**ICMIEE-PI-140119**

# Implementation of a waste-heat driven vapor absorption-based car air conditioning system

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**ABSTRACT**
Waste-heat driven air conditioning system eliminates compressor and introduces absorber, pump, and generator. Thus, the consumption of electricity is reduced and waste-heat of exhaust gas is used as energy input. The most common refrigerant-absorbent pair in vapor absorption system is $NH_3$-water due to its availability, low cost, and high refrigerating effect. The drawback of using $NH_3$ as refrigerant is its toxicity, corrosiveness, and explosiveness. A small capacity vapor absorption system is first analyzed for its implementation in car air conditioning unit. Components are designed based on capacity 2 kW. The appropriate equations describing the working properties are specified. The cooling load, COP of the unit, variation of absorption rate with $NH_3$ weight percentage, variation of evaporator outlet temperature are examined. The experimentally obtained COP is 0.24 which is less than the designed COP 0.58. Absorber heat rejection increases with increasing absorber temperature.

Keywords: Vapor absorption system; Automobile air conditioning; Waste heat; COP.

## 1. Introduction

With the development of modern technology, the engineers have become more concerned with the further technological development of the existing systems and eliminate the drawbacks of those systems. This includes improvisation of performance and efficiency, reduction in cost, precision, energy savings, usage of renewable energy, and many more. In this context, some alternative systems of car air conditioning have been developed.

Today, almost all car air-conditioning systems are charged with R-134a. However, alternatives with lower GWP than R-134a are desirable. Some new systems are being developed in order to revitalize the use of ecologically safe refrigerants. For example, a system for car air-conditioning using $CO_2$ as the refrigerant has been developed by Lorentzen and Pettersen (1993). The testing of a laboratory prototype has shown that $CO_2$ is an acceptable refrigerant for car air-conditioners.

Due to the international attempt to find alternative energies, absorption refrigeration has become a prime system for many cooling applications. Where thermal energy is available the absorption refrigerator can very well substitute the vapor compression system. It is a well-known fact that a large amount of heat energy associated with the exhaust gases from an engine is wasted. A rough energy balance of the available energy in the combustion of fuel in a motor car engine shows that one third is converted into shaft work, one third is lost at the radiator and one third is wasted as heat at the exhaust system (Greene and Lucas, 1969). Even for a relative small car-engine, such as for the Nissan1400, 15 kW of heat energy can be utilized from the exhaust gas (Wang, 1997). This heat is enough to power an absorption refrigeration system to produce a refrigeration capacity of 5 kW.

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This project is based on the construction of a car cooling system by using the heat of exhaust gas. Here, the prime vision is to use the waste energy of exhaust gas and to construct a cooling system by giving the heat input in the refrigeration cycle from that exhaust gas.

In the conventional cooling system of a car, usually vapor compression refrigeration cycle is used. Here, a certain amount of energy from engine crank shaft is used to run the compressor. On the other hand, a huge amount of heat energy is released to environment as waste heat. If that heat is used as energy input of cooling process, it is possible to save a huge amount of energy. Moreover, the use of vapor absorption refrigeration system instead of vapor compression system eliminates the compressor. As a result, load on engine decreases and efficiency of engine increases.

## 2. Working Procedure of a Single Effect Aqua-Ammonia Air Conditioning System of a Car

The Working principle is shown in fig.1

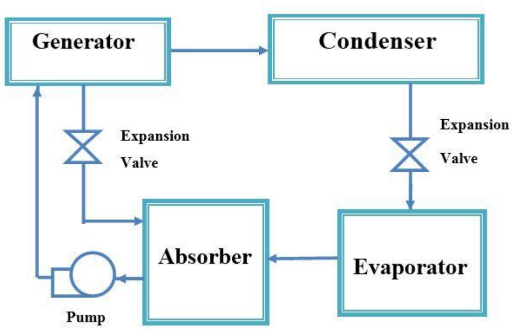


Fig.1.a) Block Diagram of Simplified Single-effect Aqua-Ammonia Absorption System

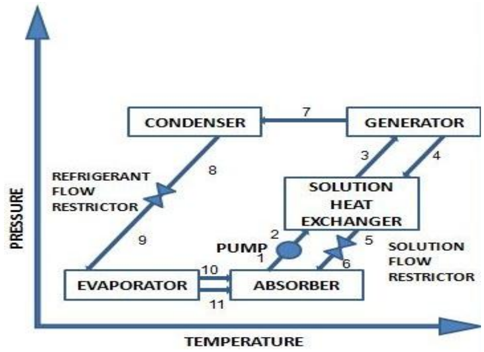


Fig.1.b) P-T diagram of Aqua-ammonia absorption cooling cycle.

