Theoretical Study of the Absorption Refrigeration Cycle Using Water-Lithium Bromide as Working Pair for Cold Storage Application

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Abstract. This study aims to investigate the performance of a cooling absorption of the water-lithium bromide cycle for a cold storage application which has a capacity of five tonnes of fish. The cooling absorption system utilizes the waste heat of a 6 MW steam power plant as the heat source for the generator. The thermodynamics analysis is carried out to determine both the heat rejection rate of the power plant and the performance of the refrigeration absorption cycle. The performance of the cooling system is investigated by varying the temperature of each component of the refrigeration absorption cycle (i.e generator, absorber, condenser, and evaporator). Also, the influence of temperature on the heat input (to the generator), heat output (from the absorber), and the pump power, is examined. The results show that the Coefficient of Performance (COP) of the refrigeration absorption system increases when the temperature of the absorber, condenser, and evaporator rises. On the contrary, increasing the temperature of the generator leads to the reduction of the COP of the system. It is also found that the pump power increases when the temperature of the absorber rises, while on the contrary decreases when the generator temperature increases. The results also reveal that the amount of heat input into the generator is directly proportional to the temperature of the generator. While the heat output of the absorber decreases when the temperature of the absorber increases.

1. Introduction
The vapor absorption cycle is one of the commonly used cooling systems besides vapor compression. The advantage of the vapor absorption cycle is that it uses heat to increase pressure. The vapor absorption cycle utilizes the waste heat as the input energy of the generator allowing the condition for the weak solution changes to produce the cooling effect. This technology can significantly reduce the electricity used for the cooling system, thereby, reducing the operational cost.

There are several studies on the absorption refrigeration system carried out by the researchers, including the theoretical [1-2] and experimental [3-4]. Manu and Chandrashekar [1] conducted a theoretical study on the single-stage lithium bromide-water vapor absorption heat pump. Maji [2] investigated the performance of solar-assisted absorption refrigeration which includes the ejector into the system with the Lithium bromide-water as the working pair. An experiment was conducted by

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Hanriot et al., [3] to study the effect of the technical parameters on the performance of the ammonia-water refrigeration cycle used automotive exhaust gas as the source heat.

This hypothetical study investigates the performance of a water lithium bromide absorption refrigeration, using the waste heat of a 6 MWe steam power plant, for cold storage application. The cold storage has a capacity of five tons of fish. The study is conducted base on the thermodynamic analysis. The performance of the absorption refrigeration cycle is examined considering the various temperatures of each component of the system.

2. Research Methodology

2.1 Cold Storage

The cold storage design is based on Charlie’s work [5]. The cold storage has a capacity of five tons of fish. It has dimensions of 6 m long, 2.5 m wide, 2.6 m high and 0.194 m thick. The cold storage cooling load is 115.28 kW. Table 1 shows the design parameters of the cold storage and Tables 2 presents the materials of the cold storage and their thermal conductivities.

Table 1. Design parameter of the cold storage.

| Design Parameter                        | Value |
|----------------------------------------|-------|
| Ambient air temperature (°C)           | 27.5  |
| Outdoor Air Relative Humidity (%)      | 75    |
| Wind Speed (mph)                       | 7.5   |
| Indoor air temperature (°C)            | -7    |
| Indoor Air Relative Humidity (%)       | 90    |

Table 2. Wall materials and their thermal conductivities.

| Material                | Thickness (m) | Thermal conductivity (Kkal/h °C) | Material                | Thickness (m) | Thermal conductivity (Kkal/h °C) |
|-------------------------|---------------|-------------------------------|-------------------------|---------------|-------------------------------|
| Wall                    |               |                               | Ceiling                 |               |                               |
| Carbon steel            | 0.002         | 30.95                         | Carbon steel            | 0.002         | 30.95                         |
| Glass wool              | 0.03          | 0.032                         | Glass wool              | 0.045         | 0.032                         |
| Polystyrene             | 0.05          | 0.033                         | Polystyrene             | 0.1           | 0.033                         |
| Air gap                 | 0.11          | 0.095                         | Air gap                 | 0.21          | 0.095                         |
| Aluminium               | 0.002         | 176                           | Aluminium               | 0.02          | 176                           |
| Floor                   |               |                               | Door                    |               |                               |
| Asphalt shingles        | 0.005         | 0.053                         | Carbon steel            | 0.002         | 30.95                         |
| Pine wood board         | 0.005         | 0.126                         | Glass wool              | 0.03          | 0.032                         |
| Polystyrene             | 0.1           | 0.033                         | Polystyrene             | 0.031         | 0.033                         |
| Glass wool              | 0.04          | 0.032                         | Air gap                 | 0.03          | 0.095                         |
| Carbon steel            | 0.004         | 30.95                         | Aluminum                | 0.002         | 176                           |

