Chapter 7

Exergetic Costs for Thermal Systems

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Additional information is available at the end of the chapter

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Abstract

Exergy costing to estimate the unit cost of products from various power plants and refrigeration system is discussed based on modified-productive structure analysis (MOPSA) method. MOPSA method provides explicit equations from which quick estimation of the unit cost of products produced in various power plants is possible. The unit cost of electricity generated by the gas-turbine power plant is proportional to the fuel cost and inversely proportional to the exergetic efficiency of the plant and is affected by the ratio of the monetary flow rate of non-fuel items to the monetary flow rate of fuel. On the other hand, the unit cost of electricity from the organic Rankine cycle power plant with heat source as fuel is proportional to the unit cost of heat and the ratio of the monetary flow rate of non-fuel items to the generated electric power, independently. For refrigeration system, the unit cost of heat is proportional to the consumed electricity and inversely proportional to the coefficient of performance of the system, and is affected by the ratio of the monetary flow rate of non-fuel items to the monetary flow rate of consumed electricity.

Keywords: exergy, thermoeconomics, unit exergy cost, power plant, refrigeration system

1. Introduction

Exergy analysis is an effective tool to accurately predict the thermodynamic performance of any energy system and the efficiency of the system components and to quantify the entropy generation of the components [1–3]. By this way, the location of irreversibilities in the system is determined. Furthermore, thermoeconomic analysis provides an opportunity to estimate the unit cost of products such as electricity and/or steam from thermal systems [4, 5] and quantifies monetary loss due to irreversibility for the components in the system [6]. Also, thermoeconomic analysis provides a tool for optimum design and operation of complex thermal systems such as cogeneration power plant [7] and efficient integration of new and renewable energy systems [8]. Recently, performance evaluation of various plants such as sugar plant [9], drying plant [10], and geothermal plant [11] has been done using exergy and thermoeconomic analyses. In this chapter, a procedure to obtain the unit cost of products from the power plants and refrigeration system is presented by using modified-productive structure analysis (MOPSA) method. The power plants considered in this chapter are gas-turbine power plant and organic Rankine cycle power plant. These systems generate electricity as a product by consuming the heat resultant from combustion of fuel and by obtaining heat from any hot stream as fuel, respectively. In addition, MOPSA method is applied to an air-cooled air conditioning system, which removes heat like a product while the consumed electricity is considered as fuel. Explicit equations to estimate the unit cost of electricity generated by the gas-turbine power plant and organic Rankine cycle plant, and the unit cost of heat for the refrigeration system are obtained and the results are presented.

2. A thermoeconomic method: modified productive structure analysis (MOPSA)

2.1. Exergy-balance and cost balance equations

A general exergy-balance equation that can be applied to any component of thermal systems may be formulated by utilizing the first and second law of thermodynamics [12]. Including the exergy losses due to heat transfer through the non-adiabatic components, and with decomposing the material stream into thermal and mechanical exergy streams, the general exergy-balance equation may be written as [6]

$$\dot{E}_x^{CHE} + \left(\sum_{inlet} \dot{E}_x^T - \sum_{outlet} \dot{E}_x^T\right) + \left(\sum_{inlet} \dot{E}_x^P - \sum_{outlet} \dot{E}_x^P\right) + T_o\left(\sum_{inlet} \dot{S}_i - \sum_{outlet} \dot{S}_i + \dot{Q}_{cv}/T_o\right) = \dot{E}_x^W \quad (1)$$

The fourth term in Eq. (1) is called the neg-entropy which represents the negative value of the rate of lost work due to entropy generation, which can be obtained from the second law of thermodynamics. The term $\dot{E}_x^{CHE}$ in Eq. (1) denotes the rate of exergy flow of fuel, and $\dot{Q}_{cv}$ in the fourth term denotes heat transfer interaction between a component and the environment, which can be obtained from the first law of thermodynamics.

$$\dot{Q}_{cv} + \sum_{in} \dot{H}_i = \sum_{out} \dot{H}_i + \dot{W}_{cv} \quad (2)$$

However, the quantity $\dot{Q}_{cv}$ for each component, which is usually not measured, may be obtained from the corresponding exergy-balance equation with the known values of the entropy flow rate at inlet and outlet.

Exergy, which is the ability to produce work, can be defined as the differences between the states of a stream or matter at any given particular temperature and pressure and the state of the same stream at a reference state. The exergy stream per unit mass is calculated by the following equation:

$$e_x = h(T,P) - h_{ref}\left(T_{ref}, P_{ref}\right) - T_o\left[s(T,P) - s_{ref}\left(T_{ref}, P_{ref}\right)\right] \tag{3}$$

where T is temperature, P is pressure, and the subscript ref denotes reference values. The exergy stream per unit mass can be divided into its thermal (T) and mechanical (P) components as follows [3]:

$$e_x = e_x^T + e_x^P \tag{4}$$

and

$$e_x^T = \left[h(T,P) - h\left(T_{ref}, P\right)\right] - T_o\left[s(T,P) - s\left(T_{ref}, P\right)\right] \tag{5}$$

$$e_x^P = \left[h\left(T_{ref}, P\right) - h_{ref}\left(T_{ref}, P_{ref}\right)\right] - T_o\left[s\left(T_{ref}, P\right) - s_{ref}\left(T_{ref}, P_{ref}\right)\right] \tag{6}$$

Assigning a unit exergy cost to every exergy stream, the cost-balance equation corresponding to the exergy-balance equation for any component in a thermal system [13] may be written as

$$\begin{aligned} &\dot{E}_x^{CHE} C_0 + \left(\sum_{inlet} \dot{E}_{x,i}^T - \sum_{outlet} \dot{E}_{x,i}^T\right) C_T + \left(\sum_{inlet} \dot{E}_{x,i}^P - \sum_{outlet} \dot{E}_{x,i}^P\right) C_P \\ &+ T_0\left(\sum_{inlet} \dot{S}_i - \sum_{outlet} \dot{S}_j + \dot{Q}_{cv}/T_o\right) C_S + \dot{Z}_k = \dot{E}_x^W C_W, \end{aligned} \tag{7}$$

The term $\dot{Z}_k$ includes all financial charges associated with owning and operating the kth component in the thermal system. We call the thermoeconomic analysis based on Eqs. (1) and (7) as modified-productive structure analysis (MOPSA) method because the cost-balance equation in Eq. (7) yields the productive structure of the thermal systems, as suggested and developed by Lozano and Valero [5] and Torres et al. [14]. MOPSA has been proved as very useful and powerful method in the exergy and thermoeconomic analysis of large and complex thermal systems such as a geothermal district heating system for buildings [15] and a high-temperature gas-cooled reactor coupled to a steam methane reforming plant [16]. Furthermore, the MOPSA can provide the interaction between the components in the power plant through the entropy flows [17] and a reliable diagnosis tool to find faulty components in power plants [18].

2.2. Levelized cost of system components

All costs due to owning and operating a plant depend on the type of financing, the required capital, the expected life of components, and the operating hours of the system. The annualized (levelized) cost method of Moran [1] was used to estimate the capital cost of components in this study. The amortization cost for a particular plant component is given by

$$PW = C_i - S_n PWF(i, n) \tag{8}$$

The present worth of the component is converted to annualized cost by using the capital recovery factor CRF(i, n):

$$\dot{C}(\$/year) = PW \cdot CRF(i, n) \tag{9}$$

The capital cost rate of the kth component of the thermal system can be obtained by dividing the levelized cost by annual operating hours δ.

$$\dot{Z}_k = \phi_k \dot{C}_k / 3600\delta \tag{10}$$

The maintenance cost is taken into consideration through the factor ϕ_k. It is noted that the operating hours of thermal systems is largely dependent on the energy demand patterns of end users [19].