A simple vapor absorption cycle eliminates compression. It uses absorber, pump, generator, analyzer, and reflux condensed in combined form as a replacement of compressor. Here, we will use aqua-ammonia system, where ammonia is used as refrigerant and water is used as absorber.

The low pressure refrigerant from the evaporator is absorbed by the liquid solution in the absorber. The pump receives low pressure liquid from the absorber and elevates the pressure. After that, the high pressure liquid solution is delivered to the generator. In the generator, heat from a high temperature source drives off the vapor that had been absorbed by the solution. The vapor refrigerant then enters the condenser. Remaining liquid solution returns to the absorber through a throttling valve whose purpose is to provide a pressure drop to maintain the pressure difference between the generator and absorber. In condenser, the vapor is cooled and condensed. There the heat is rejected in the atmosphere. After that the condensed refrigerant is passed through the expansion valve where the pressure is released. Then the low pressure refrigerant enters the evaporator. In evaporator, the refrigerant is evaporated by using the heat extracted from the refrigerated space. Thus, the space to be refrigerated gets cooled at a temperature lower than atmospheric temperature.

The main concern is to work with the function of a generator in vapor absorption refrigeration system. We have previously discussed that generator needs work input in the form of heat energy to separate refrigerant by evaporating it from the liquid absorbent solution. We are going to provide this heat input from the exhaust gas of car engine. From the exhaust pipe, it is possible to get an average temperature of 300◦c to 400◦c. By using an air to water heat exchanger, this heat can be extracted from exhaust gas and delivered to the generator. Thus, a huge amount of waste heat from the exhaust gas can be utilized in useful purpose.

## 3. Design of Air Conditioning System of a Car

Cooling load was calculated for a passenger car and according to cooling load calculation, the other portions of an air conditioning system are designed.

### 3.1 Calculation of cooling load of a Compact Car

Overall heat transfer coefficient: [1]

$$U_{glass} = \frac{1}{\left(\frac{1}{h_A} + \frac{dx_w}{k} + \frac{1}{h_B}\right)}$$

Heat transfer through glass: (conduction)

$$Q_{glass} = A_{glass} \times U_{glass} \times ETD_{glass}$$

Heat transfer through body: (conduction)

$$Q_{body} = A_{body} \times U_{body} \times ETD_{body}$$

Heat transfer due to solar radiation: [1]

$$Q_{glass} = A_{glass} \times SHGF$$

Total sensible heat= Sum of all Sensible heats

Effective sensible heat=Sum of total sensible heat and heat gain due to occupancy

Heat gain due to infiltration: [1]

$$Q_{s,inf} = m_{inf} \times C_p \times (T_0 - T_i)$$

Effective latent heat=Sum of all latent heat, heat gain due to infiltration and heat gain due to occupancy

Grand total Heat, Q= Sum of Effective sensible heat and Effective Latent Heat

Obtained cooling load using the equations stated above is 4.393 KW.

If air locks are used in doors & windows, we can minimize filtration & by pass air considerably.

By using heat non-conducting leather or Rexene cover inside the body & seats, this heat can be considerably reduced. So, we will consider the cooling capacity of 2 KW.

### 3.2 Basic Assumptions and Design Parameters

To perform designing of equipment size and performance evaluation of a single-effect Aqua-ammonia absorption cooler basic assumptions are made. The basic assumptions are:

1. The steady state refrigerant is pure water.
2. There are no pressure changes except through the flow restrictors and the pump.
3. At points 1, 4, 8 and 11, there is only saturated liquid.
4. At point 10 there is only saturated vapor.
5. The pump is isentropic.
6. There are no jacket heat losses.
7. The capacity of the system is 2kW.