2.2 Heat Source

In this study, the heat source used as the heat input to the generator comes from the waste heat of a 6MWe steam power plant. Figure 1 shows the schematic of the steam power plant and the Rankine cycle.
The amount of heat dissipated by the condenser of the steam power plant is estimated by using the thermodynamic analysis given as follows:

\[ Q_{out} = \dot{m}(h_2 - h_3) \]  

In which, \( Q_{out} \) is the heat released by the condenser (kW), \( \dot{m} \) is the mass flow rate of the steam (kg/s), \( h_2 \) is the actual enthalpy at point 2 (kJ/kg), and \( h_3 \) is the saturated liquid enthalpy at point 3 (kJ/kg).

The mass flow rate of the steam is calculated as:

\[ \dot{m} = \frac{W_{cycle}}{(h_1-h_2)-(h_4-h_3)} \]  

In which, \( \dot{m} \) is the steam mass flow rate (kg/s), \( W_{cycle} \) is the net power output of the cycle (W), \( h_1 \) is the enthalpy of the superheated vapor at point 1 (kJ/kg), \( h_2 \) is the actual enthalpy at point 2 (kJ/kg), and \( h_3 \) is the saturated liquid enthalpy at point 3 (kJ/kg), and \( h_4 \) is specific enthalpy the pump exit.

The enthalpy at point 2 is determined as follows:

\[ h_2 = h_1 - \eta_t(h_1 - h_{2s}) \]  

In which, \( h_2 \) is the actual enthalpy at point 2 (kJ/kg), \( h_1 \) is the enthalpy of the superheated vapor at point 1 (kJ/kg), \( \eta_t \) is the steam turbine efficiency (it is assumed 85 %), \( h_{2s} \) is the ideal enthalpy at state 2s (kJ/kg).

where the enthalpy at state 2s is given as:

\[ h_{2s} = h_f + X_2. h_{fg} \]  

In which, \( h_{2s} \) is the ideal enthalpy at state 2s (kJ/kg), \( h_f \) is the saturated liquid enthalpy (kJ/kg), \( X_2 \) is the quality of the liquid-vapor mixture, and \( h_{fg} \) is the evaporation enthalpy (kJ/kg).
The enthalpy at point 4 is:

\[ h_4 = h_3 + \frac{\dot{W}_p}{\dot{m}} \]  \quad (5)

In which, \( h_4 \) is the enthalpy at point 4 (kg/s), \( h_3 \) is the saturated liquid enthalpy at point 3 (kJ/kg), \( \dot{W}_p \) is the pump power (kW), and \( \dot{m} \) is the mass flow rate of the working fluid (kg/s).

The pump power over the working mass fluid flow rate is calculated as:

\[ \frac{\dot{W}_p}{\dot{m}} = \frac{v_3(p_4-p_3)}{\eta_p} \]  \quad (6)

In which, \( \dot{W}_p \) is the pump power (kW), and \( \dot{m} \) is the mass flow rate of the working fluid (kg/s), \( v_3 \) is the specific volume of saturated liquid at point 3 (m\(^3\)/kg), \( p_4 \) and \( p_3 \) are the pressures at points 4 and 3, respectively, \( \eta_p \) is the pump efficiency (it is assumed 85%).

2.3 Absorption Refrigeration Cycle

The thermodynamic analysis is carried out to examine the performance of the absorption refrigeration cycle. Based on the energy balance on each component of the absorption refrigeration cycle (i.e. absorber, condenser, generator, and evaporator), the governing equations are then presented. The thermodynamic analysis of the absorption refrigeration cycle assumes a steady-state and steady flow. Besides, it ignores the pressure drop due to friction. Figure 2 shows a schematic of the absorption refrigeration cycle.

