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Finite Time Thermodynamic Analysis of a Real Vapour Absorption Refrigeration System with and without Heat Leak

Sanjay Khatwani*
Department of Mechanical Engineering,
Sushila Devi Bansal College of Engineering,
Madhya Pradesh, India

Govind Maheshwari
Department of Mechanical Engineering,
Institute of Engineering and Technology,
Madhya Pradesh, India

Abstract—

This article presents finite time thermodynamic analysis of a real vapour absorption refrigeration system taking into account both external as well as internal irreversibility. The complete VAR model is assumed to be consisting of two sub-systems. One sub-system (generator-absorber assembly) is assumed as an irreversible heat engine cycle whereas another sub-system (evaporator-condenser assembly) is assumed as an irreversible refrigeration cycle. This study analyzes the effect of presence of heat leak on performance of irreversible (real) vapour absorption refrigeration (VAR) system. This is achieved by adding a heat leakage term into an irreversible model of VAR system. Overall coefficient of performance (COP) of the VAR system is optimized with respect to source/sink side heat exchanger areas using Langrangian multiplier method. Furthermore, the effects of the variations in the cycle parameters on performance deviation of the VAR system are compared with and without heat leak. Results show that effect of rate of heat leak is significant along with the other irreversibilities variations on performance deviation of the VAR system.

Keywords— Finite time thermodynamics; Rate of heat leak; Vapour absorption refrigeration; Optimization; Langrangian multiplier method.

I. INTRODUCTION

Vapour Absorption Refrigeration (VAR) Systems are substitute to vapour compression refrigeration (VCR) systems. Though, unlike VCR systems, the necessary input to absorption systems is in the form of heat. (VAR) Systems are ideal when low-grade energy such as waste heat or solar energy is accessible. In view of the fact that conventional absorption systems use natural refrigerants such as water or ammonia, they are environment gracious as they eliminate the use of CFC and HCFC refrigerants. Moreover, these systems present other benefits, such as high consistency, low maintainability and a quiet and vibration-free function [1]-[2]. The Coefficient of Performance (COP) of these systems is relatively lower than conventional VCR systems [3]-[4]. Hence there is a great need to optimize these systems.

Classical thermodynamics provides bounds on thermodynamic performance characteristics based on reversible assumptions [5]-[6]. These consequential bounds are not reasonable because reversible operation means either zero power or infinite system dimension. Finite-time thermodynamics (FTT) extends thermodynamic analysis to take account of finite-time constraints similar to finite heat transfer rates, heat leaks, and friction. FTT approach has been functional to various thermodynamic systems [7]-[11]. Curzon and Ahlborn establish an expression for the efficiency at maximum power for a heat engine [12]. Many studies have analyzed the FTT optimization of absorption refrigeration systems to improve their performance [13]-[21]. These articles generally chose coefficient of performance [22], cooling load [23]-[27] and the total heat transfer area [28] as the objective functions. In irreversible thermodynamic systems heat leaks are unavoidable and such losses have considerable influence on the performance of the systems [29].

In the present study, the effect of heat leak on performance of a real VAR system is analyzed. This VAR model is considered to be an integrated system coupled by a heat engine and a refrigerator. Using finite time thermodynamics, an optimal relation for overall COP is derived in presence of heat leak by using langrangian multiplier method. First, the performance of this VAR system is evaluated in absence of heat leak as this modelling only accounts the heat resistance and irreversibilities due to internal dissipation of the working fluid. Further the same analysis is performed in presence of heat leak along with the other irreversibilities and the two results are compared.

II. ANALYSIS OF VAR SYSTEM

VAR system as shown in **Figure 1** is assumed to be a combined system, coupled by a heat engine (generator-absorber assembly) and a refrigerator (evaporator-condenser assembly). This model includes the heat leakage rate (QL) from the heat sink at temperature (TC) to the cold reservoir at temperature (TE).