Some design parameters have been set up as per design requirement. They are stated below:

| Parameter | Symbol | Value |
|---|---|---|
| Capacity | $Q_e$ | 2 kW |
| Generator solution exit temperature | $T_4$ | 90 $^0$C |
| Weak solution mass fraction | $X_1$ | 55% $NH_3$ |
| Strong solution mass fraction | $X_4$ | 60% $NH_3$ |
| Generator vapor exit temperature | $T_7$ | 85 $^0$C |

**3.3 Evaporator analysis**

Since, in the evaporator, the refrigerant is saturated water vapor and temperature ($T_{10}$) is assumed 10 $^0$C, the saturation pressure and enthalpy are calculated from curve fits [5].

Pressure and Enthalpy are determined by using these equations: [6]

$$P = 0.000000000002T^6 - 0.000000003T^5 + 0.0000002T^4 + 0.00003T^3 + 0.0014T^2 + 0.0444T$$

$$h = -0.00125397T^2 + 1.88060937T + 2500.559$$

Aqua-ammonia solution and refrigerant pressure and temperatures are obtained from these equations: [6]

$$LogP = C + \frac{D}{(T_{ref} + 273)} + \frac{E}{(T_{ref} + 273)^2}$$

$$T_{sol} = T_{ref}\Sigma A + \Sigma B$$

Now, $\Sigma A$ and $\Sigma B$ are obtained from following equations:

$$\Sigma A = A_0 X^0 + A_1 X^1 + A_2 X^2 + A_3 X^3$$
$$\Sigma B = B_0 X^0 + B_1 X^1 + B_2 X^2 + B_3 X^3$$

For range 45% < X < 70% $NH_3$
(Mass fraction, X = 60%)
$A_0$ = - 2.00755; $A_1$ = 0.16976;
$A_2$ = - 0.003133362; $A_3$ = 0.0000197668
$B_0$ = 124.937; $B_1$ = - 7.71649;
$B_2$ = 0.152286; $B_3$ = -0.0007959

Enthalpy of $NH_3$-water solution is obtained from following solution: [7]

$$h = \Sigma A + T\Sigma B + T^2\Sigma C$$

Now, $\Sigma A$ $\Sigma$B, and $\Sigma$C are obtained from following equations:

$$\Sigma A = A_0 X^0 + A_1 X^1 + A_2 X^2 + A_3 X^3 + A_4 X^4$$
$$\Sigma B = B_0 X^0 + B_1 X^1 + B_2 X^2 + B_3 X^3 + B_4 X^4$$
$$\Sigma C = C_1 X^1 + C_2 X^2 + C_3 X^3 + C_4 X^4$$

Where, these values are obtained for specific range of mass fraction:

$A_0$ = - 2024.33; $A_1$ = 163.309;
$A_2$ = -4.88161; $A_3$ = 0.06302948;
$A_4$ = -0.0002913704.
$B_0$ = 18.2829; $B_1$ = -1.1691757; $B_2$ = 0.03248041;
$B_3$ = - 0.0004034184, $B_4$ = 0.0000018520569.
$C_0$ = -0.037008214; $C_1$ = 0.03248041;
$C_2$ = -0.000081313015; $C_3$ = 0.00000099116628;
$C_4$ = -0.0000000044441207

Mass balance on evaporator is as follows:
$m_9 = m_{10} + m_{11}$

Energy balance on evaporator is as follows:

$Q_e = m_{10} h_{10} + m_{11}h_{11} - m_9h_9$

From these two equations the value of $m_9$, $m_{10}$, and $m_{11}$ can be obtained.

**3.4 Absorber analysis**

Since the values of $m_{10}$, $m_{11}$ are known, mass balance around the absorber can be stated as bellows:

$$m_1 = m_{10} + m_{11} + m_6$$
$$x_1 m_1 = x_6 m_6$$

From these two equations the value of $m_1$ and $m_6$ can be obtained.

