A Study on the Effects of System Pressure on Heat and Mass Transfer Rates of an Air Cooler

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In the present paper, the effects of inlet pressure on the heat and mass transfer rates of an air cooler are numerically predicted by a local analysis method. The pressures of the moist air vary from 2 to 4 bars. The psychrometric properties such as dew point temperature, relative humidity and humidity ratio are employed to treat the condensing water vapor in the moist air when the surface temperatures are dropped below the dew point. The effects of the inlet pressures on the heat transfer rate, the dew point temperature, the rate of condensed water, the outlet temperature of air and cooling water are calculated. The condensation process of water vapor is discussed in detail. The results of present calculations are compared with the test data and shows good agreements.

Key Words: Air Cooler, Heat and Mass Transfer Rate

Nomenclature

A	: Heat transfer area or area (m^2)
C	: Dimensional parameter, see Eq. (2)
C_p	: Specific heat (kJ/kg℃)
D_h	: Hydraulic diameter of tube (m)
f	: Fouling coefficient (m^2℃s/kJ)
H	: Length of fin (m)
h	: Heat transfer coefficient (kJ/s/m^2)
h_d	: Mass Transfer coefficient (kg/s/m^2)
I, K	: Bessel function
i, j	: Index
i_{fg}	: Latent heat of vaporization (kJ/kg)
$J(s)$, $J_i(s)$	: Modified j-factors
k	: Thermal conductivity (kJ/s/m/℃)
L_{ent}	: Entrance length of tube (m)
l	: Equivalent length of fin (m)
m, n	: Number of elements in x and y directions
Pr	: Prandtl number
$\dot{q}$	: Heat transfer rate (kJ/s)
r_1, r_2	: Inner and outer radius of circular fin (m)
Re	: Reynolds number
s	: Fin pitch (m)
T	: Hot air temperature
t	: Cold water temperature
U	: Overall heat transfer coefficient (kJ/s/m^2/℃)
V	: Mean velocity (m/s)
W	: Humidity ratio (kg/kg) or width of continuous fin (m)
x	: Coordinate
y	: Fin spacing (m) or coordinate
Δx, Δy	: Size of element (m)
δ	: Fin thickness (m)
δQ	: Local heat transfer rate at element (kJ/s)
$\delta \dot{m}$	: Local condensation rate of water vapor (kg/s)
η	: Fin efficiency

Subscript

0, 1	: Order of Bessel function
c	: Cold water
cs	: Cross sectional area
$cont$	: Fin-tube contact

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f	: Fin
f, i	: Fin in sector
h	: Hot air
i, j	: Index
s	: Fin spacing or overall surface
w	: Heat transfer surface

1. Introduction

As the high power of marine engines has been continuously on demand, more air is required into the engine for combustion of fuel and one way of increasing air flow is a turbo-charger which compresses the inlet air by expanding the exhaust gas. The compressed air is then cooled by an air cooler in order to increase the mass flow rate. The thermal duty of the air cooler is increased with engine power, while installation space of an air cooler is restricted.

A typical air cooler of marine engines is a cross-typed heat exchanger with high temperature effectiveness. Among the parameters which affect the heat transfer rate of air cooler, the system configuration can reduce the heat transfer rate due to nonuniform inlet velocity profiles. The condensation characteristics of the compressed moist air also affects the heat and mass transfer rates.

As the air is compressed, the temperature of air increases and the dew point temperature is raised also due to higher partial pressure of water vapor. As the compressed air passes through the air cooler surface where the temperature is lower than the dew point temperature, water vapor begins to condense on the surface of the heat exchanger. The condensation process on the air cooler surface involving latent heat transfer makes it complicated to predict the heat transfer rate of the air cooler. The condensed water falls down over the vertical or inclined surface and corrodes the surface of heat exchanger. It is, therefore, important to predict the region where water vapor condenses for the purpose of maintenance.

With the existing methods of LMTD and ε-NTU(Holmann, 1992), it is difficult to predict the heat transfer rate of the air cooler in the case of latent heat transfer. The enthalpy method also has limitations because water vapor condenses in the local area of the heat exchanger.

