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Flow and Heat Transfer Around an Air-Cooled Coil Condenser

Ahmet Cihan
Beykent University
Mechanical Engineering Department, Beykent University, İstanbul, Turkey
ahmetcihan@beykent.edu.tr

Kamil Kahveci, Alper Tezcan, Oktay Hacıhafızoğlu
Trakya University
Mechanical Engineering Department, Trakya University, 22180 Edirne, Turkey
kamilk@trakya.edu.tr; alper.tezcan@bshg.com; oktayh@trakya.edu.tr

Abstract -In this paper, flow and heat transfer around a compact coil condenser of a built-in refrigerator was studied numerically for various pitch values between the coils. Condenser in a housing was assumed to be cooled by an exhaust fan. This configuration is in accordance with a number of practical applications. Flow inside the housing was assumed turbulent. The fluid was assumed to be incompressible and Newtonian. The governing equations were solved by a computational fluid mechanics simulation software. The results show that when the pitch between the coils is increased, the heat transfer performance increases significantly. An increase of 27% in heat transfer rate is seen when the pitch value is increased from 6mm to 10mm. Further increase in the pitch value from 10mm to 12mm provides an extra 9% increase in heat transfer rate. As the increase in pitch value also increases condenser length, the pitch value of 10mm can be considered an optimum value when there is a strong restriction on space in chassis region.

***Keywords*:** built-in refrigerator, coil condenser, exhaust fan, turbulent flow, helical pitch.

1. Introduction

Performance of a compressor-based cooling system significantly depends on the condenser. The condenser is located at the rear of the appliance in conventional refrigerators and cooling of the refrigerant in the condenser occurs by natural convection flow of the air over the condenser surface. On the other hand, compressor and condenser are located together in a chassis in built-in refrigerators. Cooling of condenser is fulfilled by forced convection with a fan or blower as natural convection flow is not enough for cooling of the condenser. The air velocity varies between 2 m/s to 3.5 m/s for economic design (Singh, 2012).When sufficient heat transfer is not provided, compressor overworks and energy consumption increases. Furthermore, refrigerant temperature at compressor outlet exceeds critical value for safe work of compressor. Therefore, condenser design parameters should be carefully specified. Another factor on condenser design is space restriction as the competition between manufactures forces them to have an aggregate space as minimum as possible. This also poses problems for condenser performance as well as assembly and serviceability problems.

As it was stated before, performance of a cooling system is strongly dependent on the performance of condenser. Therefore, there are significant amount of studies in the literature on flow and heat transfer over condenser surface. Xue et al. (2012) modelled an air-cooled condenser and simulated the cooling process. Their results show that developed model can be used to formulate a control algorithm. He et al. (2012) studied the performance of an air-cooled condenser for various values of pertinent parameters and found that fan flow rate rises slightly, with the increase of ambient temperature. Yang et al. (2012) investigated the thermal-flow characteristics of wave-finned flat tube bundles in air-cooled condensers and observed that the influence of the buoyant motion on the heat transfer is negligible but the buoyancy effect on the pressure drop should be taken into consideration at low Reynolds numbers. The air-side thermal-hydraulic performance of spiral wire-on-tube condensers was investigated experimentally by Barbosa and Sigwalt (2012). Sixteen prototypes were manufactured and tested in the aforementioned study and the configuration that yields maximum heat transfer and minimum pressure drop was proposed for the usage in cooling applications. Lee et al. (2010) investigated heat transfer performance of multi-coil condensers by different coil configurations. Their results show that a better heat transfer performance can be achieved by merely changing the configuration of the coil arrangements. The results show that an increase up to by 7.85% in air flowrate and up to 5.29% in heat transfer can be achieved by a suitable configuration. Elsayed et al. (2012) made an experimental investigation on coil condenser and found that cooling COP improves as the coil diameter decreases. Wu et al. (2010) conducted a study on the effect of liquid–vapor separation on performance of an air-cooled condenser. They found that condenser with liquid–vapor separation performs better even when the heat transfer area is 37% smaller. Wang and Honda (2001) studied the effect of tube diameter and tube-side fin geometry on heat transfer performance of an air-cooled performance. They found that the tube diameter, fin length and the number of fins are effective parameters on heat transfer performance of the condenser. Parker et al. (2005) worked on development of high efficiency condenser fans and proposed a fan design which provides high air flow rates with a 26% less fan motor power.