3. Gas-turbine power plant

A schematic of a 300 MW gas-turbine power plant considered in this chapter is shown in **Figure 1**. The system includes five components: air compressor (1), combustor (2), gas turbines (3), fuel preheater (5), and fuel injector (6). A typical mass flow rate of fuel to the combustor at full load condition is 8.75 kg/s and the air–fuel mass ratio is about 50.0. Thermal and mechanical exergy flow rates and entropy flow rate at various state points shown in

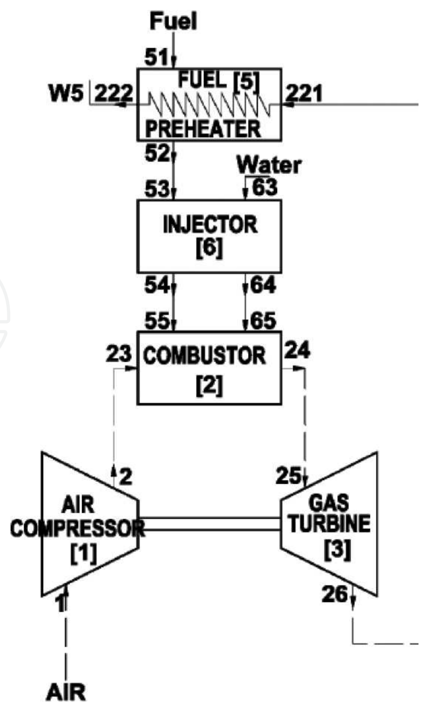

Figure 1. Schematic of a gas-turbine power plant.

States	$\dot{m}$ (kg / s)	P (MPa)	T (°C)	$\dot{E}_x^T$ (MW)	$\dot{E}_x^P$ (MW)	$\dot{S}$ (MW / K)
1	862.722	0.103	15.000	0.000	−0.558	0.121
2	862.722	1.025	323.589	88.176	164.572	0.193
23	862.722	1.025	323.589	88.176	164.572	0.193
24	891.056	1.025	1130.775	702.452	173.550	1.201
25	891.056	1.025	1130.775	702.452	173.550	1.201
26	891.056	0.107	592.700	261.996	2.661	1.262
51	17.500	0.103	15.000	0.000	0.018	0.001
52	17.500	0.103	185.000	1.563	0.018	0.018
53	17.500	0.103	185.000	1.563	0.018	0.018
54	17.500	1.025	415.314	7.735	5.337	0.021
55	17.500	1.025	415.314	7.735	5.337	0.021
63	10.833	0.103	(1.000)	6.064	0.000	0.004
64	10.833	1.025	418.176	12.338	0.010	0.006
65	10.883	1.025	418.176	12.338	0.010	0.006
221	11.111	3.540	220.100	2.417	0.038	0.028
222	11.111	3.540	72.941	0.239	0.038	0.011

Table 1. Property values and thermal, and mechanical exergy flows and entropy production rates at various state points in the gas-turbine power plant at 100% load condition.

Figure 1 are presented in **Table 1**. These flow rates were calculated based on the values of measured properties such as pressure, temperature, and mass flow rate at various state points.

3.1. Exergy-balance equation for gas-turbine power plant

The following exergy-balance equations can be obtained by applying the general exergy-balance equation given in Eq. (1) to each component in the gas-turbine power plant.

Air compressor

$$\left(\dot{E}_{x,1}^{T} - \dot{E}_{x,2}^{T}\right) + \left(\dot{E}_{x,1}^{P} - \dot{E}_{x,2}^{P}\right) + T_o\left[\left(\dot{S}_1 - \dot{S}_2\right) + \dot{Q}_{[1]}/T_o\right] = \dot{E}_{x,[1]}^{W} \tag{11}$$

Combustor

$$\begin{aligned}\dot{E}_x^{CHE} &+ \left(\dot{E}_{x,23}^{T} + \dot{E}_{x,55}^{T} + \dot{E}_{x,65}^{T} - \dot{E}_{x,24}^{T}\right) + \left(\dot{E}_{x,23}^{P} + \dot{E}_{x,55}^{P} + \dot{E}_{x,65}^{P} - \dot{E}_{x,24}^{P}\right) \\ &+ T_o\left[\dot{S}_{23} + \dot{S}_{55} + \dot{S}_{65} - \dot{S}_{24} + \dot{Q}_{[2]}/T_o\right] = 0\end{aligned} \tag{12}$$

Turbine

$$\left(\dot{E}_{x,25}^{T} - \dot{E}_{x,26}^{T}\right) + \left(\dot{E}_{x,25}^{P} - \dot{E}_{26}^{P}\right) + T_o\left(\dot{S}_{25} - \dot{S}_{26} + \dot{Q}_{[3]}/T_o\right) = \dot{E}_{x,[3]}^{W} \tag{13}$$

Fuel preheater

$$\left(\dot{E}^T_{x,51} - \dot{E}^T_{x,52}\right) + \left(\dot{E}^P_{x,51} - \dot{E}^P_{x,52}\right) + \left(\dot{E}^T_{x,221} - \dot{E}^T_{x,222}\right) + \left(\dot{E}^P_{x,221} - \dot{E}^P_{x,222}\right) + T_o\left[\dot{S}_{51} - \dot{S}_{52} + \dot{S}_{221} - \dot{S}_{222} + \dot{Q}_{[5]}/T_o\right] = 0 \tag{14}$$

Steam injector

$$\left(\dot{E}^T_{x,53} - \dot{E}^T_{x,54} + \dot{E}^T_{x,63} - \dot{E}^T_{x,64}\right) + \left(\dot{E}^P_{x,53} - \dot{E}^P_{x,54} + \dot{E}^P_{x,63} - \dot{E}^P_{x,64}\right) + T_o\left(\dot{S}_{53} - \dot{S}_{54} + \dot{S}_{63} - \dot{S}_{64} + \dot{Q}_{[6]}/T_o\right) = \dot{E}^W_{x,[6]} \tag{15}$$

Boundary

$$\left(\dot{E}^T_{x,1} + \dot{E}^T_{x,51} + \dot{E}^T_{x,63} - \dot{E}^T_{x,26}\right) + \left(\dot{E}^P_{x,1} + \dot{E}^P_{x,51} + \dot{E}^P_{x,63} - \dot{E}^P_{x,26}\right) + \left(\dot{E}^T_{x,221} - \dot{E}^T_{x,222}\right) + \left(\dot{E}^P_{x,221} - \dot{E}^P_{x,222}\right) + T_o\left(\dot{S}_1 + \dot{S}_{51} + \dot{S}_{63} - \dot{S}_{26} + \dot{S}_{221} - \dot{S}_{222} + \dot{Q}_{[boun]}/T_o\right) = 0 \tag{16}$$

The net flow rates of the various exergies crossing the boundary of each component in the gas-turbine power plant at 100% load condition are shown in **Table 2**. Positives values of exergies indicate the exergy flow rate of "products," while negative values represent the exergy flow rate of "resources" or "fuel." The irreversibility rate due to entropy production in each component acts as a product in the exergy-balance equation. The sum of exergy flow rates of products and resources equals to zero for each component and the overall system; this zero sum indicates that perfect exergy balances are satisfied.

3.2. Cost-balance equation for gas-turbine power system

When the cost-balance equation is applied to a component, a new unit cost must be assigned to the component's principle product, whose unit cost is expressed as Gothic letter. After a unit

Component	Net exergy flow rates (MW)				Irreversibility rate (MW)
	$\dot{E}^W_{(k)}$	$\dot{E}^{CHE}_x$	$\dot{E}^T_x$	$\dot{E}^P_x$	
Compressor	−274.04	0.00	88.18	165.13	20.73
Combustor	0.00	−881.22	594.20	3.63	283.39
Gas turbine	593.74	0.00	−440.46	−170.89	17.61
Fuel preheater	0.00	0.00	−0.61	0.00	0.61
Steam injector	−18.68	0.00	11.91	5.33	1.44
Boundary	0.00	0.00	−253.22	−3.20	
Total	301.02	−881.22	253.22	3.20	323.78

Table 2. Exergy balances of each component in the gas-turbine power plant at 100% load condition.