A. Analysis of Generator-Absorber Assembly:

This sub-system of VAR system i.e. heat engine cycle part will not affected by introducing heat leak term in the model. Hence expression for optimized thermal efficiency (η) of heat engine cycle will be same as derived by authors in a previous study [30], by applying langrangian multiplier method for optimization subject to conditions of total heat transfer area of generator-absorber assembly (A_h) where A_h is sum of heat transfer area of generator (A_g) and heat transfer area of absorber (A_a). This expression is given by

$$\eta = 1 - \frac{\left(\frac{T_A}{\psi_S}\right)}{\left(T_G - \frac{Q_G}{A_h}\left(\frac{1}{\sqrt{U_g}} + \frac{1}{\sqrt{U_a \psi_s}}\right)^2\right)} \tag{1}$$

Where T_A is temperature of absorber thermal reservoir, T_G is temperature of generator thermal reservoir, Ψ_S is internal irreversibility coefficient of generator-absorber assembly, Q_G is heat transfer rate from the heat source at temperature T_G to the generator, U_g and U_a are overall heat transfer coefficients in the generator and evaporator, respectively.

B. Analysis of Condenser-Evaporator Assembly:

The included heat leak is between the heat sink at temperature (T_C) and the space to be cooled at temperature (T_E). Hence only this sub-system i.e. refrigerator cycle will be affected by heat leak consideration. Derivation of optimal expression for coefficient of performance (COP) of condenser-evaporator assembly in presence of rate of heat leak using langrangian multiplier method is as follows:

Rate of heat transfer (Q_E) from low temperature reservoir i.e. space to be cooled at temperature T_E to the evaporator is given by

$$Q_E = U_e A_e (T_E - T_e) \tag{2}$$

where U_e, A_e and T_e are overall heat transfer coefficient, heat transfer area and working fluid temperature, respectively in the evaporator.

Similarly, rate of heat rejection from condenser (Q_C) to the heat sink is given by

$$Q_C = U_c A_c (T_c - T_C) \tag{3}$$

where U_c, A_c and T_c are overall heat transfer coefficient, heat transfer area and working fluid temperature, respectively in the condenser.

Heat leakage rate Q_L at temperature T_C from the heat sink to the cold reservoir at temperature T_E is given by

$$Q_L = K_L (T_C - T_E) \tag{4}$$

where K_L is the heat leakage coefficient as expressed in [31].

According to first law of thermodynamics, work input rate of refrigeration cycle is given by [21]

$$W = U_c A_c (T_c - T_C) - U_e A_e (T_E - T_e) - K_L (T_C - T_E) \tag{5}$$

According to second law of thermodynamics [30],

$$\Delta S = \frac{Q_E - Q_L}{T_e} - \psi'_S \frac{U_c A_c \,(T_c - T_C)}{T_c} = 0 \tag{6}$$

where ψ'_s is an internal irreversibility coefficient of condenser-evaporator assembly, $\psi'_s = 1$, corresponds to endoreversible system and $\psi'_s < 1$ is for real system. Total heat transfer area of condenser-evaporator assembly (A_r) is sum of heat transfer area of condenser (A_c) and heat transfer area of evaporator (A_e).

Working fluid temperature of evaporator (T_e) may be obtained rearranging the terms of Eq. (2)

$$T_e = T_E - \frac{Q_E}{U_e A_e} \tag{7}$$

Working fluid temperature of condenser (Tc) may be obtained by using Eq. (2) and Eq. (6) and is given by

$$T_c = \frac{T_e \psi'_S U_c A_c T_C}{\left(\left(T_E - \frac{Q_E}{U_e A_e}\right)(\psi'_s U_c A_c)\right) - (Q_E - Q_L)} \tag{8}$$

Now, Eq. (6) can be written as

$$\frac{Q_C}{Q_E - Q_L} = \frac{\frac{T_c}{\psi'_S}}{T_e} \tag{9}$$

C. COP of Condenser-Evaporator Assembly:

COP of condenser-evaporator assembly is given by

$$COP = \left(\frac{Q_E - Q_L}{Q_C - Q_E - Q_L}\right) \tag{10}$$

By rearranging the terms of Eq (10), we have

$$COP = \left(\frac{Q_C}{Q_E - Q_L} - 1\right)^{-1} \tag{11}$$

But, by Eq (8), Eq (9), and Eq (11); COP of condenser-evaporator assembly can be written as

$$COP = \left(\frac{T_C/\psi'_S}{T_E - Q_E\left(\frac{1}{U_e A_e} + \frac{1}{U_c A_c \psi'_s}\right) + \frac{Q_L}{U_c A_c \psi'_s}} - 1 \right)^{-1} \tag{12}$$

D. Optimization of COP of Condenser-Evaporator assembly:

Optimization of COP of condenser-evaporator assembly can be achieved by applying Langrangian multiplier method subject to conditions of total heat transfer area of the refrigeration cycle ($A_r = A_c + A_e$). Here, optimization is performed with respect to heat transfer area of condenser (A_c) and heat transfer area of evaporator (A_e).

