NUMERICAL ESTIMATION OF THE CONDENSATE FLOW RATE ON THE CONDENSER PIPE

Nihal UĞURLUBİLEK[1]

ABSTRACT : *In this paper, using Fluent computational fluid dynamics program, the condensing water flow rate was numerically estimated calculating the outer surface- heat transfer coefficient of the condenser pipe. An user defined function wroten with C++ code was interpreted to Fluent program. The fluid is water, the flow is single phase, steady state and turbulent. The outer environment is saturated vapour. This method; especially, in the experimental studies thought the condensing condition, the fact that the heat transfer coefficient on the pipe's outer surface and condensate flow rate are predicted numerically and some probable ideas about the experiment can be discussed previously is able to be obtained.*

KEYWORDS : *Heat transfer, condensation, condenser, numerical.*

YOĞUŞTURUCUDA YOĞUŞMA SUYU DEBİSİNİN SAYISAL TAHMİNİ

ÖZET : *Bu çalışmada, hesaplamalı akışkanlar dinamiği programı Fluent kullanılarak, yoğuşturucu borusunun dış yüzeyindeki ısı taşınım katsayısı hesaplanarak yoğuşma suyu debisi sayısal olarak tahmin edilmiştir. Bunun için kullanıcı tanımlı bir fonksiyon C++ dilinde yazılıp programa aktarılmıştır. Yoğuşturucu içinden akan akışkan su, akış ise tek fazlı, sürekli ve türbülanslıdır. Dış ortamda doymuş su buharı vardır. Bu yöntem, özellikle, yoğuşma ortamının düşünüldüğü deneysel çalışmalarda, boru dış yüzey ısı taşınım katsayısının ve yoğuşma suyu debisinin sayısal olarak tahmin edilmesi ve deneyle ilgili bazı olası fikirlerin deney öncesinden tartışılabilmesine olanak sağlayabilmektedir.*

ANAHTAR KELİMELER : *Isı geçişi, Yoğuşma, Yoğuşturucu, Sayısal.*

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I. INTRODUCTION

Enhancement of heat transfer in heat exchangers is an important issue in terms of saving energy and material. The research of improvement the heat transfer continues today with variety of active and passive methods tested by researchers [1]. There are three factors that affect the overall heat transfer coefficient [2]. These are the convective heat transfer coefficients of the inner and outer surface of the pipe and the thermal conductivity of the pipe. It can be understood better that the effect of the improvement method on the heat transfer to change of the inner surface heat transfer coefficient. In this regard, minimizing other two factors, the outer surface heat transfer coefficient and thermal conductivity of the pipe, the effect of heat transfer improvement method is more clearly revealed. Heat exchanger tubes have generally high thermal conductivity, so its thermal resistances is very low. The outer surface convective heat transfer coefficient must be kept very high so that reduce of external surface thermal resistance. In case of boiling or condensation on the outer surface of the pipe, because the heat transfer coefficient receives very high values (2,500-100,000 W/m^2K), the effect of the outer surface thermal resistance can be minimized [2]. In this case, providing to take very small values of the convection resistance of the outer surface and thermal conductivity resistance of the pipe affecting the overall heat transfer coefficient, it is accepted that the heat transfer is only connected to the inner surface convective resistance. Thus, the effect of the method investigated on the heat transfer or the inner surface heat transfer coefficient will be more clearly understood.

In this paper, for the case of the film type condensation where the outer pipe surface, the outer surface heat transfer coefficient of the pipe was calculated numerically per unit length using FLUENTcode [3] as a computational fluid dynamics program. An user-defined function wroten in C ++ program was interpreted to the Fluent so as to calculate the outer surface heat transfer coefficient depending on the temperature of the pipe wall. The condenser pipe was set in the saturated vapour environment 328 K. The mass flow rates of the water flowed into condenser correspond to 28,000 <Re <230,000 range. The inlet water temperature was 303 K. The condenser pipe was made of aluminum material. This method was used for the case of bubble pool boiling for another study well [4]. The condensing water flow rate was calculated based on the outer surface heat transfer coefficient estimated numerical method.