2. Analysis

The condenser and exhaust fan considered in the study are shown in Fig.1. The condenser is a triple helix coil condenser and it is inside of a housing of diameter 170mm. The length of the housing is 140mm for the pitch of 6mm, 175mm for the pitch of 8mm, 210mm for the pitch of 10mm, and 245mm for the pitch of 12mm. Total length of the condenser tube is approximately 26m. The outside and inside diameters of the condenser tube are 4.76 mm and 3.36mm respectively. The coil diameters are 100, 130, and 160mm. The total number of turns of each helix is 17. Fan produces a mass flow rate of 0.035kg/s. Pressure jump along the fan changes according to $\Delta p=38- -14.94722u_n$. This relation was obtained from the performance curve of the fan.

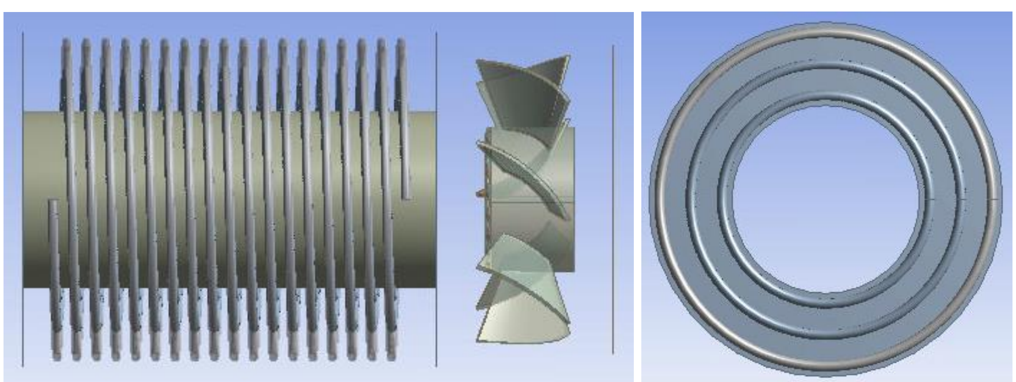

Fig. 1. Configuration of condenser-fan system.

Turbulent flow over the condenser surface has been modelled by standard k-ε model. The governing equations for this turbulent model are as follows:

Continuity Equation

$$\frac{\partial \rho}{\partial t}+\frac{\partial}{\partial t}\left(\rho u_j\right)=0 \quad (1)$$

where ρ is the density, u is the velocity.

Momentum Equation

$$\frac{\partial \rho u_i}{\partial t}+\frac{\partial}{\partial x_j}\left(\rho u_i u_j\right)=-\frac{\partial p'}{\partial x_i}+\frac{\partial}{\partial x_j}\left[\mu_{eff}\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)\right]+S_M \tag{2}$$

where S_M is the sum of body forces. μ_{eff} is the effective viscosity counting the turbulence. p' is the modified pressure defined as:

$$p'=p+\frac{2}{3}\rho k+\frac{2}{3}\mu_{eff}\frac{\partial u_j}{\partial x_j} \tag{3}$$

The k-ε turbulence model is based on the eddy viscosity concept.

$$\mu_{eff}=\mu+\mu_t \tag{4}$$

where μ_t is the turbulence viscosity. In the k-ε model, the turbulence viscosity is assumed to be linked to the turbulence kinetic energy and dissipation by the following relation:

$$\mu_t=C_\mu\rho\frac{k^2}{\varepsilon} \tag{5}$$

where C_μ is a constant, k is the turbulence kinetic energy, and ε is the dissipation rate.

The following transport equations can be written for turbulence kinetic energy and dissipation rate:

$$\frac{\partial(\rho k)}{\partial t}+\frac{\partial}{\partial x_j}\left(\rho u_j k\right)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+P_k-\rho\varepsilon+P_{kb} \tag{6}$$

$$\frac{\partial(\rho \varepsilon)}{\partial t}+\frac{\partial}{\partial x_j}\left(\rho u_j \varepsilon\right)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]+\frac{\varepsilon}{k}\left(C_{\varepsilon 1}P_k-C_{\varepsilon 2}\rho\varepsilon+C_{\varepsilon 1}P_{\varepsilon b}\right) \tag{7}$$

where $C_{\varepsilon 1}$, $C_{\varepsilon 2}$ are constants. σ_k and σ_ε are the turbulent Prandtl numbers for k and ε. P_{kb} and $P_{\varepsilon b}$ represent the effect of the buoyancy forces and they are defined as:

$$P_{kb}=-\frac{\mu_t}{\rho\sigma_\rho}g_i\frac{\partial \rho}{\partial x_i} \quad , \quad P_{kb}=-\frac{\mu_t}{\rho\sigma_\rho}\rho\beta g_i\frac{\partial T}{\partial x_i} \quad , \quad P_{\varepsilon b}=C_3 max(0,P_{kb})sin\emptyset \tag{8}$$

where T is the temperature, σ_ρ is the turbulence Schmidt number. It is $\sigma_\rho=0.9$ for Boussinesq model and $\sigma_\rho=1$ for full buoyancy model. Dissipation coefficient is $C_3=1$.