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cost is assigned to the principal product of each component, the cost-balance equations corresponding to the exergy-balance equations are as follows:

Air compressor

$$\left(\dot{E}^{T}_{x,1}-\dot{E}^{T}_{x,2}\right)C_T+\left(\dot{E}^{P}_{x,1}-\dot{E}^{P}_{x,2}\right)\mathbf{C_{1P}}+T_o\left[\left(\dot{S}_1-\dot{S}_2\right)+\dot{Q}_{[1]}/T_o\right]C_S+\dot{Z}_{[1]}=\dot{E}^{W}_{[1]}C_W \tag{17}$$

Combustor

$$\begin{aligned}&\dot{E}^{CHE}_{x}C_o+\left(\dot{E}^{T}_{x,23}+\dot{E}^{T}_{x,55}+\dot{E}^{T}_{x,65}-\dot{E}^{T}_{x,24}\right)\mathbf{C_{2T}}+\left(\dot{E}^{P}_{x,23}+\dot{E}^{P}_{x,55}+\dot{E}^{P}_{x,65}-\dot{E}^{P}_{x,24}\right)C_P\\&\quad+T_o\left[\dot{S}_{23}+\dot{S}_{55}+\dot{S}_{65}-\dot{S}_{24}+\dot{Q}_{[2]}/T_o\right]C_S+\dot{Z}_{[2]}=0\end{aligned} \tag{18}$$

Turbine

$$\left(\dot{E}^{T}_{x,25}-\dot{E}^{T}_{x,26}\right)C_T+\left(\dot{E}^{P}_{x,25}-\dot{E}^{P}_{26}\right)C_P+T_o\left(\dot{S}_{25}-\dot{S}_{26}+\dot{Q}_{[3]}/T_o\right)C_S+\dot{Z}_{[3]}=\dot{E}^{W}_{[3]}\mathbf{C_W} \tag{19}$$

Fuel preheater

$$\begin{aligned}&\left(\dot{E}^{T}_{x,51}-\dot{E}^{T}_{x,52}+\dot{E}^{T}_{x,221}-\dot{E}^{T}_{x,222}\right)\mathbf{C_{5T}}+\left(\dot{E}^{P}_{x,51}-\dot{E}^{P}_{x,52}+\dot{E}^{P}_{x,221}-\dot{E}^{P}_{x,222}\right)C_P\\&\quad+T_o\left[\dot{S}_{51}-\dot{S}_{52}+\dot{S}_{221}-\dot{S}_{222}+\dot{Q}_{[5]}/T_o\right]C_P+\dot{Z}_{[5]}=0\end{aligned} \tag{20}$$

Steam injector

$$\begin{aligned}&\left(\dot{E}^{T}_{x,53}-\dot{E}^{T}_{x,54}+\dot{E}^{T}_{x,63}-\dot{E}^{T}_{x,64}\right)C_T+\left(\dot{E}^{P}_{x,53}-\dot{E}^{P}_{x,54}+\dot{E}^{P}_{x,63}-\dot{E}^{P}_{x,64}\right)\mathbf{C_{6P}}\\&\quad+T_o\left(\dot{S}_{53}-\dot{S}_{54}+\dot{S}_{63}-\dot{S}_{64}+\dot{Q}_{[6]}/T_o\right)C_S+\dot{Z}_{[6]}=\dot{E}^{W}_{x,[6]}C_W\end{aligned} \tag{21}$$

Applying the general cost-balance equation to the system components, five cost-balance equations are derived. However, these equations present eight unknown unit exergy costs, which are C_T, C_S, C_W, C_{1P}, C_{2T}, C_P, C_{5T}, and C_{6P}. To calculate the value of these unknown unit exergy costs, three more cost-balance equations are required. These additional equations can be obtained from the thermal and mechanical junctions and boundary of the plant.

Thermal exergy junction

$$\begin{aligned}&\left(\dot{E}^{T}_{x,23}+\dot{E}^{T}_{x,55}+\dot{E}^{T}_{x,65}-\dot{E}^{T}_{x,24}+\dot{E}^{T}_{x,51}-\dot{E}^{T}_{x,52}+\dot{E}^{T}_{x,221}-\dot{E}^{T}_{x,222}\right)\mathbf{C_T}\\&=\left(\dot{E}^{T}_{x,23}+\dot{E}^{T}_{x,55}+\dot{E}^{T}_{x,65}-\dot{E}^{T}_{x,24}\right)C_{2T}+\left(\dot{E}^{T}_{x,51}-\dot{E}^{T}_{x,52}+\dot{E}^{T}_{x,221}-\dot{E}^{T}_{x,222}\right)C_{5T}\end{aligned} \tag{22}$$

Mechanical exergy junction

$$\begin{aligned}&\left(\dot{E}^{P}_{x,1}-\dot{E}^{P}_{x,2}+\dot{E}^{P}_{x,53}-\dot{E}^{P}_{x,54}+\dot{E}^{P}_{x,63}-\dot{E}^{P}_{x,64}\right)\mathbf{C_P}=\left(\dot{E}^{P}_{x,1}-\dot{E}^{P}_{x,2}\right)C_{1P}\\&\quad+\left(\dot{E}^{P}_{x,53}-\dot{E}^{P}_{x,54}+\dot{E}^{P}_{x,63}-\dot{E}^{P}_{x,64}\right)C_{6P}\end{aligned} \tag{23}$$

Boundary

$$\left(\dot{E}^{T}_{x,1}+\dot{E}^{T}_{x,51}+\dot{E}^{T}_{x,63}-\dot{E}^{T}_{x,26}+\dot{E}^{T}_{x,221}-\dot{E}^{T}_{x,222}\right)C_T+\left(\dot{E}^{P}_{x,1}+\dot{E}^{P}_{x,51}+\dot{E}^{P}_{x,63}-\dot{E}^{P}_{x,26}+\dot{E}^{P}_{x,221}-\dot{E}^{P}_{x,222}\right)C_P$$
$$+T_o\left(\dot{S}_1+\dot{S}_{51}+\dot{S}_{63}-\dot{S}_{26}+\dot{S}_{221}-\dot{S}_{222}+\dot{Q}_{[boun]}/T_o\right)\mathbf{C_S}+\dot{Z}_{[boun]}=0 \tag{24}$$

In **Table 3**, initial investments, the annuities including the maintenance cost, and the corresponding monetary flow rates for each component are given. The cost flow rates corresponding to a component's exergy flow rates at 100% load condition are given in **Table 4**. The same sign convention for the cost flow rates related to products and resources was used as the case of exergy balances shown in **Table 2**. The lost cost due to the entropy production in a component is consumed cost. The fact that the sum of the cost flow rates of each component in the plant becomes zero, as verified in **Table 4**, shows that all the cost balances for the components are satisfied.

The overall cost-balance equation for the power system is simply obtained by summing Eqs. (17)–(24).

$$\dot{E}^{CHE}_{x}C_o+\sum_{i=1}^{n}\dot{Z}_{[i]}=\left(\dot{E}^{W}_{x,[1]}+\dot{E}^{W}_{x,[3]}+\dot{E}^{W}_{x,[6]}\right)C_W \tag{25}$$

From the above equation, the unit cost of electricity for the gas-turbine power system is given as [1]

$$C_W=\frac{C_o}{\eta_e}\left[1+\frac{\sum Z_{[i]}}{C_o\dot{E}^{CHE}_{x}}\right] \tag{26}$$

The production cost depends on fuel cost and the exergetic efficiency of the system, and is affected by the ratio of the monetary flow rate of non-fuel items to the monetary flow rate of fuel. With the unit cost of fuel, C_o = 5.0 \$/GJ, an exergetic efficiency of the gas-turbine power

Component	Initial investment cost (US\$$10^6$)	Annualized cost (×US\$$10^3$/year)	Monetary flow rate (US\$/h)
Compressor	36.976	4744.997	628.712
Combustor	2.169	278.340	36.880
Gas turbine	29.213	3748.799	496.716
Fuel preheater	7.487	960.780	127.303
Steam injector	14.787	1897.562	251.427
Total	90.542	11,630.478	1531.038

Table 3. Initial investments, annualized costs, and corresponding monetary flow rates of each component in the gas-turbine power plant.