Hence, by introducing Langrangian operator:

L(COP)= COP + λ(C)

where constraint, C is given by, $A_r – A_c – A_e = 0$

$$L(COP) = \left(1 - \frac{\frac{T_C}{\psi'_S}}{T_E - Q_E\left(\frac{1}{U_e A_e} + \frac{1}{U_c A_c \psi'_s}\right) + \frac{Q_L}{U_c A_c \psi'_s}} \right)^{-1} + \lambda(A_r - A_c - A_e) \tag{13}$$

Euler Langrangian equation,

$$\frac{\partial L}{\partial A_e} = 0 \quad and \quad \frac{\partial L}{\partial A_c} = 0 \quad gives$$

$$\frac{{A_e}^2}{{A_c}^2} = \frac{\psi'_s U_c}{U_e}\left(\frac{Q_E}{Q_E - Q_L}\right) \tag{14}$$

$$\frac{A_e}{A_c} = \sqrt{\frac{\psi'_S U_c}{U_e}} \sqrt{\frac{Q_E}{Q_E - Q_L}} \tag{15}$$

Using the Eq. (12) and Eq. (15), we have

$$COP = \left(\frac{\left(\frac{T_C}{\psi'_S}\right)}{\left(T_E - \frac{Q_E}{A_r}\left(\frac{1}{U_e} + \frac{\sqrt{(1-\zeta)}}{\sqrt{U_e}\sqrt{\psi'_s U_c}} + \left(\frac{1+\zeta}{\psi'_s U_c}\right) * \left(1 + \frac{\sqrt{\psi'_s U_c}}{\sqrt{U_e}\sqrt{1-\zeta}}\right) \right) \right)} - 1 \right)^{-1} \tag{16}$$

where ζ is heat leak factor and is defined as the fraction of rate of heat leakage for a given rate of heat transfer to evaporator. Hence,

$$\zeta = \frac{Q_L}{Q_E} \tag{17}$$

Using the Eq. (4) and Eq. (17), we have

$$\zeta = \frac{K_L(T_C - T_E)}{Q_E} \tag{18}$$

E. Optimization of Overall Coefficient of Performance of The VAR System:

Overall coefficient of performance (ω) is the product of thermal efficiency of generator–absorber assembly and COP of the evaporator–condenser assembly. Optimal overall coefficient of performance (ω) of the VAR system is given by:

$\omega = \eta * COP$

Using the Eq. (1) and Eq. (16), we have

ω =

$$\left(1 - \frac{\left(\frac{T_A}{\psi_S}\right)}{\left(T_G - \frac{Q_G}{A_h}\left(\frac{1}{\sqrt{U_g}} + \frac{1}{\sqrt{U_a \psi_s}}\right)\right)^2} \right) \left(\frac{\left(\frac{T_C}{\psi'_S}\right)}{\left(T_E - \frac{Q_E}{A_r}\left(\frac{1}{U_e} + \frac{\sqrt{(1-\zeta)}\sqrt{U_e}}{\sqrt{\psi'_s U_c}} + \left(\frac{1+\zeta}{\psi'_s U_c}\right) * \left(1 + \frac{\sqrt{\psi'_s U_c}}{1-\zeta}\right) \right) \right)} - 1 \right)^{-1} \tag{19}$$

III. RESULTS AND DISCUSSION

The effect of variation of rate of heat leak on optimal performance evaluation of VAR system is analyzed in this study. Figure 2 shows that value of overall coefficient of performance is decreases considerably with increase in values of heat leak factor (ζ) from zero (when Q_L is zero) to one (when Q_L is equal to Q_E). Table 1 shows the optimal values of overall COP for various internal and external irreversibilities. The optimal values of overall COP are evaluated with and without heat leak in a totally reversible system, an endoreversible system and a totally irreversible system for the given values of other parameters.

Further effects of variation of other parameters like evaporator temperature and distribution of heat exchange areas on overall coefficient of performance of the system are also compared in both cases i. e. in the absence of heat leak as well as in the presence of heat leak. Authors have used the same values of overall heat transfer coefficients and same value of initial parameters (the fixed parameters) for the present comparative analysis.