The temperature and pressure around the absorber can be calculated by using the following equations:

$$LogP = C + \frac{D}{(T_{ref} + 273)} + \frac{E}{(T_{ref} + 273)^2}$$
$$T_{sol} = T_{ref}\Sigma A + \Sigma B$$

Here,
$$\Sigma A = A_0 X^0 + A_1 X^1 + A_2 X^2 + A_3 X^3$$
$$\Sigma B = B_0 X^0 + B_1 X^1 + B_2 X^2 + B_3 X^3$$
$A_0$ = - 2.00755; $A_1$ = 0.16976;
$A_2$ = - 0.003133362; $A_3$ = 0.0000197668
$B_0$ = 124.937; $B_1$ = - 7.71649;
$B_2$ = 0.152286; $B_3$ = - 0.0007959
C = 7.05; D = -1596.49; E = -104095.5

The equation of enthalpy is as bellows:

$$h = \Sigma A + T\Sigma B + T^2\Sigma C$$

Here,
$$\Sigma A = A_0 X^0 + A_1 X^1 + A_2 X^2 + A_3 X^3 + A_4 X^4$$
$$\Sigma B = B_0 X^0 + B_1 X^1 + B_2 X^2 + B_3 X^3 + B_4 X^4$$
$$\Sigma C = C_1 X^1 + C_2 X^2 + C_3 X^3 + C_4 X^4$$
$A_0$ = - 2024.33; $A_1$ = 163.309;
$A_2$ = -4.88161; $A_3$ = 0.06302948; $A_4$ = -0.0002913704.

$B_0$ = 18.2829; $B_1$ = -1.1691757;
$B_2$ = 0.03248041; $B_3$ = - 0.0004034184;
$B_4$ = 0.0000018520569.
$C_0$ = -0.037008214; $C_1$ = 0.03248041;
$C_2$ = -0.000081313015; $C_3$ = 0.00000099116628,
$C_4$ = -0.0000000044441207.

The enthalpy at point 5 is not known but can be determined from the energy balance.

$$m_2h_2 + m_4h_4 = m_3h_3 + m_5h_5$$

For, T = 34.9 $^0$C, $X_0$ = 55/100 = 0.55, Density of $NH_3$ solution is obtained as: [7]

$$\rho = 1145.36 + 470.84X_0 + 1374.79X_0^2 - (0.333393 + 0.571749X_0)(273 + T)$$

Since, all variables are known, the pump power can be calculated as:

$$w = m_1v_1(p_2 - p_1)$$

Energy balance on the absorber is stated by the following equation:

$$Q_a = m_{10}h_{10} + m_{11}h_{11} + m_6h_6 - m_1h_1$$

### 3.5 Generator analysis

The heat input to the generator is determined from energy balance equations:

$$Q_g = m_4h_4 + m_7h_7 - m_3h_3$$

$$m_7 = m_3 - m_4$$

### 3.6 Condenser analysis

The condenser heat can be determined from an energy balance equation given below:

$$Q_c = m_7(h_7 - h_8)$$

### 3.7 Coefficient of performance

The COP is defined as follows:

$$COP = \frac{Q_e}{Q_g} = \frac{2}{3.45} = 0.58$$

According to the previous calculations Energy flows at various component of the system are shown in table below:

| Description | Symbol | kW |
|---|---|---|
| Capacity | $Q_e$ | 2 |
| Pump work | W | 0.067 |
| Absorber heat rejected | $Q_a$ | 2.8 |
| Heat input to the generator | $Q_g$ | 3.45 |
| Condenser heat rejected | $Q_c$ | 2.62 |

## 4. Design of Heat Exchangers

In single-pass heat exchangers, the temperature difference $\Delta T$ between the hot and the cold fluids is not constant but it varies with distance along the heat exchanger. In the heat transfer analysis, it is convenient to establish a mean temperature difference $(\Delta T_m)$ between the hot and cold fluids such that the total heat transfer rate $Q$ between the fluids can be determined from the following expression: [3]

$$Q = AU\Delta T_m$$

Mean temperature difference is calculated by using the equation stated below:

$$\Delta T_m = F\Delta T_{ln} = F\left(\frac{\Delta T_0 - \Delta T_L}{\ln(\Delta T_0/\Delta T_L)}\right)$$

The overall heat transfer coefficient ($U$) based on the outside surface of the tube is defined as [3]

$$U = \frac{1}{(D_0/D_i)(1/h_i) + (D_0/D_i)F_i + (1/2k)D_0 ln\,(D_0/D_i) + F_0 + 1/h_0}$$

For the design of the heat exchangers, the cooling water inlet and outlet temperatures are assumed. The cooling water inlet temperature depends exclusively on the available source of water, which may be a cooling tower or a well.