Component	$\dot{C}_W$ (US$/h)	$\dot{C}_o$ (US$/h)	$\dot{C}_T$ (US$/h)	$\dot{C}_P$ (US$/h)	$\dot{C}_S$ (US$/h)	$\dot{Z}$ (US$/h)
Compressor	−17732.47	0.00	4217.91	15,071.00	−927.63	−628.71
Combustor	0.00	−15861.96	28238.85	341.28	−12681.19	−36.88
Gas turbine	38419.49	0.00	−21068.79	−16066.19	−787.79	−496.72
Fuel preheater	0.00	0.00	154.92	0.00	−27.52	−127.30
Steam injector	−1208.41	0.00	569.65	954.76	−64.44	−251.43
Boundary	0.00	0.00	−12112.54	−300.85	14488.57	−2075.18
Total	19478.61	−15861.96	0.00	0.00	0.00	−3616.22

Table 4. Cost flow rates of various exergies and neg-entropy of each component in the gas-turbine power plant at 100% load condition.

plant, 0.341, and a value of the ratio of the monetary flow rate of non-fuel items to the monetary flow rate of fuel, 0.22, the unit cost of electricity estimated from Eq. (26) is approximately 17.97 $/GJ. However, one should solve Eqs. (17)–(24) simultaneously to obtain the unit cost of electricity and the lost cost flow rate occurred in each component.

4. Organic Rankine cycle power plant using heat as fuel

A schematic of the 20-kW ocean thermal energy conversion (OTEC) plant [20] operated by organic Rankine cycle, which is considered to apply MOPSA method, is illustrated in **Figure 2**. Five main components exist in the system: the evaporator (1), turbine (2), condenser (3), receiver tank (4) and pump (5). The refrigerant stream is heated by a heat source in the evaporator, and then the refrigerant stream is divided into two streams. A portion of this stream is passed through the throttling valve and reaches the receiver tank, while the remaining part of the refrigerant stream leaving from evaporator is sent to turbine. A portion of the stream flowing to turbine is throttled and bypassed to turbine outlet. The “pipes” are introduced into the analysis as a component to consider the heat and pressure losses in the pipes and the exergy removal during the throttling processes. Refrigerant of R32 is used as a working fluid in the organic Rankine cycle. At the full load condition, the mass flow rate of the refrigerant is 3.62 kg/s. The warm sea water having mass flow rate of 86.99 kg/s is used as a heat source for the plant, while the cold sea water having mass flow rate of 44.85 kg/s is used as a heat sink for the plant. The reference temperature and pressure for the refrigerant R32 are −40°C and 177.60 kPa, respectively. For water, the reference point was taken as 0.01°C, the triple point of water.

4.1. Exergy-balance equations for the organic Rankine cycle power plant

The exergy-balance equations obtained using Eq. (1) for each component in the organic Rankine cycle plant shown in **Figure 2** are as follows.

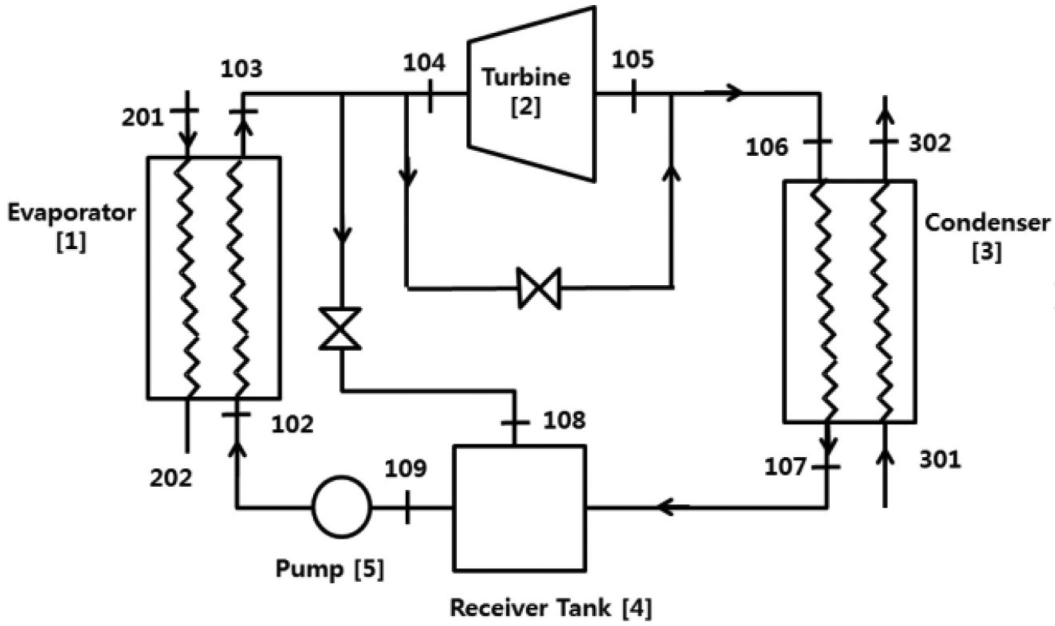

Figure 2. Schematic of an organic Rankine cycle power plant using warm water as a fuel.

Evaporator

$$\left(\dot{E}^{T}_{x,102} - \dot{E}^{T}_{x,103}\right) + \left(\dot{E}^{P}_{x,102} - \dot{E}^{P}_{x,103}\right) + \left(\dot{E}_{x,201} - \dot{E}_{x,202}\right) + T_o\left[\left(\dot{S}_{102} - \dot{S}_{103}\right) + \left(\dot{S}_{201} - \dot{S}_{202}\right) + \dot{Q}_{[1]}/T_o\right] = 0 \tag{27}$$

Turbine

$$\left(\dot{E}^{T}_{x,104} - \dot{E}^{T}_{x,105}\right) + \left(\dot{E}^{P}_{x,104} - \dot{E}^{P}_{x,105}\right) + T_o\left(\dot{S}_{104} - \dot{S}_{105} + \dot{Q}_{[2]}/T_o\right) = \dot{E}^{W}_{x,[2]} \tag{28}$$

Condenser

$$\left(\dot{E}^{T}_{x,106} - \dot{E}^{T}_{x,107}\right) + \left(\dot{E}^{P}_{x,106} - \dot{E}^{P}_{x,107}\right) + \left(\dot{E}_{x,301} - \dot{E}_{x,302}\right) + T_o\left[\left(\dot{S}_{106} - \dot{S}_{107}\right) + \left(\dot{S}_{301} - \dot{S}_{302}\right) + \dot{Q}_{[3]}/T_o\right] = 0 \tag{29}$$

Receiver tank

$$\left(\dot{E}^{T}_{x,107} + \dot{E}^{T}_{x,108} - \dot{E}^{T}_{x,109}\right) + \left(\dot{E}^{P}_{x,107} + \dot{E}^{T}_{x,108} - \dot{E}^{P}_{x,109}\right) + T_o\left(\dot{S}_{107} + \dot{S}_{108} - \dot{S}_{109} + \dot{Q}_{[4]}/T_o\right) = 0 \tag{30}$$

Pump

$$\left(\dot{E}^{T}_{x,109} - \dot{E}^{T}_{x,102}\right) + \left(\dot{E}^{P}_{x,109} - \dot{E}^{P}_{x,102}\right) + T_o\left(\dot{S}_{109} - \dot{S}_{102} + \dot{Q}_{[5]}/T_o\right) = \dot{E}^{W}_{x,[5]} \tag{31}$$

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Pipes

$$\begin{aligned}&\left(\alpha\dot{E}^{T}_{x,103}-\dot{E}^{T}_{x,104}\right)+\left(\dot{E}^{T}_{x,105}-\dot{E}^{T}_{x,106}\right)+\left((1-\alpha)\dot{E}^{T}_{x,103}-\dot{E}^{T}_{x,108}\right)\\&\quad+\left(\alpha\dot{E}^{P}_{x,103}-\dot{E}^{P}_{x,104}\right)+\left(\dot{E}^{P}_{x,105}-\dot{E}^{P}_{x,106}\right)+\left((1-\alpha)\dot{E}^{P}_{x,103}-\dot{E}^{P}_{x,108}\right)\\&\quad+T_o\left(\dot{S}_{103}+\dot{S}_{105}-\dot{S}_{104}-\dot{S}_{106}-\dot{S}_{108}+\dot{Q}_{pipes}/T_o\right)=0\end{aligned}\tag{32}$$

Boundary

$$\begin{aligned}&-\left(\dot{E}_{x,201}-\dot{E}_{x,202}\right)-\left(\dot{E}_{x,301}-\dot{E}_{x,302}\right)\\&-T_o\left(\dot{S}_{201}-\dot{S}_{202}+\dot{S}_{301}-\dot{S}_{302}+\dot{Q}_{boun}/T_o\right)=0\end{aligned}\tag{33}$$

The α term given in Eq. (32) is the ratio of the bypass streams from state 103 to 108. The value of the α term can be calculated by applying the mass and energy conservation equations to the receiver tank. The stream bypassed from state 103 to 105 may be neglected. An example of exergy calculation for the organic Rankine cycle plant using a stream of warm water at 28°C as a heat source to the evaporator [20] is shown in **Table 5**. As mentioned in the previous section, a positive value of exergy flow rate represents “product,” while a negative value of exergy flow rate indicates “fuel.” The last two columns clearly indicate that the electricity comes from expenditure of heat input.