II. METHOD

In this study, the flow in the condenser pipe was accepted steady and turbulent. The working fluid is taken as the water. The governing equations of incompressible fluid flow are given as;
Continuity equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}$$

Momentum equation in x-direction:

$$\frac{\partial(uu)}{\partial x} + \frac{\partial(uv)}{\partial y} = -\frac{\partial P}{\partial x} + \frac{1}{\text{Re}}\left(1 + \frac{v_t}{v}\right)\nabla^2 u \tag{2}$$

Momentum equation in y-direction:

$$\frac{\partial(uv)}{\partial x} + \frac{\partial(vv)}{\partial y} = -\frac{\partial P}{\partial y} + \frac{1}{\text{Re}}\left(1 + \frac{v_t}{v}\right)\nabla^2 v \tag{3}$$

Energy equation:

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \frac{1}{\text{Re}\,\text{Pr}}\left(1 + \frac{\alpha_t}{\alpha}\right)\nabla^2 T \tag{4}$$

where u,v, α, v and P are the velocity of the fluid in x and y direcitons, thermal diffusion coefficient, kinematic viscosity, static pressure, respectively. The standart k–ε model [5] was used to solve turbulent flow.

The equations for k and ε are given as:

$$u\frac{\partial k}{\partial x} + v\frac{\partial k}{\partial y} = \frac{1}{\text{Re}}\left(\frac{v_t}{\sigma_k}\right)\nabla^2 k + p - \varepsilon \tag{5}$$

$$u\frac{\partial \varepsilon}{\partial x} + v\frac{\partial \varepsilon}{\partial y} = \frac{1}{\text{Re}}\left(\frac{v_t}{\sigma_\varepsilon}\right)\nabla^2 \varepsilon - C_1 S_\varepsilon - \rho C_2 \frac{\varepsilon^2}{k} \tag{6}$$

where p,k, ε, v_t are the production of turbulent kinetic energy, turbulent kinetic energy, turbulent dissipation rate, turbulent viscosity and $c_\varepsilon, c_k, C_1, C_2$ are model coefficients, respectively [3].

Uniform velocity profile is accepted at the inlet of pipe and no- slip boundary condition (i.e. $u = v = 0$) is assumed on the pipe wall. The turbulence intensity is taken as 5 % at the inlet and outlet of the pipe.

Simple algoritm [6] is used to solve the governing equations by the commercial code FLUENT [3]. The convergence criteria is accepted 10^{-3} for the all variables.

It was taken as the numerical solution region in the radial and axial direction

$0 \le y \le d/2$ and $0 \le x \le L$ respectively, due to the axial symmetry for the pipe flow (Figure1).

The diameter (d) and length (L) of the pipe were assumed 0.02 and 1m, respectively. Therefore (L / d > 10), it was said that the flow is fully developed [2].
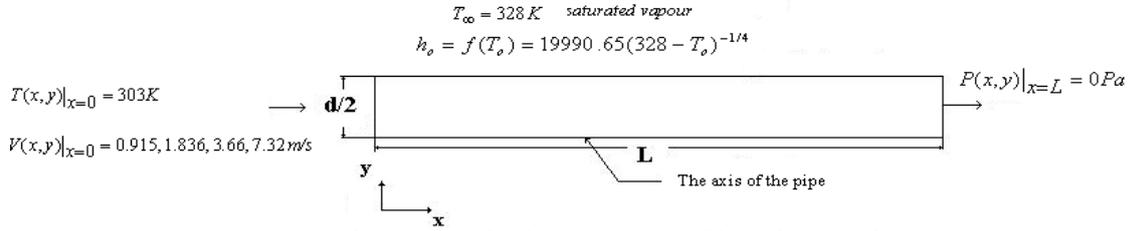
$T_\infty = 328\,K$ saturated vapour

$h_o = f(T_o) = 19990.65(328 - T_o)^{-1/4}$

$T(x,y)|_{x=0} = 303K$

$V(x,y)|_{x=0} = 0.915, 1.836, 3.66, 7.32\,m/s$

d/2

$P(x,y)|_{x=L} = 0\,Pa$

L

y

The axis of the pipe

x

Figure 1. *The numerical solution region and boundary conditions.*

Rectangular geometric pattern was made as the grid structure. Grid independent study is performed by comparing the solutions of different grid levels applied the geometry. The total number of nodes were 41,041. Because of the importance of the boundary layer, it was performed more frequent nodes near the wall (Figure 2).
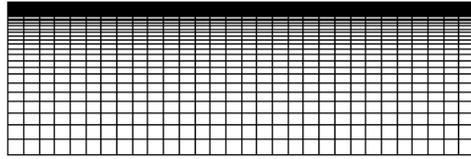
Figure 2. *The grid structure.*

The friction factor f is given by

$$f = \frac{\Delta P\,d}{\rho u_m^2/2} \tag{7}$$

and Nusselt number

$$Nu = \frac{h\,d}{k} \tag{8}$$

where ΔP, u_m and h are pressure difference between the inlet and outlet, the average inlet velocity of the water and the inner surface heat transfer coefficient of the pipe, respectively.