In the absence of heat leak, overall COP increases from 0.88 to 1.15 for a variation of TE from 268 K to 278 K i.e. for 10 degrees. It means that COP varies in the range of 0.27. In the presence of heat leak and for the variation of TE in the same range, overall COP increases from 0.84 to 1.10. It means that COP varies in the range of 0.26. This variation is approximately similar to first case but the absolute values of COP are lower as compared to the first case. This comparison is shown in figures 3 and 4 respectively.

Figures 5 and 6 shows the effect of variation in the distribution of heat exchanger areas on performance of VAR system in the absence of heat leak and in the presence of heat leak respectively. This analysis shows that the surface area, A_r of evaporator-condenser assembly should be more than surface area, Ah of generator-absorber assembly in both cases for maximum COP but in the presence of heat leak, the optimum surface area, A_r will be even more then in the absence of heat leak. These results are similar to those derived by Zheng, et al. [18] in a study of irreversible VAR model which included the heat leak from the heat sink to the space to be cooled.

IV. CONCLUSION

This analysis of real vapour absorption refrigeration system incorporates the influence of heat leak and hence this model permits a more realistic prediction in the performance of irreversible VAR system. This system is assumed to be attached with an irreversible heat engine and an irreversible refrigerator. Langrangian multiplier method is applied to establish the optimal expression for overall coefficient of performance. Effect of variation of rate of heat leak will have no effect on heat engine cycle of VAR system. Hence optimization of only evaporator-condenser assembly (irreversible refrigerator cycle) is performed with respect to evaporator and condenser heat exchanger areas in presence of rate of heat leak. Study of various graphs shows the effect of different input characteristics on performance of vapour absorption system. It is found that variation of rate of heat leak has considerable effect on overall coefficient of performance but these effects are not as severe as due to internal irreversibility variation. The outcomes of this study are helpful for optimal design and performance improvement of absorption refrigeration cycles.

NOMENCLATURE

A	=surface area of heat exchanger (m^2)
COP	=coefficient of performance of evaporator–condenser assembly
L	=Langrangian operator
Q	=heat transfer rate to and from the system (kW)
Ψ_S	=internal irreversibility coefficient of generator-absorber assembly
Ψ'_S	=internal irreversibility coefficient evaporator–condenser assembly
S	=entropy ($kJ \cdot K^{-1}$)
T	=temperature (K)
U	=overall heat transfer coefficient ($kW \cdot m^{-2} \cdot K^{-1}$)
W	=work output of heat engine cycle or work input to refrigeration cycle (kJ)
Greek symbols	
ω	=overall COP of vapour absorption refrigeration system
λ	=Langrangian multiplier
η	=efficiency of heat engine
ζ	=heat leak factor
Subscripts	
a	=absorber
c	=condenser
e	=evaporator
g	=generator
G	=generator thermal reservoir
A	=absorber thermal reservoirs
C	=condenser thermal reservoir
E	=evaporator thermal reservoir

FIGURES AND TABLES

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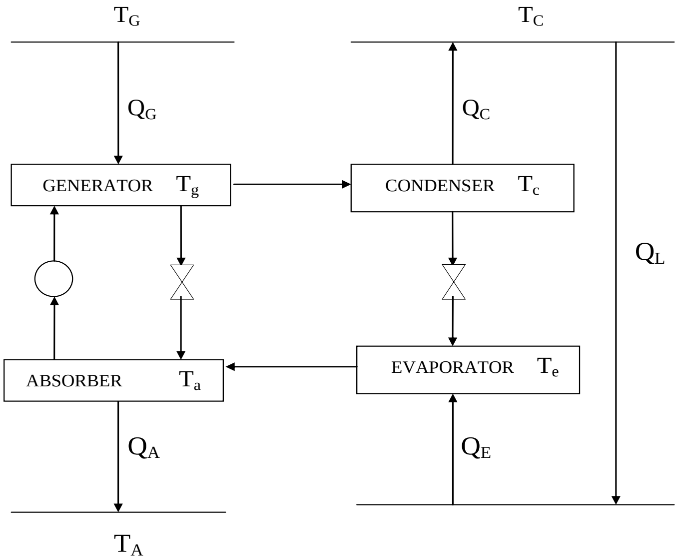