Jung et al.(1991) introduced a local analysis method which divided the heat exchanger into many elements and solved the governing heat transfer equations. They derived a temperature correction method to solve the linear equations which satisfied energy balance at each element and their results showed a good agreement whena compared with a published data.

The local analysis method has many advantages. It saves computational time as well as it can treat the local variations of overall heat transfer coefficients and flow conditions. Jung(1994) also reported that a nonuniform inlet velocity profile reduced the heat transfer rate of a heat exchanger. It is, however, noted that the past studies can be applied only to the cases of sensible heat transfer.

Jung and Hong(1997) developed a local analysis method to treat the latent heat transfer. They calculated the effects of atmospheric humidities on heat transfer rates and concluded that due to the latent heat transfer the outlet temperature of the air cooler was increased.

Hill and Jeter(1991) evaluated the performance of heat exchangers. They analyzed both the dry and wet coils by varying the temperature of inlet air and heat exchanger surface. Hu et al. (1994) examined the effects of water droplets on the heat transfer characteristics over the heat exchanger surface. These studies were focused on the atmospheric pressure. In the present study, the effects of inlet pressure on the heat and mass transfer rates of the air cooler are predicted and the associated condensation process is discussed. The predictions of the model are also compared with the test data.

2. Basic Equations

Water vapor of moist air can condense at a local region of heat exchanger surface with the combined heat and mass transfer. At the other region without condensation only sensible heat is transferred. The analysis method must be able to model both cases.

2.1 Air-side heat transfer

The heat flux from moist air is expressed by the following equation when water vapor condenses.

$$\frac{\dot{q}}{A_h}=h_h(T-T_w)+h_d(W-W_w)\,i_{fg} \qquad (1)$$

The parameter C, which relates the temperatures and humidity ratios of the bulk moist air and the heat transfer surface, is defined by the following equation.

$$C=\frac{W-W_w}{T-T_w} \qquad (2)$$

By combining Eqs. (1) and (2), the heat flux from the moist air as a result of combined heat and mass transfer is expressed as the following equation.

$$\frac{\dot{q}}{A_h}=(h_h+h_dCi_{fg})\,[T-T_w] \qquad (3)$$

The first term in the parenthesis of the right hand side of Eq. (3) is a convective heat transfer coefficient and the second term is an equivalent quantity resulted from condensation of water vapor. The second term becomes zero in the case of sensible heat transfer. Equation (3) can be used in both dry and wet coil.

2.2 Fin efficiency with mass transfer

The tube and the continuous fins of the air cooler are joined tightly by a pipe expander. The fin efficiency in combined heat and mass transfer is lower than the value obtained in the sensible heat transfer. Although the basic definition is unchanged, the analysis tends to be more complex and not exact. Jung(1996) developed a program to calculate the fin efficiency by a sector method which divided the continuous fins into many sectors. The fin efficiency of each sector is calculated by assuming that each sector is a circular fin as given in Eq. (4).

$$\eta_{f,i}=\frac{2}{(r_2/r_1+1)\,ml}\,\frac{I_1(mr_2)K_1(mr_1)-I_1(mr_1)K_1(mr_2)}{I_0(mr_1)K_1(mr_2)+I_1(mr_2)K_0(mr_1)} \qquad (4)$$

In Eq. (4), the effect of combined heat and mass transfer on the fin efficiency is accounted by modifying the parameter, m, as given in Eq. (5).

$$m=\sqrt{\frac{1}{\delta k}(h_h+h_dCi_{fg})} \qquad (5)$$

To get a converged fin efficiency, which is averaged by the area, the number of sectors is increased. The fin efficiency of continuous fin can be expressed by the following equation.

$$\eta_f=\frac{\sum_{i=1}^{N}\eta_{f,i}A_{f,i}}{\sum_{i=1}^{N}A_{f,i}} \qquad (6)$$

The overall surface efficiency is defined as the ratio of the actual heat transfer rate from the surface to the maximum possible heat transfer rate from the surface if the entire fin surface are maintained with the base temperature. The overall surface efficiency can be given as the following expression.