![Figure 2. The schematic of the absorption refrigeration cycle.](image)

1) Evaporator

The energy balance at the evaporator is illustrated in figure 3 and represents mathematically as follows:
\[ Q_e = \dot{m}(h_{10} - h_9) \]  \hspace{1cm} (7)

In which, \( Q_e \) is the heat transfer rate of the evaporator (kW), \( \dot{m} \) is the mass flow of the refrigerant (kg/s), \( h_9 \) and \( h_{10} \) are the enthalpy at points 9 and 10 (kJ/kg), respectively.

The mass flow rate of the refrigerant can be calculated by rearranging the Equation (7)

\[ \dot{m} = \frac{Q_e}{h_{10} - h_9} \]  \hspace{1cm} (8)

Figure 3. The energy balance at the evaporator.

2) Condenser

The energy balance at the condenser is illustrated schematically in figure 4 and presented mathematically in Equation (9).

\[ Q_c = \dot{m}(h_7 - h_8) \]  \hspace{1cm} (9)

In which, \( Q_c \) is the heat transfer rate of the condenser (kW), \( \dot{m} \) is the refrigerant mass flow rate (kg/s), \( h_7 \) and \( h_8 \) are the enthalpy at points 7 and 8 (kJ/kg), respectively.

3) Absorber

Figure 5 shows the energy balance at the absorber and mathematically is presented by Equation (10).

\[ Q_a = \dot{m}h_{10} + \dot{m}_{ss}h_6 - \dot{m}_{ws}h_1 \]  \hspace{1cm} (10)
In which, $Q_a$ is the heat transfer rate of the absorber (kW), $\dot{m}$ is the refrigerant mass flow rate (kg/s), $\dot{m}_{ss}$ is the mass flow rate of strong solution (kg/s), $\dot{m}_{ws}$ is the mass flow rate of weak solution (kg/s), $h_1, h_6$ and $h_{10}$ are the enthalpy at points 1, 6, and 10.

Where

$$\lambda = X_{ws}/(X_{ss} - X_{ws})$$

The $\lambda$ can be also calculated as

$$\lambda = \dot{m}_{ss}/\dot{m}$$

The mass flow rate of the strong solution can be calculated by rearranging the Equation (12)

$$\dot{m}_{ss} = \dot{m}\lambda$$

While the mass flow rate of the week solution is given as

$$\dot{m}_{ws} = \dot{m} + \dot{m}_{ss}$$

In which, $\lambda$ is the circulation ratio, $X_{ws}$ is the percentage of LiBr in the weak solution (%), $X_{ss}$ is the percentage of LiBr in the strong solution (%), $\dot{m}_{ss}$ is the mass flow rate of the strong solution (kg/s), $\dot{m}_{ws}$ is the mass flow rate of weak solution (kg/s), $\dot{m}$ is the refrigerant mass flow rate (kg/s).

4) Heat Exchanger

A heat exchanger is usually provided to increase the efficiency of the absorption refrigeration cycle. The heat exchanger is used to cool the strong solution (returning from the generator to the absorber) by transferring its heat to the weak solution (pumping from the absorber to the generator). The energy balance of the heat exchanger is shown schematically in Figure 6 and presented as

$$\dot{m}_{ws}(h_3 - h_2) = \dot{m}_{ss}(h_4 - h_5)$$

Rearrange Equation (15), the enthalpy at point 3 is

$$h_3 = \frac{\dot{m}_{ss}(h_4 - h_5)}{\dot{m}_{ws}} + h_2$$

![Figure 6. The energy balance at the heat exchanger.](image)

5) Generator

The energy balance at the generator is illustrated in figure 7 and presented by Equation (17).
Figure 7. The energy balance at the generator.

\[ Q_g = \dot{m}_{ws} h_7 + \dot{m}_{ss} h_4 - \dot{m}_{ws} h_3 \]  \hspace{1cm} (17)

6) Pump

The energy balance for the pump is shown in figure 8 and mathematically presented by Equation (18).

\[ W_p = \dot{m}_{ws} (\Delta H)/\rho \]  \hspace{1cm} (18)

In which \( W_p \) is the pump power (kW), \( \dot{m}_{ws} \) is the mass flow rate of weak solution (kg/s), \( \Delta H \) is the pressure drop (kPa), \( \rho \) is the density (kg/m\(^3\)).