4.2. Cost-balance equations for the organic Rankine cycle power plant

By assigning a unit cost to every thermal exergy of the refrigerant stream (C_{1T}, C_{2T}, C_{3T}, and C_T), mechanical exergy for the refrigerant stream (C_P), cold water (C_3), neg-entropy (C_s), and electricity (C_W), the cost-balance equations corresponding to the exergy-balance equations which are Eqs. (27)–(33) are given as follows. When the cost-balance equation is applied to a specific component, one may assign a unit cost to its main product, which is represented by a Gothic letter.

Component	Refrigerant	Water stream	Irreversibility rate	Heat transfer rate	Work input/output rate
Evaporator	224.59	−233.21	17.52	−8.90	—
Turbine	−24.24	—	3.31	0.83	20.10
Condenser	−178.00	171.26	5.22	1.51	—
Receiver tank	−2.52	—	−11.68	14.20	—
Pump	1.50	—	1.69	−0.15	−3.04
Pipes	−21.33	—	20.31	1.02	—
Boundary	—	61.95	−36.36	−25.58	—
Total	0.00	0.00	0.00	−17.06	17.06

Table 5. Exergy balances for each component in the organic Rankine cycle plant (Unit: kW) [20].

Evaporator

$$\left(\dot{E}^{T}_{x,102}-\dot{E}^{T}_{x,103}\right)\mathbf{C_{1T}}+\left(\dot{E}^{P}_{x,102}-\dot{E}^{P}_{x,103}\right)C_P+\left(\dot{E}_{x,201}-\dot{E}_{x,202}\right)C_2 \\ +T_o\left[\left(\dot{S}_{102}-\dot{S}_{103}\right)+\left(\dot{S}_{201}-\dot{S}_{202}\right)+\dot{Q}_{[1]}/T_o\right]C_S+\dot{Z}_{[1]}=0 \tag{34}$$

Turbine

$$\left(\dot{E}^{T}_{x,104}-\dot{E}^{T}_{x,105}\right)C_T+\left(\dot{E}^{P}_{x,104}-\dot{E}^{P}_{x,105}\right)C_P \\ +T_o\left(\dot{S}_{104}-\dot{S}_{105}+\dot{Q}_{[2]}/T_o\right)C_S+\dot{Z}_{[2]}=\dot{E}^{W}_{x,[2]}\mathbf{C_W} \tag{35}$$

Condenser

$$\left(\dot{E}^{T}_{x,106}-\dot{E}^{T}_{x,107}\right)C_T+\left(\dot{E}^{P}_{x,106}-\dot{E}^{P}_{x,107}\right)C_P+\left(\dot{E}_{x,301}-\dot{E}_{x,302}\right)\mathbf{C_3} \\ +T_o\left[\left(\dot{S}_{106}-\dot{S}_{107}\right)+\left(\dot{S}_{301}-\dot{S}_{302}\right)+\dot{Q}_{[3]}/T_o\right]C_S+\dot{Z}_{[3]}=0 \tag{36}$$

Receiver tank

$$\left(\dot{E}^{T}_{x,107}+\dot{E}^{T}_{x,108}-\dot{E}^{T}_{x,109}\right)\mathbf{C_{2T}}+\left(\dot{E}^{P}_{x,107}+\dot{E}^{P}_{x,108}-\dot{E}^{P}_{x,109}\right)C_P \\ +T_o\left(\dot{S}_{107}+\dot{S}_{108}-\dot{S}_{109}+\dot{Q}_{[4]}/T_o\right)C_S+\dot{Z}_{[4]}=0 \tag{37}$$

Pump

$$\left(\dot{E}^{T}_{x,109}-\dot{E}^{T}_{x,102}\right)C_T+\left(\dot{E}^{P}_{x,109}-\dot{E}^{P}_{x,102}\right)\mathbf{C_P} \\ +T_o\left(\dot{S}_{109}-\dot{S}_{102}+\dot{Q}_{[5]}/T_o\right)C_S+\dot{Z}_{[5]}=\dot{E}^{W}_{x,[5]}C_W \tag{38}$$

Pipes

$$\left[\left(\alpha\dot{E}^{T}_{x,103}-\dot{E}^{T}_{x,104}\right)+\left(\dot{E}^{T}_{x,105}-\dot{E}^{T}_{x,106}\right)+\left((1-\alpha)\dot{E}^{T}_{x,103}-\dot{E}^{T}_{x,108}\right)\right]\mathbf{C_{3T}} \\ +\left[\left(\alpha\dot{E}^{P}_{x,103}-\dot{E}^{P}_{x,104}\right)+\left(\dot{E}^{P}_{x,105}-\dot{E}^{P}_{x,106}\right)+\left((1-\alpha)\dot{E}^{P}_{x,103}-\dot{E}^{P}_{x,108}\right)\right]C_P \\ +T_o\left(\dot{S}_{103}+\dot{S}_{105}-\dot{S}_{104}-\dot{S}_{106}-\dot{S}_{108}+\dot{Q}_{pipes}/T_o\right)C_s+\dot{Z}_{pipes}=0 \tag{39}$$

Boundary

$$-\left(\dot{E}_{x,201}-\dot{E}_{x,202}\right)C_2-\left(\dot{E}_{x,301}-\dot{E}_{x,302}\right)C_3 \\ -T_o\left(\dot{S}_{201}-\dot{S}_{202}+\dot{S}_{301}-\dot{S}_{302}+\dot{Q}_{boun}/T_o\right)\mathbf{C_S}+\dot{Z}_{boun}=0 \tag{40}$$

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Seven cost-balance equations for the five components of the plant, pipes, and the boundary were derived with eight unknown unit exergy costs of C_{1T}, C_{2T}, C_{3T}, C_T, C_P, C_3, C_S, and C_W. We can obtain an additional cost-balance equation for the junction of thermal exergy of the refrigerant stream.

Thermal junction

$$\begin{aligned}\left(\dot{E}^T_{x,102}-\dot{E}^T_{x,103}\right)C_{1T}+\left(\dot{E}^T_{x,107}+\dot{E}^T_{x,108}-\dot{E}^T_{x,109}\right)C_{2T}\\+\left(\dot{E}^T_{x,103}+\dot{E}^T_{x,105}-\dot{E}^T_{x,104}-\dot{E}^T_{x,106}-\dot{E}^T_{x,108}\right)C_{3T}\\=\left[\dot{E}^T_{x,102}+\dot{E}^T_{x,105}+\dot{E}^T_{x,107}-\dot{E}^T_{x,104}-\dot{E}^T_{x,106}-\dot{E}^T_{x,109}\right]\mathbf{C_T}\end{aligned} \tag{41}$$

With Eq. (41), we have all the necessary cost-balance equations to calculate the unit cost of all exergies (C_{1T}, C_{2T}, C_{3T}, C_T, and C_3, neg-entropy (C_s) and a product (electricity, C_W) by input (given) of thermal energy (C_2) to the evaporator. The overall cost-balance equation for the Rankine power plant can be obtained by summing Eqs. (34)–(41), which is given by

$$abs\left[\sum\dot{C}_H+\left(\sum\dot{Z}_k+\dot{Z}_{boun}\right)\right]=\dot{E}^W_x C_W \tag{42}$$

where $\sum\dot{C}_H=\sum\dot{Q}_k C_S$ is the net cost flow rate due to the heat transfer to/from the organic Rankine cycle plant. The term $\dot{Z}_{boun}$ in Eq. (42) may represent the cost flow rate related to the construction of the plant [6]. Rewriting Eq. (42), we have the unit cost of electricity from the Rankine cycle power plant.

$$C_W=abs\left[\sum\dot{C}_H+\left(\sum\dot{Z}_k+\dot{Z}_{boun}\right)\right]/\dot{E}^W_x \tag{43}$$

where $\dot{E}^W_x$ is the net electricity obtained from the organic Rankine cycle plant and abs denotes the absolute value of the quantity in parentheses.