### 4.1 Absorber Design

For this design, the solution film can flow downward either on horizontal or on vertical tubes. The design of the horizontal tubes for the absorber, although theoretically well studied, presents a great problem with the shell tightness because of the large length of welds. In the case of this study the water vapor produced in the evaporator is absorbed in the flow of the $NH_3$-water solution and is condensing on the heat exchanger tubes. The design of the heat exchanger therefore requires values for heat and mass transfer coefficients.

The fig.2 shows a model of absorber that has been used in car air conditioning unit.

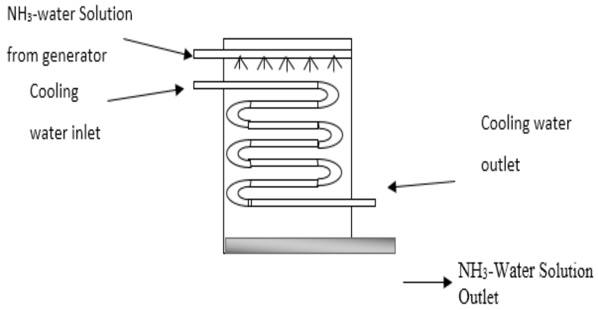


Fig.2: Schematic representation of an absorber

Design parameters of an absorber are stated in table below:

| Parameter | Value |
|---|---|
| Tube dimension | Tube outside diameter $D_o$ = 12.7 mm and inside diameter $D_i$ =11.7 mm |
| Cooling water inlet temperature | 25°C |
| Cooling water outlet temperature | 26°C |
| Mass flow rate of cooling water (m) | 0.21 kg/s |
| Absorber load $(Q_a)$ | 2.8kW |
| Solution cooling | From 44.6°C to 34.9°C |
| Absorber pressure | 1.227kPa |
| Inlet solution mass flow rate | 0.0117975 kg/s |

The independent variables, which affect the problem, are solution mass flow rate, solution inlet concentration, and absorber pressure and wall temperature. The data are correlated with the introduction of the “absorption percentage ($A_p$), defined as:

$$A_p = \frac{C_{in} - C_{out}}{C_{in} - C_{eq}} \times 100$$

Determination of the equilibrium concentration, $C_{eq}$ requires the solution of the following set of expressions:

$A = -2.00755 + 0.16976\,X - 3.133362 \times .001\,X^2 + 1.97668 \times 0.00001\,X^3$

$B = 321.128 - 19.322\,X + 0.374382\,X^2 - 2.0637 \times .001\,X^3$

C = 6.21147
D = -2886.373

$T' = (-2\,E / [D + (D^2 - 4\,E\,(C - \log(P / 6894.8))]^{.5}) - 459.72$
$T_W = (5/9)*(AT' + B - 32)$

E = -337269.46

The above set of expressions requires an iterative type of solution to find $C_{eq}$, given $T_W$ and P. In the case of this study $T_W$=31 °C and P=1227 Pa, therefore $C_{eq}$= 0.52 and from Eq. stated above, $A_P$=62.5. $A_P$ is correlated to the length of plate (L) by the expression:

$$L = am^b$$

Where, $a = -132\left(\frac{\ln(100 - A_p)}{86}\right), b = 1.33$

An iterative solution gives, $m$=0.0292 kg/m-s corresponding to the area of 5.4 m length pipe.
The next step is to check the area of pipes needed to cool the solution to the required level.
Patnaik *et al.* (1993), suggest that Wilke’s correlation, valid for constant heat flux wall with progressively decreasing difference from isothermal wall outside the entrance region, can be used for the falling film.
It is assumed that the flow is fully developed in a wavy, laminar regime and that the bulk solution temperature profile is linear with respect to the transverse coordinate (Patnaik *et al.*, 1993). Wilke’s correlation is:

$$h_s = \frac{k_s}{\delta}\left(0.29\,(Re_s)^{0.53}\,Pr_s^{\;0.344}\right)$$

The film thickness $\delta$ is given by:

$$\delta = \left(\frac{3\mu\Gamma}{\rho^2 g}\right)^{1/3}$$

And the solution Reynolds number ($Re_s$) for the tube *is:*

$Re_s$=4Γ/μ

### 4.2 Generator Design

The generator provides sensible heat and latent heat of vaporization. The sensible heat raises the inlet stream temperature up to the saturation temperature. This amount of heat, typical in practice, is 13% of the total heat required [6]. The heat of vaporization consists of the heat of vaporization of pure water and the latent heat of mixing of the liquid solution.