The density is calculated as:

\[ \rho = \rho(LiBr) \times X_{ss} + \rho(air) \times (100 - X_{ss}) \]  \hspace{1cm} (19)

7) Coefficient of Performance (COP)

The Coefficient of Performance of the absorption refrigeration cycle is calculated as follows

\[ COP = Q_e / (Q_g + W_p) \]  \hspace{1cm} (20)

In which, \( COP \) is the coefficient of performance of the absorption refrigeration cycle, \( Q_e \) is the heat transfer rate of the evaporator (kW), \( Q_g \) is the heat transfer rate of the generator (kW), \( W_p \) is the pump power (kW).

3. Hypothetical Study

The performance of the absorption refrigeration cycle is investigated under specific ranges of temperatures of the refrigeration system’s components (i.e generator, absorber, condenser, evaporator). The influence of the absorber and generator temperatures on the power input of the pump is also examined. The simulation of the performance of the absorption refrigeration cycle is conducted initially by utilized parameters as given in table 3. Based on these reference parameters, the performance of the cooling system considering the different component temperature of the cooling system is carried out. Table 4 presents the temperature ranges used in the simulation, as suggested by Sauchin Kausik [6], for each component of the absorption refrigeration cycle.
### Table 3. The reference temperature of the absorption refrigeration cycle components.

| Parameter                      | Value |
|--------------------------------|-------|
| Temperature of condenser (°C)  | 40    |
| Temperature of evaporator (°C) | 2.5   |
| Temperature of absorber (°C)   | 20    |
| Temperature of generator (°C)  | 85    |

### Table 4. The variable temperature of the absorption refrigeration cycle components.

| Parameter                  | Temperature Ranges |
|----------------------------|--------------------|
| Temperature of Generator (°C) | 55-90             |
| Temperature of condenser (°C)  | 24-46             |
| Temperature of absorber (°C)   | 16-32             |
| Temperature of evaporator (°C)  | 2.5-10            |

The input parameters of the governing equations are determined based on the reference temperature and state condition of the working fluid at each component of the absorption refrigeration cycle. For the cycle with lithium bromide water pair, the water serves as the refrigerant for the system. The saturation pressure for condensation in the condenser at 40 °C can be obtained from the steam table which is equal to 0.074 bar (7.384 kPa). The generator pressure is assumed to operate at the same pressure as the condenser. The saturation pressure for saturated steam form in the evaporator at 2.5 °C is equal to 0.00737 bar (0.737 kPa), which also corresponds to the pressure at the evaporator. Based on this information, the input parameters (i.e enthalpy) used in the governing equations can be determined. The enthalpy of the saturated water and superheated water vapor at given temperatures can be determined from the steam table. While the enthalpy and concentrations of the mixed solution are obtained from the LiBr-Water graph, pressure-temperature-concentration-enthalpy (P-T-X-h). As an example, the table 5 presents input parameters for the governing equations at reference component temperatures (see table 3).

### Table 5. Input parameter at the reference temperature of each component.

| Point | Temperature (°C) | Pressure (kPa) | Enthalpy (kJ/kg) | LiBr Concentration (%) |
|-------|------------------|----------------|------------------|------------------------|
| 1     | 20               | 0.737          | 5.53             | -178                   | 50 |
| 2     | 20               | 7.384          | 55.50            | -178                   | 50 |
| 3     | -                | 7.384          | 55.50            | -                      | 50 |
| 4     | 85               | 7.384          | 55.50            | -80                    | 59 |
| 5     | 20               | 7.384          | 55.50            | -187                   | 59 |
| 6     | 20               | 0.737          | 5.53             | -187                   | 59 |
4. Result and discussion

4.1. Analysis of potential heat output of steam power plant used for the absorption refrigeration cold storage

In designing cold storage with an absorption refrigeration cycle, it is necessary to calculate the amount of the heat input to the generator to ensure the cold storage properly operates. In this study, the heat supplied to the absorption refrigeration cycle is extracted from the waste heat of a 6MWe steam power plant. The heat output of the power plant is analyzed thermodynamically based on the parameters provided by the Siemen SST-200 steam turbine, as shown in Table 6.

| Power output | 6 MWe |
|--------------|-------|
| Inlet pressure | 120 bar |
| Inlet temperature | 540 °C |
| Back Pressure | 2 bar |
| Efficiency | 85% |
| Speed | 14,600 rpm |

From the calculation, it is obtained that the waste heat the steam power plant corresponds to 16.2 MW. According to Osama [8], the potential for waste heat can be used by the absorption refrigeration cycle ranges from 58-62%. These values correspond to 9.2-9.9 MW of the waste heat of the power plant.