Figure 3 shows that the unit cost of electricity from the organic Rankine cycle plant and the net cost flow rate due to the heat transfer rate to the plant vary depending on the unit cost of warm water in the evaporator, C_2, appeared in Eq. (34). As the unit cost of warm water increases, the net cost flow rate due to heat transfer to the plant decreases while the unit cost of electricity increases. The cross point between the line for the unit cost of electricity and the line for the total cost flow rate due to heat transfer determines unit cost of electricity. The unit cost of electricity and the net cost flow rate due to heat transfer for a case whose detailed calculation results shown in **Table 6** are \$0.205 and −\$0.941/kWh, respectively. The value of the unit cost C_2 appeared in the cost balance equation, Eq. (34), is approximately \$0.117/kWh for this particular case, which may be considered as a fictional one.

Detailed calculation results reveal that the unit cost of electricity from an organic Rankine cycle plant can be obtained from the following equation:

$$C_W=C_2'+\left(\sum\dot{Z}_k+\dot{Z}_{boun}\right)/\dot{E}^W_x \tag{44}$$

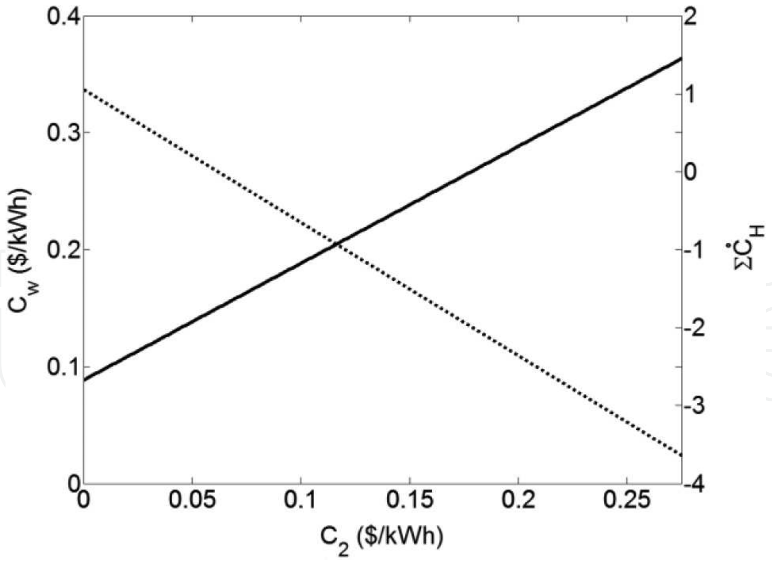

Figure 3. Unit cost of electricity, C_W (solid lines), and net cost flow rate due to heat transfer to the plant, $\sum \dot{C}_H$ (dotted line), depending on the unit cost of supplied hot water to evaporator C_2, for the case shown in **Table 6**.

Component	$\dot{C}_T$	$\dot{C}_P$	$\dot{C}_S$	$\dot{C}_H$	$\dot{C}_W$	$\dot{C}_{wsw}$	$\dot{C}_{dsw}$	$\dot{Z}_k$
Evaporator	27.502	−0.026	0.967	−0.491	—	−27.285	—	−0.666
Turbine	−3.101	−0.613	0.183	0.046	4.126	—	—	−0.640
Condenser	−23.569	−0.004	0.288	0.084	—	—	23.867	−0.666
Receiver tank	0.021	0.012	−0.645	0.784	—	—	—	−0.172
Pump	0.079	0.678	0.093	−0.008	−0.624	—	—	−0.218
Pipes	−0.932	−0.048	1.121	0.056	—	—	—	−0.198
Boundary	—	—	−2.006	−1.412	—	27.285	−23.867	—
Total	−0.000	0.000	−0.000	−0.941	3.502	—	—	−2.561

Using hot water from an incinerator plant as the heat source, C_2 =$0.117/kWh, C_2' =$0.055/kWh [20].
Solutions of cost-balance equations [Unit:$/kWh].
C_{1T} = 0.122, C_{2T} = −0.008, C_{3T} = 0.044, C_T = 0.132, C_P = 0.750, C_3 = 0.139, C_W = 0.205, C_S = 0.055.

Table 6. Cost flow rates of various exergies, lost work rate due to heat transfer, heat transfer rate, and work input/out rate of each component in the organic Rankine cycle plant (Unit: $/h).

From Eqs. (43) and (44), one can deduce that

$$C_2' = abs\left(\sum \dot{C}_H\right)/\dot{E}_x^W \tag{45}$$

The calculated value of C_2' using Eq. (45) is approximately $0.055/kWh, which is quite different from the C_2 = $0.177/kWh, a value determined from **Figure 3**. The value of C_2' which is the ratio of the absolute value of net cost flow rate of heat to the produced electricity was found to be a real unit cost of hot water stream [20]. Equation (44) tells us that the unit cost of electricity from

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the organic Rankine cycle is determined by the sum of the unit cost of heat and the ratio of the monetary flow rate of non-fuel items to the produced electric power.

5. 200 kW air-cooled air conditioning unit

Even though the performance evaluation of a household refrigerator using thermoeconomics was performed [21], estimation of the unit cost of heat supplied to the room by air conditioning unit was never tried. In this section, the unit cost of heat for a 120-kW air-cooled air conditioning unit is obtained, which is helpful for the cost comparison between air conditioning unit operated by electricity and absorption refrigeration system running by heat [22].

5.1. Exergy-balance equations for the air conditioning units

The exergy-balance equations obtained using Eq. (1) for each component in an air-cooled air conditioning units shown in **Figure 4** are as follows. The heat transfer interactions with environment for the compressor, TXV, and suction line are neglected.

Compressor

$$\left(\dot{E}^{r,T}_{x,1} - \dot{E}^{r,T}_{x,2}\right) + \left(\dot{E}^{r,P}_{x,1} - \dot{E}^{r,P}_{x,2}\right) + T_o\left(\dot{S}^r_1 - \dot{S}^r_2\right) = E^W_{x,comp} \quad (46)$$

Condenser

$$\left(\dot{E}^{r,T}_{x,2} - \dot{E}^{r,T}_{x,3}\right) + \left(\dot{E}^{r,P}_{x,2} - \dot{E}^{r,P}_{x,3}\right) + \left(\dot{E}^{a}_{x,7} - \dot{E}^{a}_{x,8}\right) + T_o\left(\dot{S}^r_2 - \dot{S}^r_3 + \dot{S}^a_7 - \dot{S}^a_8 + \dot{Q}_{con}/T_o\right) = 0 \quad (47)$$

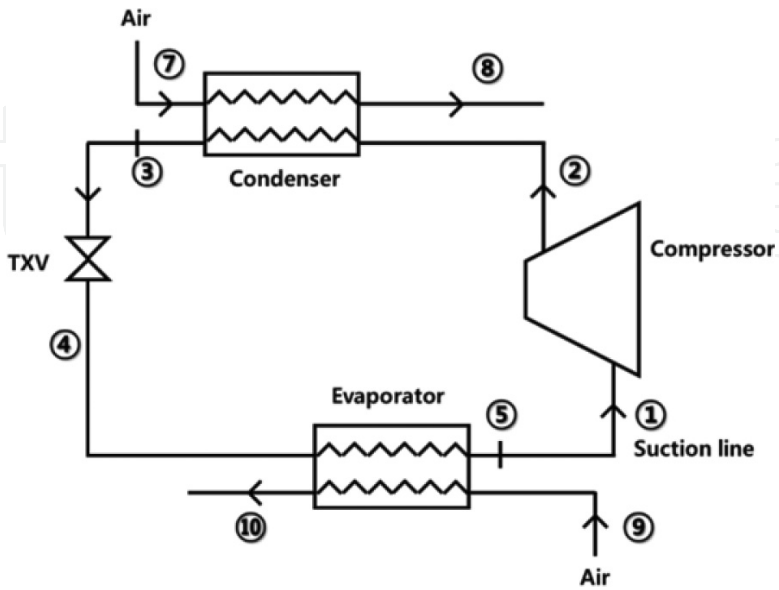

Figure 4. Schematic of a 120-kW air-cooled air conditioning system.