Figure 1. Representation of single stage vapour absorption refrigeration system

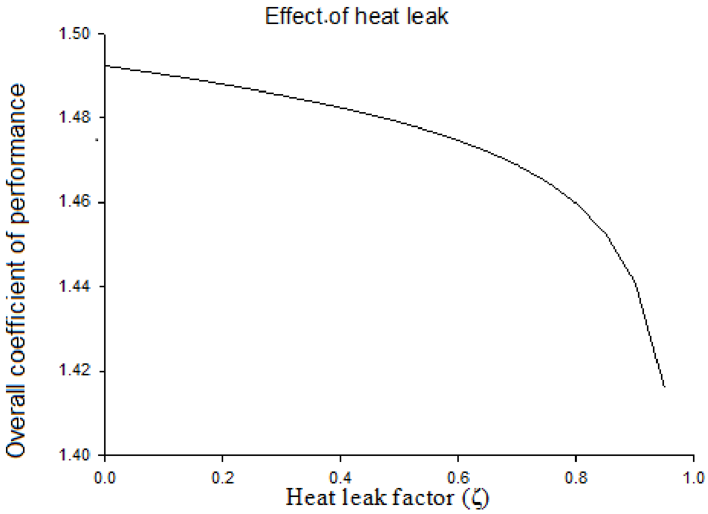

Figure 2: Effect of heat leak factor on overall coefficient of performance

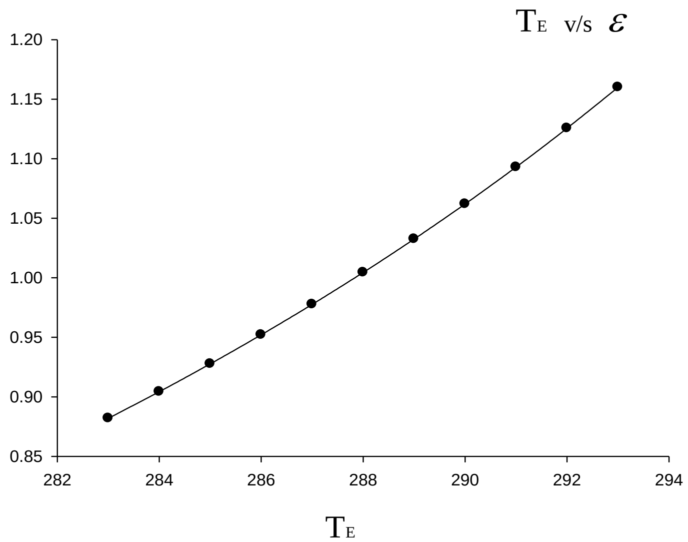

Figure 3: Effect of T_E on overall coefficient of performance in the absence of heat leak

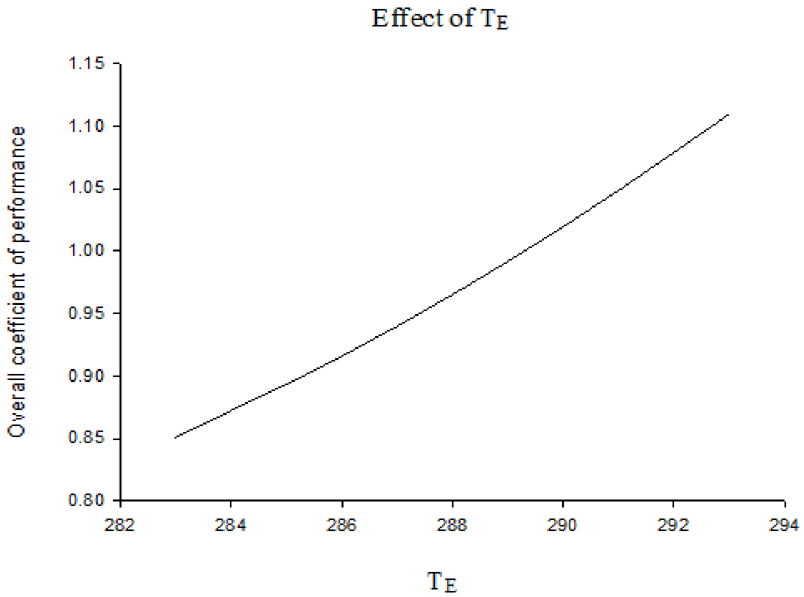

Figure 4: Effect of T_E on overall coefficient of performance in the presence of heat leak