4.2. The Effect of generator temperature on the COP of the absorption refrigeration cycle

Figure 9 shows the effect of the generator temperature on the COP of the absorption refrigeration cycle. It is observed that the highest COP is produced at a generator temperature of 65 °C. When the generator temperature is increased to 70 °C, it can be seen that the COP of the absorption refrigeration cycle significantly decreases. This tendency is affected by the escalation of the strong solution concentration (Xss) leading to a drastic decline in the circulation ratio (λ) from 50 to 13.16. As a result, it affects the mass flow rate of both the strong (ṁss) and weak (ṁws) solutions and contributes to an increase in generator heat input (Qg). In the temperature range of 70-90 °C, the insignificant escalation of the strong solution concentration (Xss) causes the circulation ratio (λ) to decrease gradually. From the figure, it can be concluded that the higher the generator temperature, the lower the COP of the absorption refrigeration cycle is attained.
4.3. The effect of absorber temperature on the COP of the absorption refrigeration cycle

Figure 10 presents the variation in COP of the absorption refrigeration cycle at a specific range of the absorber temperature. From the figure, it can be seen that the lowest COP is generated at an absorber temperature of 16 °C. The COP rises gradually to the highest value at the absorber temperature of 32 °C. The increase in COP value is influenced by the increase in enthalpy of the solution in the absorber (h1 and h2) and also the concentration of the weak solution of the lithium bromide (Xws) due to changes in the absorber temperature. Thus, it can be concluded that the COP of the absorption refrigeration cycle increases directly proportional to the increase of the absorber temperature.

4.4. The effect of condenser temperature on the COP of the absorption refrigeration cycle

Figure 11 shows the effect of the condenser temperature on the COP of the refrigeration absorption cycle. From the figure, it can be seen that the COP value of the refrigeration absorptions cycle increases over the condenser temperature. It is observed that the COP of the refrigeration system significantly increases from the temperature of 24°C-36°C. This tendency occurs due to the increase in the saturated temperature of refrigerant during the condensation process in the condenser is in line with the increase in the saturated pressure of the refrigerant in the condenser. The rise of the condenser pressure renders the generator pressure to increase as both the condenser and the generators operate at
the same pressure. As a result, it decreases the concentration of the strong solution of the Li Br (Xss), yielding the circulation ratio to increase (λ). The increase in the value of λ is directly proportional to the increase in the mass flow rate of the refrigerant solution (ṁ), the weak solution (ṁws), and the strong solution (ṁss), consequently decreases the amount of heat input supplied to the generator (Qg). Due to the COP of the absorption refrigeration cycle is inversely proportional to the heat input supplied to the generator (See Equation (20)), thus, the lower the heat input supplied, the higher the COP of the refrigeration cycle attained. It is observed that the COP of the absorption refrigeration cycle insignificantly changes when the temperature of the condenser is increased from 36–40 °C. At the temperature range of 40–46 °C, the amount of heat input supplied to the generator (Qg) is decreased significantly which results in a significant increase of COP of the absorption refrigeration cycle.

![Figure 11. Variation of COP with condenser temperature.](image)

4.5. The effect of evaporator temperature on the COP of the absorption refrigeration cycle

![Figure 12. Variation of COP with an evaporator temperature](image)

Figure 12 shows that the variation in the evaporator temperature with the COP of the absorption refrigeration cycle. From the figure, it appears that the increase in the COP value of the absorption refrigeration cycle is directly proportional to the increase in temperature of the evaporator. This is due to the increase in the enthalpy of the refrigerant leaving the evaporator is in line with the evaporator temperature. As a result, it decreases the amount of mass flow rate of the refrigerant needed to be circulated in the refrigeration cycle. A decrease in the mass flow rate of the refrigerant renders a reduction in the mass flow rate of the strong and weak solutions and circulation ratio. Besides, the changes in the evaporator temperature affect the mass flow rate and pressure of the working fluid.
These phenomena affect the pump power and the amount of heat input at the generator that influences the COP of the refrigeration absorption cycle. This result is also confirmed by Senapati [7].