P_k is the turbulence production due to viscous forces, and it is modelled as:

$$P_k=\mu_t\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)\frac{\partial u_i}{\partial x_j}-\frac{2}{3}\frac{\partial u_k}{\partial x_k}\left(3\mu_t\frac{\partial u_k}{\partial x_k}+\rho k\right) \tag{9}$$

Energy Equation

$$\frac{\partial(\rho h_{tot})}{\partial t}-\frac{\partial p}{\partial t}+\frac{\partial}{\partial x_j}\left(\rho u_j h_{tot}\right)=\frac{\partial}{\partial x_j}\left(\lambda\frac{\partial T}{\partial x_j}-\rho\overline{u_j h}\right)+\frac{\partial}{\partial x_j}\left[u_i\left(\tau_{ij}-\rho\overline{u_i u_j}\right)\right]+S_E \tag{10}$$

where λ is the thermal conductivity coefficient, τ is the stress tensor, $\rho\overline{u_i u_j}$ are the Reynolds stresses, and S_E is the heat source. The mean total enthalpy h_{tot} is defined as:

$$h_{tot}=h+\frac{1}{2}u_i u_i+ \tag{11}$$

The total enthalpy contains a contribution from the turbulent kinetic energy defined as:

$$k = \frac{1}{2}\overline{u_i^2} \tag{12}$$

The turbulence parameters were selected as (Launder and Spalding, 1972):

$$C_\mu = 0.09 \ \ , \ \ C_{\varepsilon 1} = 1.44 \ \ , \ \ C_{\varepsilon 2} = 1.92 \ \ , \ \ \sigma_k = 1.0 \ \ , \ \ \sigma_\varepsilon = 1.3 \tag{13}$$

In this study, temperature of the condenser walls were assumed to be at 37°C Temperature along the condenser was assumed to be constant. This is a reasonable assumption because the working fluid generally becomes saturated in a short distance through the condenser tubes.

3. Results and Discussion

Flow and heat transfer over condenser surface has been investigated for four different helical pitch values, 6, 8, 10, and 12mm. Flow domain was meshed by triangular mesh elements. More mesh elements were used in the regions where the large velocity and temperature gradients were expected to develop. The computations were carried by the CFD solver FLUENT based on the finite volume approach. Convergence criteria were taken as 10^{-4} for all dependent variables in the study.

The velocity and temperature fields for different pitch values are shown in Figures 2-5. As it is well known, cross flow over cylinder or tube surfaces exhibit complex flow patterns. The fluid approaching the tube branches out and encircles the tube and boundary layers are formed along the top and bottom surface of the tube. The fluid particles on the mid-plane strike the tube at the stagnation point and fluid comes to a complete stop and an increase in the pressure is seen at that point. The pressure decreases in the flow direction while the fluid velocity increases. At relatively high velocities, boundary layer detaches from the surface, forming a separation region behind the tube. Flow in the wake region is characterized by vortex formation and pressures much lower than the stagnation point pressure. For relatively low Re numbers, which is the case in this study, heat transfer in the wake region is lower than the front side of the tube because of the recirculating region at the backside of the tube with lower velocities. When the pitch gets smaller, only small recirculating zones are seen behind the tubes because of the strong viscous forces. As a result, a region with very low velocities and high temperatures is present between the tubes as sein in figures. When the pitch increases, the recirculation behind the cylinder gets stronger. This is clear from the higher velocities and lower temperatures observed in figures. As it is also seen from the figures, velocities take higher values between the spirals because of the decrease in flow area. Velocity along the condenser increases in the core region between the spirals as the viscous forces along the tubes gets higher values.