For turbulent flow, the friction factor [7] is

$$f = (0.7904\ln \mathrm{Re} - 1.64)^{-2} \tag{9}$$

For fully developed turbulent flow, the Nusselt number correlation [8], for heating

$$Nu = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \tag{10}$$

(for Re>10,000). The thermo-physical properties of the water used as fluid was taken at the inlet temperature (303 K).

For the inlet of condenser pipe, the temperature and velocities were taken constant and their values in the range of 28,000 <Re<230,000 are given as

$$V(x,y)\big|_{x=0} = 0.915; 1.836; 3.66 \text{ and } 7.32 \ m/s \tag{11}$$

$$T(x,y)\big|_{x=0} = 303 \ K \tag{12}$$

For the outlet of the condenser, it was adopted that the pipe outlet is open to the atmosphere. Therefore; the relative pressure is

$$P(x,y)\big|_{x=L} = 0 \ Pa \tag{13}$$

The tube material was taken the aluminum and its thermal conductivity is 203 W / mK. The water vapor ($T =328$ K) in the environment was condensed as a result of the heat transfered to the water (303 K) passed through the condenser pipe.

It can be given the convective boundary condition on the pipe outer surface

$$h_o(T_o - T_\infty) = k\frac{dT}{dn}\bigg|_{y=d/2} \tag{14}$$

where h_o and T_o are the outer surface heat transfer coefficient and temperature of the pipe, respectively. The water vapor at the ambient begins to condensate by transferring heat to the water flowed through the condenser pipe, therefore; the heat passing from the unit surface can be wroten

$$q'' = h_o(T_\infty - T_o) \tag{15}$$

according to Newton's law of cooling. The heat transfer coefficient of the outer surface of the condenser (h_o) was found by Equation (16) provided for the film type condensation [9]

$$h_o = 0.729\left[\frac{g\,\rho_l(\rho_l - \rho_v)\,k_l^3\,h'_{fg}}{\mu_l\,d}\right]^{1/4}(T_\infty - T_o)^{-1/4} \tag{16}$$

where h'_{fg} is the corrected latent heat [10]

$$h'_{fg} = h_{fg}(1 + 0.68\,Ja) \tag{17}$$

$$Ja = C_{p,l}\frac{(T_\infty - T_o)}{h_{fg}} \tag{18}$$

where Ja is the Jacobs number. In this study $Ja < 0.1$, therefore; it can be said that the correlation presented by Equation (16) is a reliable correlation for the average heat transfer coefficient [11]. Replacing to the Equation (16) the latent heat value of the saturated vapor at the ambient temperature (328 K) and thermo-physical properties of the saturated water at the film temperature (Table 1), the outer surface heat transfer coefficient (h_o) can be calculated

$$h_o = f(T_o) = 19,990.65(328 - T_o)^{-1/4} \tag{19}$$

as a function of the outer surface temperature (T_o). In this study, Equation (19) was coded in C + + code and it was numerically calculated interpreting into the Fluent program as a user-defined function.

Table 1. *Thermo-physical properties of the saturated water and the saturated vapor.*

Dynamic viscosity of the saturated water, μ_l	540×10^{-6} Ns/m^2
Density of the saturated water, ρ_l	988.142 kg/m^3
Thermal conductivity of the saturated water, k_l	0.644 W/mK
Specific heat of the saturated water, $C_{p,l}$	4,181 J/kgK
Latent heat of the saturated vapor, h_{fg}	2,370 kJ/kg
Density of the saturated vapor, ρ_v	0.1045 kg/m^3