$$\eta_s=1-\frac{A_f}{A_h}(1-\eta_f) \qquad (7)$$

2.3 Overall heat transfer coefficient

The sensible and latent heat from the air raises the water-side temperature without phase change. The water-side heat transfer rate is expressed as follows.

$$\frac{\dot{q}}{A_c}=h_c(t_w-t) \qquad (8)$$

By using Eq. (3), Eq. (7) and Eq. (8) and including the air-side and water-side fouling as well as contact thermal resistance, the overall heat transfer coefficient can be written as follows.

$$\frac{1}{U}=\frac{1}{(h_h+h_dCi_{fg})\,\eta_s}+\frac{1}{h_cA_c/A_h}+f_h+f_{cont}\frac{A_h}{A_{cont}}+f_t\frac{A_h}{A_c} \qquad (9)$$

2.4 Heat and mass transfer equations at local elements

Consider a local element by Δx and Δy in Fig. 1 which illustrates a cross-type heat exchanger with both fluids unmixed. The air is cooled as the compressed air passes through the air cooler, and where the surface temperature is lower than the dew point temperature, the water vapor in the air condenses on the surface. The condensation rate of water vapor at a local element can be expressed as follows:

$$(\delta \dot{m})_{i,j}=\{h_d \eta_s (W-W_w)\}_{i,j} A_h/mn = \frac{(\rho A_{cs} V)_h}{m}(W_{i,j}-W_{i,j+1}) \tag{10}$$

Eq. (10) implies that the condensation rate of water vapor on the surface of a local element is equal to the air flow rate multiplied by the difference in humidity ratios between the inlet and outlet of the element.

Due to the sensible and latent heat transfer, the energy flow rates of both fluids which enter and exit an element are different, but the following energy flow rates must be conserved.

$$\delta Q_{i,j}-(\delta \dot{m} i_{fg})_{i,j} = \frac{(\rho C_p A_{cs} V)_h}{m}(T_{i,j+1}-T_{i,j}) \tag{11}$$

$$\delta Q_{i,j}=\frac{(\rho C_p A_{cs} V)_c}{n}(t_{i,j+1}-t_{i,j}) \tag{12}$$

$$\delta Q_{i,j}=\frac{UA_h}{mn}\left(\frac{T_{i,j}+T_{i,j+1}}{2}-\frac{t_{i+1,j}-t_{i,j}}{2}\right) \tag{13}$$

The term of left hand side of Eq. (11) indicates the sensible heat transfer rate, and four sets of equations for $\delta Q_{i,j}$, $\delta \dot{m}_w$, $T_{i,j}$, and $t_{i,j}$ are derived. There remain m number of T and W and n number of t unknown but these values can be determined from the inlet conditions. Finally all equations are in closed forms. All properties are calculated at the mean temperature of upstream and downstream of an element. Because Eqs. (11) ~ (13) are linked to each other, an iterative method is required to solve the equations. The relative error of temperature to the values of previous iteration for both fluids is set to 10^{-4} for the convergence criterion.

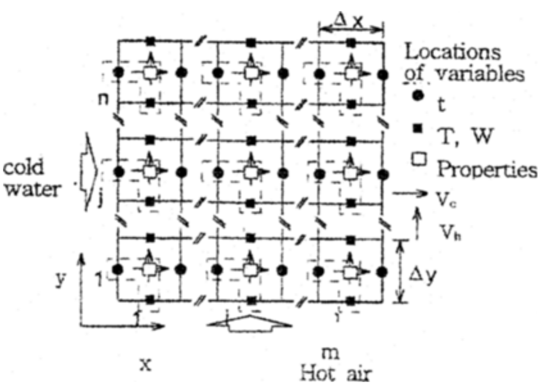


Fig. 1 Illustration of element and locations of variables for cross-type heat exchanger

3. Results and Discussions

3.1 Correlations and calculation conditions

The air cooler is a cross-counter-flow heat exchanger with two water-side passes and one air-side pass as shown in Fig. 2. McQuiston and Parker (1994) proposed a heat transfer correlation for smooth plate-fin-tube coils with four rows of tube. For film-type condensation, the modifying functions for sensible heat transfer and the total heat transfer are as follow.