The average temperature at the outlet of the condenser are given in Table 1 for various pitch values. Table 1 also contains heat transfer rate from condenser and percentage increase in heat transfer rate when pitch value is increased. As it can be observed, average temperature first shows a significant increase when the pitch value is increased. Increase in average temperature slows down when the pitch value is further increased. When pitch is increased from 6mm to 10mm, increase in heat transfer rate reaches up to approximately 27%. Further increase in pitch value from 10mm to 12mm provides an extra 9% increase. Increase in pitch value also increases condenser length. This is undesirable because of the space restriction in chassis. Therefore, the pitch value of 10mm can be considered an appropriate pitch value if the space restriction forces a lower condenser length.

4. Conclusion

Flow and heat transfer around a compact coil condenser of a built-in refrigerator was studied numerically for various pitch values between the coils in this study. The results show that the pitch between the coils is an important parameter in heat transfer performance of the coil condenser. The results show that heat transfer rate increases 27% when the pitch value is increased from 6mm to 10mm. Further increase in the pitch value from 10mm to 12mm supplies an extra 9% increase in heat transfer rate. As the increase in pitch value also increases the condenser length, the pitch value of 10mm can be considered an optimum value when there is a strong space restriction in chassis region.

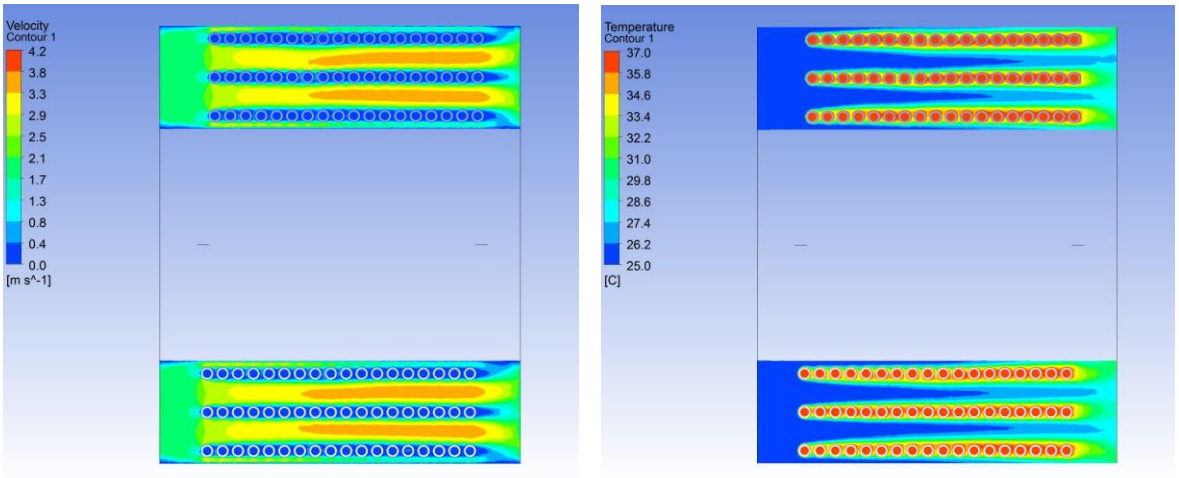

Fig. 2. Velocity and temperature fields for the pitch value of 6mm.

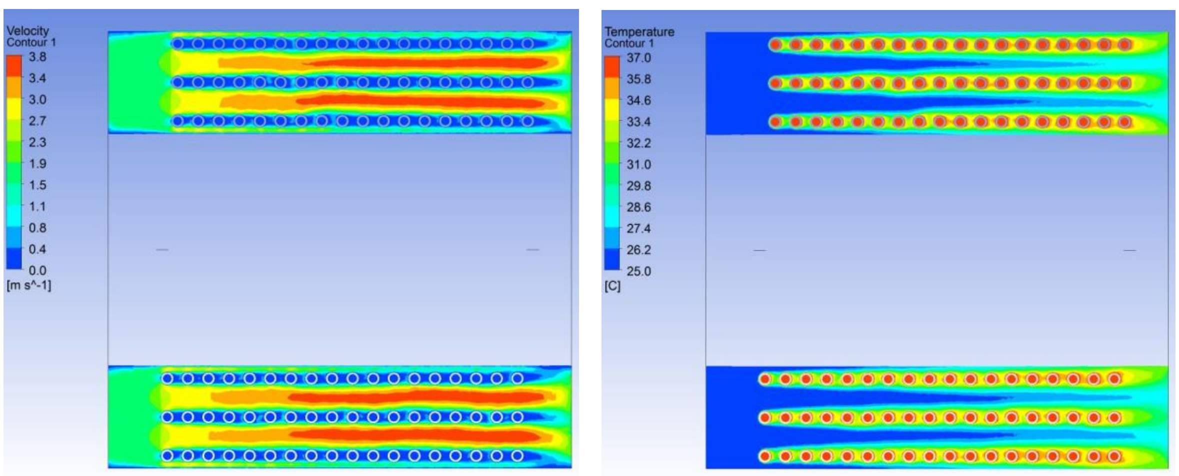

Fig. 3. Velocity and temperature fields for the pitch value of 8mm.