TXV

$$\left(\dot{E}^{r,T}_{x,3} - \dot{E}^{r,T}_{x,4}\right) + \left(\dot{E}^{r,P}_{x,3} - \dot{E}^{r,P}_{x,4}\right) + T_o\left(\dot{S}^{r}_{3} - \dot{S}^{r}_{4}\right) = 0 \tag{48}$$

Evaporator

$$\left(\dot{E}^{r,T}_{x,4} - \dot{E}^{r,T}_{x,5}\right) + \left(\dot{E}^{r,P}_{x,4} - \dot{E}^{r,P}_{x,5}\right) + \left(\dot{E}^{a}_{x,9} - \dot{E}^{a}_{x,10}\right) + T_o\left(\dot{S}^{r}_{4} - \dot{S}^{r}_{5} + \dot{S}^{a}_{9} - \dot{S}^{a}_{10} + \dot{Q}_{evap}/T_o\right) = 0 \tag{49}$$

Suction line

$$\left(\dot{E}^{r,T}_{x,5} - \dot{E}^{r,T}_{x,1}\right) + \left(\dot{E}^{r,P}_{x,5} - \dot{E}^{r,P}_{x,1}\right) + T_o\left(\dot{S}^{r}_{5} - \dot{S}^{r}_{1}\right) = 0 \tag{50}$$

Superscripts r and a given in the above equations represent the fluid stream of the refrigerant and air, respectively, and W denotes work. The amount of heat transferred to the environment in each component was neglected in the exergy-balance equations.

In Eqs. (47) and (49), the difference in the exergy and entropy for air stream is just the difference in the enthalpy so that these terms can be written with help of Eq. (2) as

$$\left(\dot{E}^{a}_{x,7} - \dot{E}^{a}_{x,8}\right) + T_o\left(\dot{S}^{a}_{7} - \dot{S}^{a}_{8}\right) = \left(\dot{H}^{a}_{7} - \dot{H}^{a}_{8}\right) = -\dot{Q}^{H}_{env} \tag{51}$$

$$\dot{E}^{a}_{x,9} - \dot{E}^{a}_{x,10} + T_o\left(\dot{S}^{a}_{9} - \dot{S}^{a}_{10}\right) = \left(\dot{H}^{a}_{9} - \dot{H}^{a}_{10}\right) = \dot{Q}^{H}_{room} \tag{52}$$

The deposition of heat into the environment and the heat transferred to room are hardly considered to be dissipated to the environment. For such heat delivery system, it may be reasonable that the delivered heat rather than its exergy is contained in the exergy-balance equation. With help of Eqs. (51) and (52), the exergy-balance equation for the condenser and evaporator become

$$\left[\left(\dot{E}^{r,T}_{x,2} - \dot{E}^{r,T}_{x,3}\right) + \left(\dot{E}^{r,P}_{x,2} - \dot{E}^{r,P}_{x,3}\right) + T_o\left(\dot{S}^{r}_{2} - \dot{S}^{r}_{3}\right)\right] + \left[-\dot{Q}^{H}_{env} + \dot{Q}_{con}\right] = 0 \tag{47′}$$

$$\left[\left(\dot{E}^{r,T}_{x,4} - \dot{E}^{r,T}_{x,5}\right) + \left(\dot{E}^{r,P}_{x,4} - \dot{E}^{r,P}_{x,5}\right) + T_o\left(\dot{S}^{r}_{4} - \dot{S}^{r}_{5}\right)\right] + \left[\dot{Q}^{H}_{room} + \dot{Q}_{evap}\right] = 0 \tag{49′}$$

The terms, $\dot{Q}_{con}$ in Eq. (47′) and $\dot{Q}_{evap}$ in Eq. (49′) represent the irreversibility corresponding to the terms $-\dot{Q}^{H}_{env}$ and $\dot{Q}^{H}_{room}$, respectively. The pair terms in the second bracket in Eqs. (47′) and (49′) are equal in magnitude but opposite in sign to vanish completely because the terms in the first bracket in those equations vanish. This assumption is legitimate since the entropy generation due to the heat transfer between flow streams [23] in the condenser and evaporator was calculated to be negligibly small.

The simulated data for the difference in the thermal and mechanical exergy flow rates at each component under normal operation for a 120-kW air-cooled air conditioning system [24] is displayed in **Table 7**. The cooling capacity of the system ($\dot{Q}^{H}_{room}$) is considered as the heat

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Component	$\Delta\dot{E}_x^{T,r}$	$\Delta\dot{E}_x^{P,r}$	$\dot{Q}_H$	$\dot{E}_x^W$	$\dot{I}$
Compressor	0.18	22.85		−32.10	9.07
Condenser	−8.95	−0.09	(−88.92)		(88.92) 9.04
TXV	18.29	−21.97			3.68
Evaporator	−9.52	−0.66	(121.02)		(−121.02) 10.18
Suction line		−0.13			0.13
Total	0.0	0.0	32.10	−32.10	0.0

The numerical values in parentheses are the heat flow rate of air (third column) and the corresponding lost work rate (fifth column).

Table 7. Exergy balances for each component in the 120-kW air conditioning system at normal operation (Unit: kW).

gained by the refrigerant in the evaporator. However, the heat output to the environment through the condenser was taken to satisfy the exergy-balance equation for the condenser as well as the overall system. The irreversibility rate due to the entropy generation at each component was calculated using the exergy-balance equations for each component given from Eq. (46) to (50). The values in the parentheses in the third and fifth columns represent the first and second quantity inside the second bracket in Eqs. (47′) and (49′), respectively. Note that minus and plus sign indicate the resource or fuel and product exergies, respectively, as usual. The sign of the irreversibility rate is minus at the evaporator, while it is plus at other units which play as boundary.

5.2. Cost-balance equations for the air-cooled air conditioning units

By assigning a unit cost to every thermal and mechanical exergy stream of the refrigerant (C_T, C_P), lost work (C_S), heat (C_H), and work (C_W), the cost-balance equations corresponding to the exergy-balance equations, i.e., Eqs. (46), (47′), (48), (49′), and (50), are as follows. In this particular thermal system, a unit to a principal product for each component is not applied because the working fluid that flows through all the components makes a thermodynamic cycle.

Compressor

$$\left(\dot{E}_{x,1}^{r,T} - \dot{E}_{x,2}^{r,T}\right)C_T + \left(\dot{E}_{x,1}^{r,P} - \dot{E}_{x,2}^{r,P}\right)C_P + T_o\left(\dot{S}_1^r - \dot{S}_2^r\right)C_S + \dot{Z}_{comp} = \dot{E}_{x,comp}^{W}C_W \quad (53)$$

Condenser

$$\left(\dot{E}_{x,2}^{r,T} - \dot{E}_{x,3}^{r,T}\right)C_{1T} + \left(\dot{E}_{x,2}^{r,P} - \dot{E}_{x,3}^{r,P}\right)C_P - \dot{Q}_{env}^{H}\cdot 0 + \left[T_o\left(\dot{S}_2^r - \dot{S}_3^r\right) + \dot{Q}_{con}\right)]C_S + \dot{Z}_{con} = 0 \quad (54)$$

TXV

$$\left(\dot{E}_{x,3}^{r,T} - \dot{E}_{x,4}^{r,T}\right)C_T + \left(\dot{E}_{x,3}^{r,P} - \dot{E}_{x,4}^{r,P}\right)C_P + T_o\left(\dot{S}_3^r - \dot{S}_4^r\right)C_S = 0 \quad (55)$$