Sensible j-factor;

$$J(s)=0.84+4\times 10^{-5} Re_s^{1.25} \tag{14}$$

Total j-factor;

$$J_t(s)=(0.95+4\times 10^{-5} Re_s^{1.25})\left(\frac{s}{s-y}\right)^2 \tag{15}$$

Heat transfer coefficient of dry coil is obtained by the McQuiston and Parker's correlation. For the wetted surface condition, heat and mass transfer coefficients are calculated by multiplying the McQuiston and Parker's correlation by Eqs. (14) ~ (15), respectively. A Gnielinski's correlation (Gnielinski, 1976) is used for the

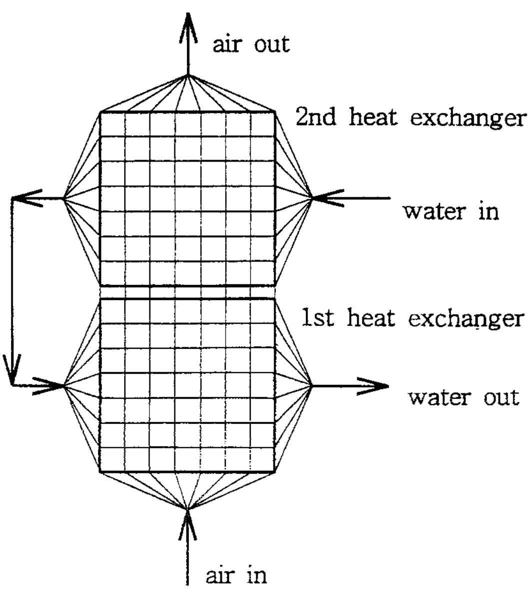


Fig. 2 Schematic diagram of an air cooler

water-side heat transfer coefficient and it is given as follows.

$$h_c=0.012\frac{k_c}{D_h}(Re_{D_h}{}^{0.87}-280)\,Pr_c^{0.4}\left[1+\left(\frac{D_h}{L_{ent}}\right)\right]^{2/3} \quad (16)$$

$$(2{,}300<Re_{D_h}<10^6,\ 1.5<Pr_c<500)$$

In the present calculation, the fouling coefficient of both fluids is assumed to be negligible and a constant contact thermal resistance of $0.086 m^2 °C s/kJ$ is used. The major calculation conditions are summarized in Table 1. The inlet temperature of the air is calculated from the relation of temperature and pressure during the adiabatic compression process with consideration of the efficiency of turbo-charger. As shown in Table 2, the higher the compression ratio, the higher the inlet temperature of air cooler.

3.2 Effects of system pressure on heat and mass transfer rates

The prediction of heat rates of the air cooler by the proposed model is compared with the test data provided by Dong Wha Precision Co., LTD, as shown in Fig. 3. The test data were obtained from the air cooler installed on the engine. The solid symbols in Fig. 3 indicate the case of condensing water vapor. The results of heat transfer rates shown in Fig. 3 are obtained from the different sets of inlet pressure, inlet temperature of water, fin pitches, outside diameter of tube and frontal area available for the actual test conditions. The comparison of the calculated heat rates and the test data shows a good agreement.

In the present study, the mass flow rate of air is assumed to be constant in order to compare the heat transfer rates with varying the inlet pressure. Table 3 shows the variations of the heat transfer rate, outlet temperature of air and condensation rate of water vapor as the inlet pressure changes. In this set of calculations the inlet temperatures are given in Table 2. The outlet temperature of air and the total heat transfer rate including latent heat transfer increases with the inlet pressure.