III. RESULTS AND DISCUSSIONS

In this study, for the flow in a pipe occured the phenomenon of condensation on its surface the outer surface heat transfer coefficient was numerically calculated in the range of $28,000 < Re < 230,000$ depending on the pipe outer surface temperature using a user-defined function coded in C++ via Fluent program (Figure 3c). The pipe diameter and length are taken 0.02 and 1 m, respectively. The rectangular grid structure was applied for the domain. The standart k-ε turbulence method and Simple algoritm were used to simulate turbulent flow.
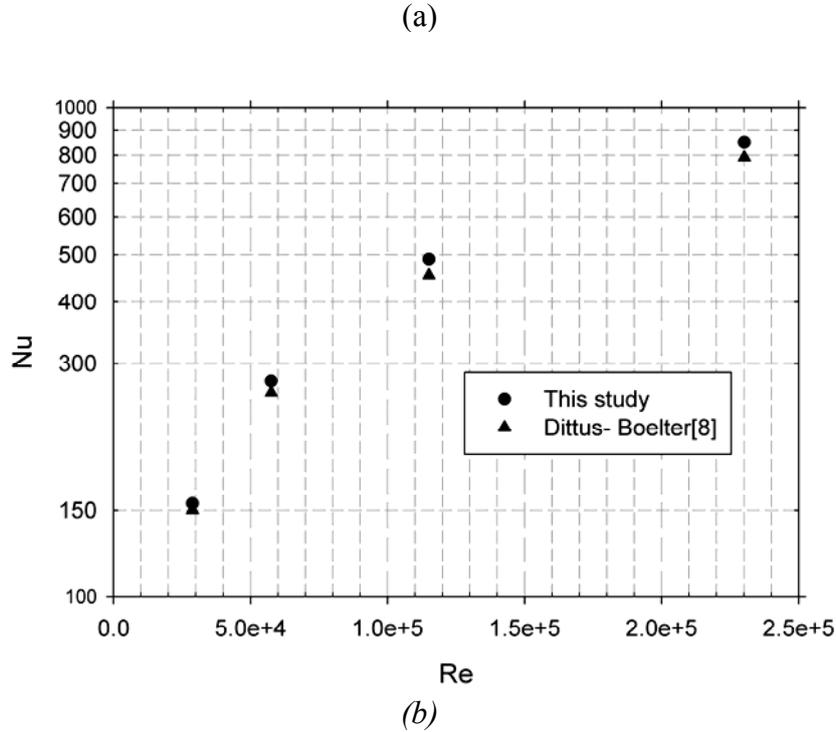
(a)

(b)
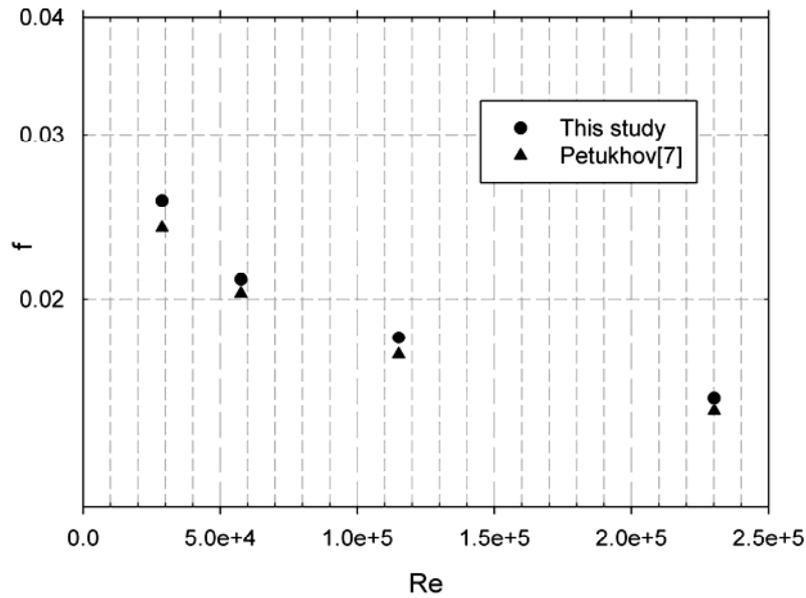
Figure 3. *Friction factor (a) and Nusselt number (b) versus Reynolds number.*

Figure 3a-b illustrates the comparison of the present prediction of the inner surface friction factor and Nusselt number with Equation (9,10) reported by Petukhov [7] and Dittus-Boelter [8]. It can be observed that the numerical results fairly agree well with the inner surface friction factor and Nusselt number derived from the Equation (9,10) (Fig. 3a, b).

Figure 4 showes the comparison of the present prediction of the outer surface heat transfer coefficient of the pipe with Equation (19) derived from Equation (16) reported by Rohsenow [9]. It can be observed that the numerical results fairly agree well with the outer surface heat transfer coefficient obtained from the Equation (19) in the range of Re=28,000-230,000. So it was checked that the user-defined function wroten by C ++ was treated correct in Fluent program (Figure 4).