4.6. The effect of generator temperature on the pump power of the absorption refrigeration cycle

Figure 13 shows the relationship between generator temperature and pump power in the absorption refrigeration cycle. It can be seen that the highest pump power (Wp) is generated at a generator temperature of 65 °C. The pump power (Wp) drops significantly when the generator temperature rises to 70 °C. This tendency occurs due to the increase in the value of the strong solution (Xss) causing a drastic decrease in the value of the circulation ratio (λ), thereby reducing the mass flow rate of both the strong (ṁss) and the weak (ṁws) solutions. The significant reduction of the mass flow rate of the weak solution (ṁws) implies a remarkable reduction in pump power (Wp) needed to circulate the weak solution (ṁws). While at a temperature of 70 °C-90 °C, it is observed that the pump power (Wp) demand has decreased slowly. This is due to an increase in the concentration of the strong solution (Xss) causes the circulation ratio (λ) to slowly decrease, thereby reducing both the mass flow rate of strong (ṁss) and weak (ṁws) solutions.

4.7. The effect of absorber temperature on the pump power of the absorption refrigeration cycle

From Figure 14 it can be seen that the pump power increases with increasing absorber temperature. The pump requires the lowest power (Wp) when the absorber temperature is 16 °C and the power continues to increase along with the increasing of the absorber temperature. An increase in absorber temperature causes both the flow rate (ṁws) and density (ρ) of the weak solution to increase. Based on the relationship as given by Equation (18), the pump power depends on the value of the multiplication of the mass flow rate of the weak solution (ṁws) and pressure drop (ΔH) divided the weak solution density (ρ). Thus, it can be understood that the higher the flow rate of the weak solution (ṁws), the higher the pump power (Wp) is required. Conversely, the pump power (Wp) is being lower when the density (ρ) of the weak solution is getting higher. Since the numerator value is higher than the denominator, it contributes to the enhancement of the required pump power.
Figure 14. Variation of pump power with absorber temperature.

4.8. The effect of generator temperature on the generator heat input of the absorption refrigeration cycle

Figure 15. Variation of generator heat input with generator temperature.

In the absorption refrigeration cycle, the largest energy input required for the operation of the system is the heat supplied for the generator. Figure 15 shows that the heat input supplied to the generator (Qg) is getting higher as the temperature of the generator increases. The increase in generator temperature contributes to the increase in the concentration of the strong solution (Xss), enthalpy of the weak solution leaving the heat exchanger (h3), and the enthalpy of the strong solution leaving the generator (h4). Increasing the concentration of the strong solution (Xss) renders the circulation ratio (λ) to decrease. The reduction in circulation ratio (λ) leads to the flow rate of both the weak (ṁws) and the strong (ṁss) solutions to decrease. So that it can be concluded in designing an absorption refrigeration system, the temperature of the generator must be set as low as possible (in the range of generator temperatures given from the literature) so that the system is more efficient.

4.9. The effect of absorber temperature on the absorber heat output of the absorption refrigeration cycle

It can be observed from figure 16 that the amount of heat dissipated by the absorber decreases in line with the increase of the temperature absorber. When the absorber temperature increases, the concentration (Xws) and the enthalpy (h1) of the weak solution would also increase. An increase in the concentration of the weak solution renders the mass flow rates of the weak (ṁw) and the strong (ṁss) solution to increase. This phenomenon causes the absorber required to release more heat to operate properly.
5. Conclusions

The following conclusion can be drawn from the study of the absorption refrigeration cycle for cold storage application:

1. The waste heat released by the condenser of the steam power plant has enormous potential as an energy source to operate an absorption refrigeration cooler.
2. Changes in temperature of the Condenser, Generator, Absorber, and Evaporator greatly affect the Coefficient of Performance of the absorption refrigeration cooler.
3. Changes in generator and absorber temperatures affect pump power. The change in the pump power is relatively small, thus, it can be ignored in calculating the Coefficient of Performance of the absorption refrigeration cycle.
4. The amount of heat required to supply to the generator is greatly influenced by the temperature of the generator. The lower the temperature of the generator, the lower the amount of heat required to operate the system.

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