The results in Table 3 also show that the water vapor begins to condense at about 3 bar of inlet pressure and the condensation rate of water vapor increases rapidly as the inlet pressure

Table 1 Calculation conditions

Atmospheric temperature	27°C	Water flow rate	1055kg/s
Relative humidity	60%	Inlet temperature of water	32°C
Mass flow rate of air	26kg/s		

Table 2 Air cooler inler temperature v.s. inlet pressure

Pressure	Inlet temperature	Pressure	Inlet temperature
2.0bar	123°C	3.5bar	216°C
2.5bar	158°C	4.0bar	236°C
3.0bar	189°C		

Table 3 Calculation results for various inlet pressures

Pressure (bar)	Q(W)	Outlet temperature of air(°C)	Condensation rate of water vapor(kg/s)
2.0	2290	38.9	0
2.5	3180	41.4	0
3.0	3970	43.6	8.06×10^{-4}
3.5	4670	45.6	3.94×10^{-3}
4.0	5190	47.1	7.77×10^{-3}

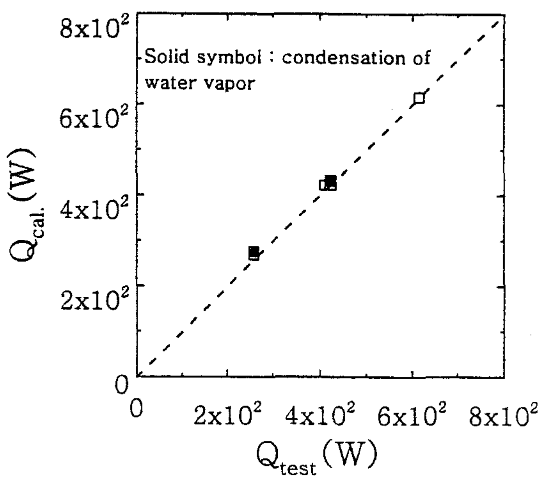


Fig. 3 Comparison of calculated heat rates with test data

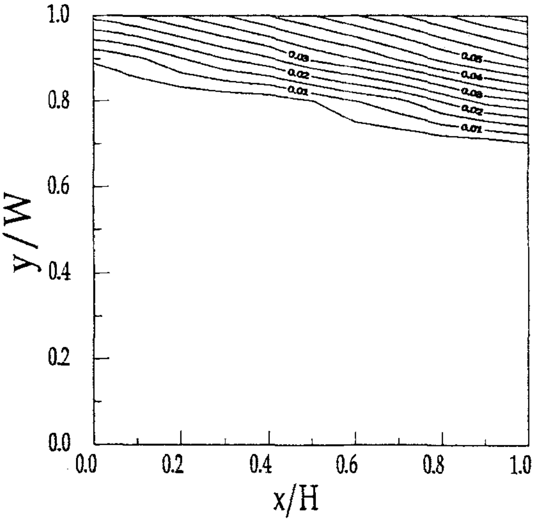
increases further. The condensation of water vapor is closely related to the dew point temperature. Therefore it is worth to discuss the effect of the pressure on the dew point temperature.

Table 4 Calculated dew point temperature v.s. inlet pressure

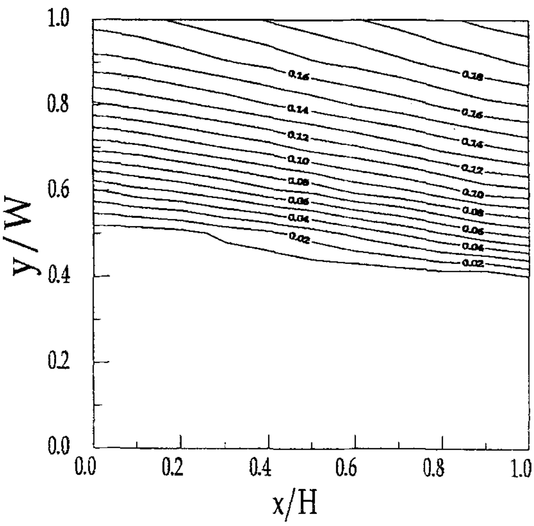
Pressure	Dew point temperature	Pressure	Dew point temperature
2.0bar	30.14℃	3.5bar	40.27℃
2.5bar	34.09℃	4.0bar	42.80℃
3.0bar	37.40℃		

The calculated dew point temperatures for various inlet pressures are shown in Table 4. The dew point temperature of moist air increases with inlet pressure, and the higher dew point temperature implies that the area of wet coil is enlarged.