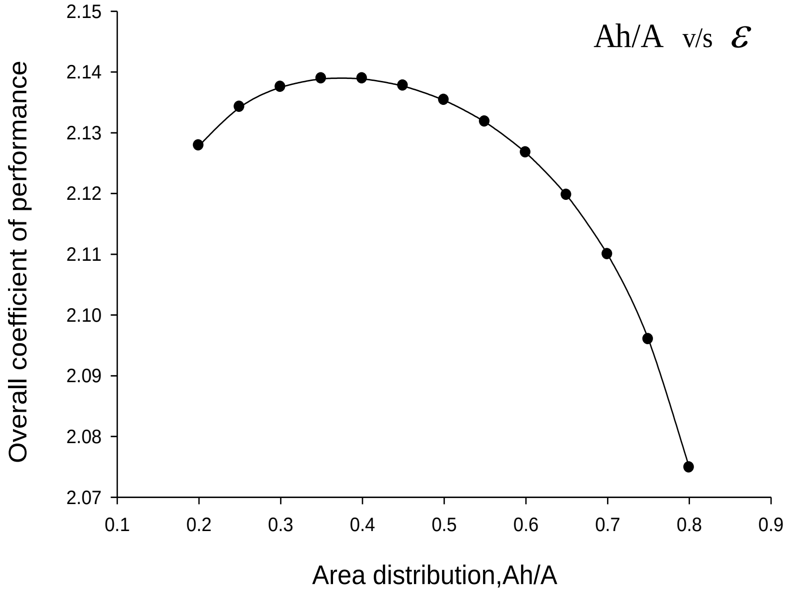

Figure 5: Effect of A_h/A on overall coefficient of performance in the absence of heat leak

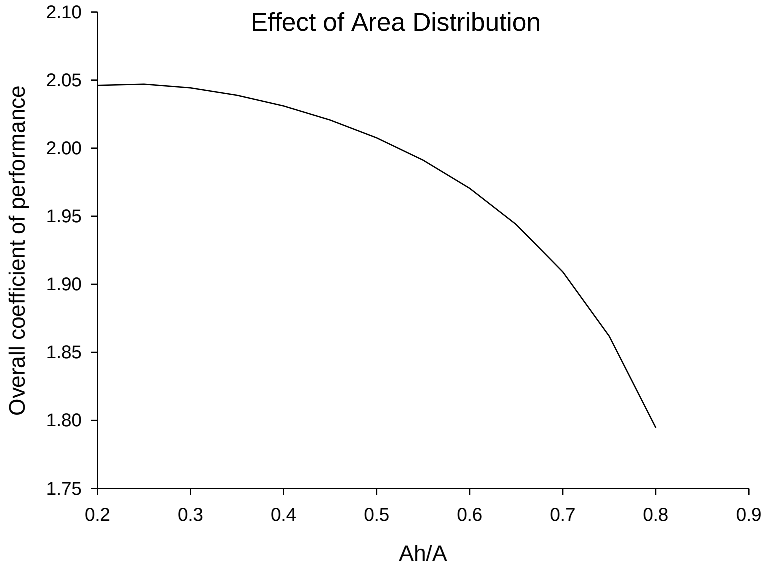

Figure 6: Effect of A_h/A on overall coefficient of performance in the presence of heat leak

Table 1: The optimal values of overall COP for various inrernal and external irreversibilities

Overall COP	In Absence of Heat Leak	In Presence of Heat Leak	Fixed Parameters
For Totally Reversible System	2.198 (for $\zeta = 0$)	-	T_A = 318 K, T_C = 310 K, T_E = 288 K, T_G = 393K
For Endoreversible System	2.1388 (for $\zeta = 0$)	2.0469 (for $\zeta = 0.24$)	T_A = 318 K, T_C = 310 K, T_E = 288 K, T_G = 393K $\psi_s = \psi'_s = 1.0$, A_h/A = Ar/A = 0.5, $U_g = U_a = U_e = U_c$ = 10.0 kW·K−1·m−2
For Totally Irreversible System	1.044 (for $\zeta = 0$)	0.9654 (for $\zeta = 0.24$)	T_A = 318 K, T_C = 310 K, T_E = 288 K, T_G = 393K $\psi_s = \psi'_s = 0.95$, A_h/A = Ar/A = 0.5, $U_g = U_a = U_e = U_c$ = 10.0 kW·K−1·m−2

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