Design parameters for generator design are stated in table below:

| Parameter | Value |
|---|---|
| Tube dimension | Tube outside diameter $D_o$ = 9.53 mm and inside diameter $D_i$ =8.5 mm |
| Generator pressure | 9.66kPa |
| Generator solution | Entering: 60% $NH_3$ at 65°C<br>Leaving: 55% $NH_3$ at 90°C |
| Generator water vapor mass flow rate (*m*) | 0.00107 kg/s |
| Generator load $(Q_g)$ | 3.45kW |

## 5. Construction

After designing all components, the construction is done according to the figure below. Stainless Steel tubes and galvanized iron plates are used for constructing these components.

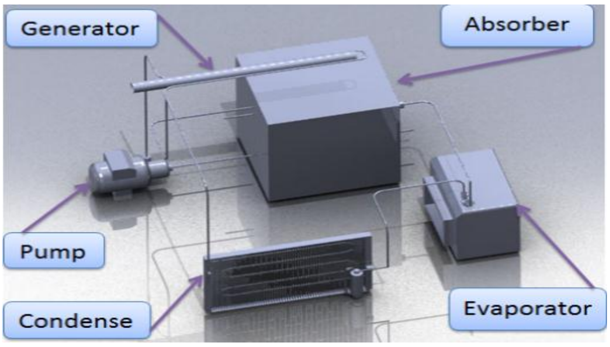


Fig.3: Schemetic Diagram of Vapor Absorption based Cooling System of a Car

## 6. Result and Conclusion

### 6.1 Inlet outlet temperature of various components

| Heat Exchangers | Cooling/Heating water temperature (°C) | |
|---|---|---|
| | Entering | Leaving |
| Generator | - | 59 |
| Condenser | - | 44 |
| Evaporator | 30 | 25 |
| Absorber | 25 | 26 |

### 6.2 COP analysis

The equation for COP calculation is: [1]

$$COP = \frac{Q_e}{Q_g}$$

$$Q_e = mC_p\Delta T = 0.84\ kW$$

$$Q_g = 3.45\ kW$$

So, COP= 0.24 which is less than the COP obtained in design before. The actual overall heat transfer coefficient of stainless steel tubes are less than the ideal heat transfer coefficient. This is because of rust on the upper surface of tubes. Another reason is the less concentration of ammonia in absorber than designed concentration.

### 6.3 Conclusion

1. A car cooling system which uses exhaust gas as heating source is designed based on some correlations and formulas involving vapor absorption cycle.
2. Each component is constructed with locally available materials.
3. COP of the cooling unit is measured and compared with design COP.
4. The COP lower than the designed COP.
5. Some performance test is done to evaluate performance of the system.
6. Cooling unit offers a better environment.
7. A Fair amount of waste energy is used for their operation.

## NOMENCLATURE

| | | |
|---|---|---|
| $Q_e$ | : | Capacity |
| *SHGF* | : | Solar Heat Gain Factor |
| *ETD* | : | Effective Temperature Difference |
| $T_{10}$ | : | Evaporator temperature, °C |
| $T_4$ | : | Generator solution exit temperature, °C |
| $X_1$ | : | Weak solution mass fraction |
| $X_4$ | : | Strong solution mass fraction |
| $T_3$ | : | Heat exchanger exit temperature, °C |
| *T7* | : | Generator vapor exit temperature, °C |
| $\rho$ | : | Density, kg/m$^3$ |
| $m_{11}$ | : | Liquid carryover from evaporator |
| *P* | : | Saturation pressure, kPa |
| *h* | : | Enthalpy |
| *W* | : | Pump work, kJ |
| $Q_a$ | : | Absorber heat rejected, kJ |
| $Q_g$ | : | Heat input to the generator, kJ |
| *Q* | : | Heat transfer rate, kJ/sec |
| $\Delta Tm$ | : | Mean temperature difference |
| *A* | : | Total heat transfer area, m$^2$ |
| *U* | : | Overall heat transfer coefficient, W/m$^2$-°C |
| $\Delta T_{ln}$ | : | Logarithmic mean temperature difference |
| *Re* | : | Reynolds number |
| *Pr* | : | Prandtl number |
| $\delta$ | : | Film thickness |
| *F* | : | Correction Factor for Heat Exchanger |

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