Figure 4 shows the lines of constant condensation rate per unit area for two inlet pressures. These results are for the second heat exchanger because water vapor condenses only in the second

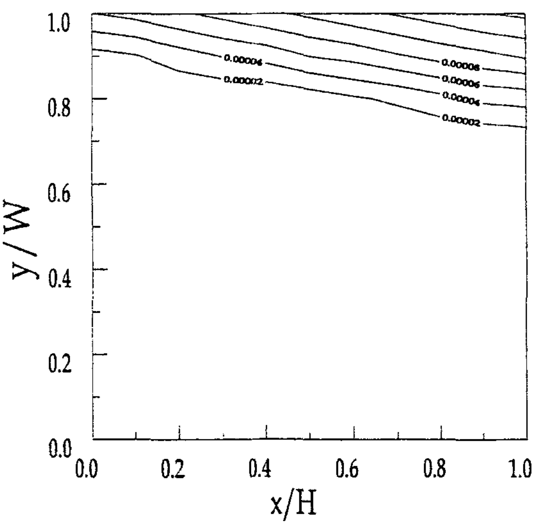


(a) At 3 bar

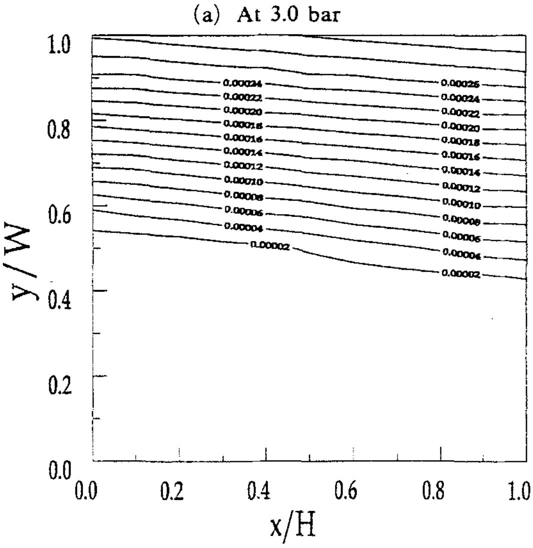


(b) At 4 bar

Fig. 4 Lines of of constant condensation rate per unit area(unit : kg of water vapor/(h·m^2))



(a) At 3.0 bar



(b) At 4 bar

Fig. 5 Lines of constant value of parameter C heat exchanger in the present calculation. It is observed in Fig. 4 that more water vapor condenses in the region of the inlet for water and the exit for air. It is also seen that for the higher inlet pressure, the area where condensation occurs is enlarged. The maximum value of condensation rate per unit area increases with inlet pressure.

The condensation rate of water vapor is related with the wall temperature and the parameter C, which is defined in Eq. (2). The variation of C due to the inlet pressure is shown in Fig. 5. This parameter is a measure of the process of moist air in the air cooler. As shown in Fig. 5, the values of C are in the range of 10^{-5} and 10^{-4} in the present calculation conditions and that the process of condensation becomes steeper at the exit of the air cooler. The maximum value of C increases with the pressure.

4. Conclusions

In the present study, the effect of the system pressure of air on the heat transfer rates of air cooler is predicted by a local analysis method. A modified model of the local analysis method to solve the combined heat and mass transfer is presented.

The discretization equations for each element were solved by an iteration method. The heat and mass transfer correlations for the air and water side heat transfer coefficients that were found in the literature were used in the calculations. The condensation process of moist air was calculated in detail by the local analysis method. From the results obtained in the present study, the following conclusions are drawn.

(1) Increment of the sensible heat transfer rate of air cooler with system pressure is due to the higher inlet temperature.

(2) Increment of the latent heat transfer rate and condensation rate with system pressure are due to the rise of dew point temperature.

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