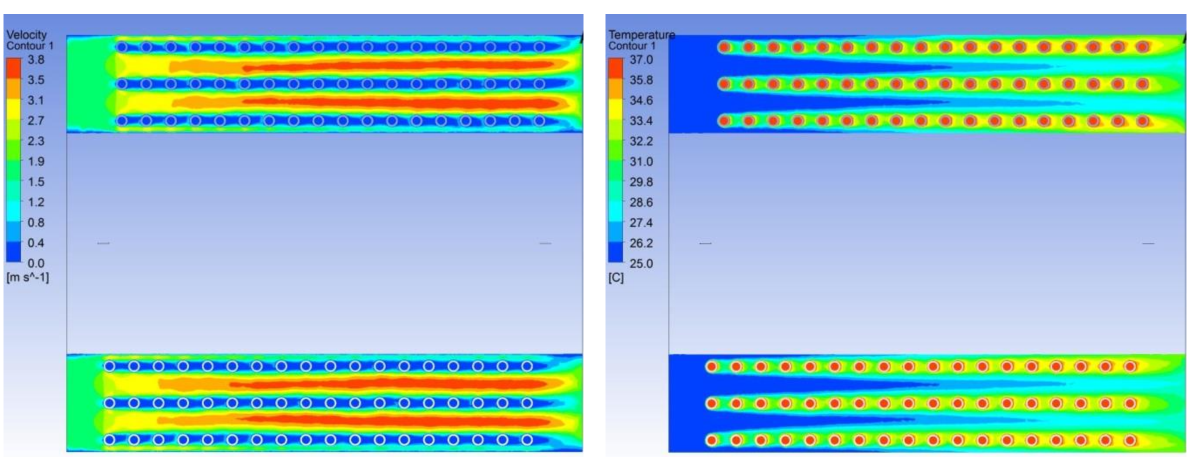

Fig. 4. Velocity and temperature fields for the pitch value of 10mm.

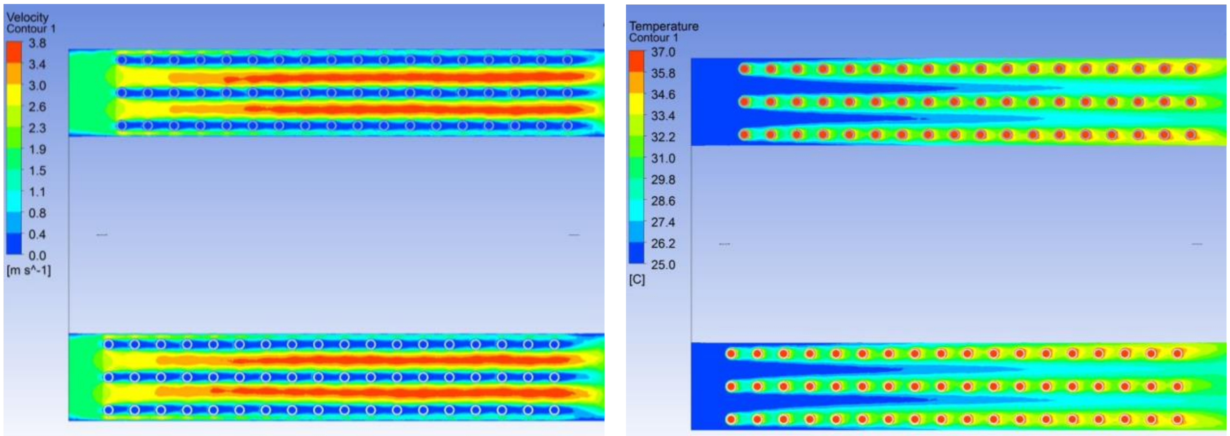

Fig. 5. Velocity and temperature fields for the pitch value of 12mm.

Table. 1. Performance of the condenser for various values of the pitch between the coils.

Pitch (mm)	T_{ave} (°C)	Q (W)	% increase
6	28.8667	0.136011173	-
8	29.5037	0.158417648	16.47
10	29.9359	0.173620283	27.65
12	30.2666	0.185252655	36.20

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