approximately larger by 6.0%-7.7% than that of the FVM. When the injection temperature was increased from 473.15 K to 523.15 K, isentropic
efficiency of the TEM was approximately larger by 4.5%-6.2% than that of the FVM, and output expansion work of the TEM was approximately larger by 3.7%-7.2% than that of the FVM.

In conclusion, the present study found that when the rotation speed of the expander was increased or the injection temperature was decreased, the effect of the thermodynamic disequilibrium loss on the performance of the two-phase reciprocating expander would be enhanced. Therefore, when the expander speed is so high or the temperature of the heat sources is relatively low, the screw expander was considered to be more suitable for the TFC systems.

Nomenclature

| Symbols   | Abbreviations                  |
|-----------|-------------------------------|
| $a$       | $a$ atmospheric               |
| $A$       | act actual                    |
| $d$       | cyl cylinder                  |
| $f$       | $d$ dead                      |
| $h$       | eq equilibrium                |
| $l$       | exp expansion                 |
| $L$       | fla flash                     |
| $m$       | FVM flash vaporization model  |
| $n$       | inj injection                 |
| $p$       | int interface                 |
| $Q$       | ise isentropic                |
| $s$       | ide ideal                     |
| $t$       | $l$ liquid                    |
| $T$       | $neq$ non-equilibrium         |
| $u$       | $tem$ residual                |
| $V$       | ORC Organic Rankine Cycle     |
| $W$       | str stroke                    |
| $\alpha$  | TEM thermodynamic equilibrium model |
| $\Delta$  | TFC Trilateral Flash Cycle    |
| $\theta$  | $v$ vapor                     |
| $\eta$    |                               |

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