For Re = 57,000; the outer surface heat transfer coefficient values obtained from Fluent program and Equation (19) calculated using the condenser surface temperature (T_o =316.9 K) obtained numerically are 10,984.1 W/m^{2}K and 10,971.7 W/m^{2}K, respectively (Figure 4).
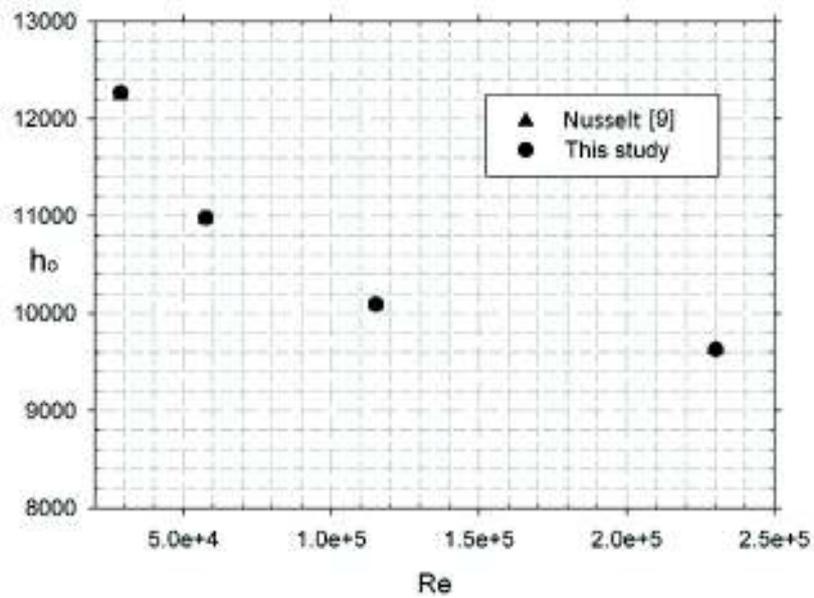
Figure 4. *Outer surface heat transfer coefficient versus Reynols number.*

For Re = 57,000; the variation of the outer surface heat transfer coefficient calculated numerically along the pipe wall was given in Figure 5.
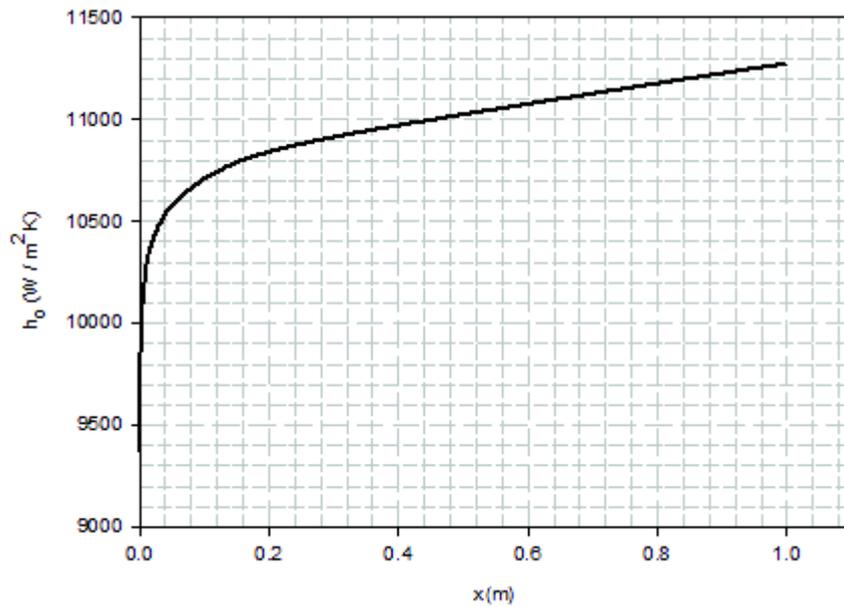

Figure 5. The outer surface heat transfer coefficient along the pipe wall (for Re=57,000).

The water flow rate condensed from the outer surface of the condenser pipe [2],

$$\dot{m} = \frac{q}{h'_{fg}} = \frac{h_o A (T_\infty - T_o)}{h'_{fg}} \tag{20}$$

It was found from Equation (20) that the water flow rate condensed on the outer surface of the condenser pipe was 0.00317 kg/s.m at the inlet temperature (303K) per unit length (Re = 57,000).

Using this method, for condensation, it was possible to calculate the outer surface heat transfer coefficient of the condenser pipe and therefore the heat transfer numerically, without experiment. Testing the accuracy of this method experimentally; it can be rendered possible in this way that estimation of the outer surface heat transfer coefficient and the condensation water flow rate , so it can be discussed before the experimental studies for the condensate case some possible ideas related to experiments.

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