Evaporator

$$\left(\dot{E}_{x,4}^{r,T}-\dot{E}_{x,5}^{r,T}\right)C_T+\left(\dot{E}_{x,4}^{r,P}-\dot{E}_{x,5}^{r,P}\right)C_P+\dot{Q}_{room}^{H}\mathrm{C_H}+\left[T_o\left(\dot{S}_4^r-\dot{S}_5^r\right)+\dot{Q}_{evap}\right]C_S+\dot{Z}_{eva}=0 \quad (56)$$

Suction line

$$\left(\dot{E}_{x,5}^{P,r}-\dot{E}_{x,1}^{P,r}\right)C_P+T_o\left[\dot{S}_5^r-\dot{S}_1^r\right]C_s+\dot{Z}_{sl}=0 \quad (57)$$

We now have five cost-balance equations to calculate two unit costs of exergies (C_T and C_P), neg-entropy (C_S), and a product, heat (C_H) by input of electricity (C_W). So, it is better to combine the cost-balance equation for the evaporator and suction line, which can be written as

$$\left(\dot{E}_{x,4}^{r,T}-\dot{E}_{x,1}^{r,T}\right)C_T+\left(\dot{E}_{x,4}^{r,P}-\dot{E}_{x,1}^{r,P}\right)C_P+\dot{Q}_{room}^{H}C_H+\left[T_o\left(\dot{S}_4^r-\dot{S}_1^r\right)+\dot{Q}_{evap}\right]C_S$$

$$+\left(\dot{Z}_{evap}+\dot{Z}_{sl}\right)=0 \quad (58)$$

The overall cost-balance equation for the air conditioning units can be obtained by summing Eqs. (53)–(55) and (58);

$$\dot{Q}_{room}^{H}C_H=\sum\dot{Z}_k+\dot{E}_x^{W}C_W \quad (59)$$

Table 8 lists the initial investment, the annuities including the maintenance cost, and the corresponding monetary flow rates for each component of the air-cooled air conditioning system. Currently, the installation cost of an air-cooled air conditioning system with a 120-kW cooling capacity is approximately \$17,000 in Korea. The levelized cost of the air conditioning units was calculated to be 0.3122\$/h with an expected life of 20 years, an interest rate of 5% and salvage value of \$850. The operating hours of the air conditioning system, which is crucial in determining the levelized cost, were taken as 4500 h. The maintenance cost was taken as 5% of the annual levelized cost of the system.

The cost flow rates of various exergies and irreversibility rate at each component in the air conditioning system at the normal operation are shown in **Table 9**. The sign convention for the cost flow rates is that minus and plus signs indicate the resource and product cost flow rates, respectively. Erroneously, reverse sign convention was used in their study on the thermoeconomic

Component	Initial investment (\$)	Annualized cost (\$/year)	Monetary flow rate (\$/h)
Compressor	5000	393.4	0.0918
Condenser	4000	314.8	0.0735
TXV	2000	157.4	0.0367
Evaporator + Suction line	6000	472.1	0.1102
Total	17,000	1337.7	0.3122

Table 8. Initial investments, annualized costs, and corresponding monetary flow rates of each component in air conditioning system with a 120-kW capacity.

Component	$\dot{C}_T$	$\dot{C}_P$	$\dot{C}_H$	$\dot{C}_W$	$\dot{C}_S$	$\dot{Z}$
Compressor	0.03506	3.73914		−3.8520	0.16960	−0.09180
Condenser	−1.74352	−0.01473			1.83175	−0.07350
TXV	3.56302	−3.59513			0.06881	−0.03670
Evaporator+ Suction line	−1.85456	−0.12928	4.16420		−2.07016	−0.11020
Total	0.0	0.0	4.16420	−3.8520	0.0	−0.3122
The unit cost of irreversibility, C_S is 0.00187 $/kWh and the unit cost of cooling capacity, C_H is 0.0344$/kWh						

Table 9. Cost flow rates of various exergies and irreversibility of each component in the air conditioning unit at normal operation (Unit: $/h).

analysis of ground-source heat pump systems [25]. The lost cost flow rate due to the entropy generation appears as consumed cost in the evaporator; on the other hand, it appears as production cost in other components. The unit cost of heat delivered to the room or the unit cost of the cooling capacity is estimated to be 0.0344$/kWh by solving the four cost-balance equations given from Eqs. (53) to (59) with unit cost of electricity of 0.120 $/kWh. The unit cost of thermal and mechanical exergies and the irreversibility are C_T = 0.1948, C_P = 0.1636, and C_S = 0.0187 $/kWh at the normal operation. It is noted that the unit cost of heat C_H can be obtained from Eq. (60) directly with known values of C_W, COP (β) and the ratio of the monetary flow rate of non-fuel items to the monetary flow rate of input (electricity). **Table 9** confirms that cost-balance balance is satisfied for all components and the overall system.

Rewriting Eq. (59), we have [25]

$$C_H = \frac{C_W}{\beta}\left[1 + \frac{\sum \dot{Z}_k}{C_W \dot{E}^{W}_{x,comp}}\right] \tag{60}$$

where β is the COP of the air conditioning units. Equation (60) provides the unit cost of cooling capacity as 0.0344 $/kWh with a unit cost of electricity of 0.120 $/kWh, β of 3.77, and a value of 0.081 for the ratio of the monetary flow rate of non-fuel items to the monetary flow rate of consumed electricity.

6. Conclusions

Explicit equations to obtain the unit cost of products from gas-turbine power plant and organic Rankin cycle plant operating by heat source as fuel and the unit cost heat for refrigeration system using the modified-productive structure analysis (MOPSA) method were obtained. MOPSA method provides two basic equations for exergy-costing method: one is a general exergy-balance equation and the other is cost-balance equation, which can be applicable to any components in power plant or refrigeration system. Exergy-balance equations can be obtained for each component and junction. The cost-balance equation corresponding to the exergy-balance equation can be obtained by assigning a unit cost to the principal product of each component. The overall exergy-costing equation to estimate the unit cost of product from the power plant and refrigeration system is obtained by summing up all the cost-balance equations for each component, junctions, and boundary of the system. However, one should solve the cost-balance equations for the components, junctions, and system boundary simultaneously to obtain the lost cost flow rate due to the entropy generation in each component. It should be noted that the lost work rate due to the entropy generation plays as "product" in the exergy-balance of the component, while the lost cost flow rate plays as "consumed resources" in the cost-balance equation. This concept is very important in the research area of thermoeconomic diagnosis [18, 26–28].

Nomenclature

C	unit cost of exergy ($/kJ)
C_i	initial investment cost ($)
C_H	unit cost of heat ($/kWh)
C_o	unit cost of fuel ($/kWh)
C_S	unit cost of lost work due to the entropy generation ($/kWh)
C_W	unit cost of electricity ($/kWh)
$\dot{C}$	monetary flow rate ($/h)
COP	coefficient of performance
CRF	capital recovery factor
e_x	exergy per mass
$\dot{E}_x$	exergy flow rate (kW)
h	enthalpy per mass
$\dot{H}$	enthalpy flow rate (kW)
i	interest rate
$\dot{I}$	irreversibility rate (kW)
$\dot{m}$	mass flow rate
PW	amortization cost
PWF(i,n)	present worth factor
$\dot{Q}_{cv}$	heat transfer rate (kW)
$\dot{S}$	entropy flow rate (kW/K)

Sn	salvage value (KRW)
T_o	ambient temperature (°C)
$\dot{W}_{cv}$	work production rate (kW)
$\dot{Z}_k$	capital cost flow rate of unit k ($/h)

Greek symbols

β	coefficient of performance
δ	operating hours
η_e	exergy efficiency
ϕ_k	maintenance factor of unit k

Subscripts

a	air stream
comp	compressor
con	condenser
env	environment
evap	evaporator
H	heat
k	kth component
r	refrigerant stream
ref.	reference condition
room	room
s	entropy
sl	suction line
W	work or electricity

Superscripts

a	air stream

CHE chemical exergy

H heat

P mechanical exergy

r refrigerant stream

T thermal exergy

W work or electricity

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