Analysis of a Direct Current Compressor for Solar Cooler

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Abstract: Refrigeration systems have higher electricity consumption ratio in overall energy consumption especially in household utilization. On the other hand, these systems have an advantage of being powered by renewable energy sources such as solar or solar-wind hybrid systems because of their lower power demand. Since the efficiency improvement of refrigeration systems is crucial for domestic and industrial systems in the present study, solar energy powered compressor for vapour compression refrigeration systems have been investigated. Compressor is the most important part and often the costliest component of any refrigerant system. There are two different options for powering the compressor such as alternative current (AC) and direct current (DC).

For that reason, this study focused on thermal analysis of critical component inside the novel DC compressor. Computational Fluid Dynamics (CFD) analysis has been realized to figure out the heat transfer mechanism inside the compressor components and also thermodynamics efficiency. Temperature distribution in the compressor during the operation was presented and the component of the compressor position re-designed with resulted parameter and discussed affects of the volumetric efficiency.

Furthermore, in this study, experimental performance analysis of the novel DC type refrigeration compressor implemented in a 50 l refrigerator to show its cooling performance and compare well known DC compressor in the market. Energy usage reduction and operational improvement potential of the solar powered DC compressor via variable speed operation were investigated. The comparison showed that variable speed operation of the novel DC compressor can be much more efficient than constant speed operation.

Keywords: Computational Fluid Dynamics, Efficiency, Direct Current Compressor, Renewable Energy.

INTRODUCTION

Air conditioning is defined as heating, cooling, humidification (or dehumidification) and ventilation with the requirement of any space. In buildings, air conditioning systems are the most energy consumed systems. Recently, increasing energy bills of air conditioning also increases solar energy utilization on these systems. Because as a sources of renewable energy, solar energy is the most suitable system for air conditioning especially in countries of the midlatitude and sub-tropical zones. A lot of studies treated the solar refrigeration especially in tropical and Sunbelt countries.

Modi *et al*. [1], achieved a COP of 2.102 for a solar powered Alternating current (AC) refrigerator at ambient temperature around 42°C. Ekren *et al*. [2], studied AC and DC compressor experimentally and concluded that DC compressor could be much more efficient than AC compressor. Furthermore, variable speed DC compressor provided more than 4% improvement in energy efficiency, and more than 7% improvement in COP compared to constant speed. The solar driven DC refrigeration transportable system and optimization of the system components was studied in El-Bahloul *et al*. [3].

The compressor is the most important part and often costliest component of any refrigerant system. Therefore, compressor needs to be designed well for efficiency improvement. A hermetically sealed reciprocating compressor consist of a compressor shell, crank-shaft, mufflers, compressor chamber and stator winding. The overall efficiency depends upon electrical efficiency, mechanical efficiency and thermodynamic efficiency [4]. The heat transfer phomenon in compressor change attributes of component and refrigerant. The compressor volumetric efficiency descriped as,

$$\eta_V = \frac{VolumetricFlowRate}{CompressorDispalcementRate} = \frac{\dot{m}\, v_e}{\dot{V}_{SW}} \tag{1}$$

The suction temperature effect volumetric efficiency. The reduction of the suction temperature at suction increases 6-10% compressor efficiency [5]. Otherwise, the gas heats up in the suction muffler when enterd into the piston. The refrigerant gas in discharge system need eject heat to the surronding by the compressor shell. Thermal energy analysis of a reciprocating compressor are determined by Todescat *et al*. [6]. The analytical and experimental modelling developed for prediction of temperature changing [7]. The temperature difference also leads to thermal stress at important part such as motor winding. Therefore, thermal analysis of the compressor is critical on the other hand measuring every component or location of the refrigerant is diffucult [8]. To overcome this

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problem, CFD programs has been used and thermal distrubution on the compressor components and refrigerant can be defined. CFD analysis is powerful and reliable but still needs experimental validation.

METHOD

Computational Fluid Dynamics (CFD) is the science of prediction fluid flow, heat and mass transfer, chemical reaction, and related phenomena. Generaly in CFD analysis three basic equations such as conservation of mass, momentum and energy have been solving.

Conservation of mass (Continuity) equation:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \quad (2)$$

Conservation of momentum equations:

$$\frac{\partial}{\partial t}(\rho u)+\frac{\partial}{\partial x}(\rho uu)+\frac{\partial}{\partial y}(\rho uv)+\frac{\partial}{\partial z}(\rho uw)= -\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}\left[\frac{\partial}{\partial x}(\mu u)\right]+\frac{\partial}{\partial y}\left[\frac{\partial}{\partial y}(\mu u)\right]+\frac{\partial}{\partial z}\left[\frac{\partial}{\partial z}(\mu u)\right] \quad (3)$$

$$\frac{\partial}{\partial t}(\rho v)+\frac{\partial}{\partial x}(\rho vu)+\frac{\partial}{\partial y}(\rho vv)+\frac{\partial}{\partial z}(\rho vw)= -\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}\left[\frac{\partial}{\partial x}(\mu v)\right]+\frac{\partial}{\partial y}\left[\frac{\partial}{\partial y}(\mu v)\right]+\frac{\partial}{\partial z}\left[\frac{\partial}{\partial z}(\mu v)\right] \quad (4)$$

$$\frac{\partial}{\partial t}(\rho w)+\frac{\partial}{\partial x}(\rho wu)+\frac{\partial}{\partial y}(\rho wv)+\frac{\partial}{\partial z}(\rho ww)= -\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}\left[\frac{\partial}{\partial x}(\mu w)\right]+\frac{\partial}{\partial y}\left[\frac{\partial}{\partial y}(\mu w)\right]+\frac{\partial}{\partial z}\left[\frac{\partial}{\partial z}(\mu w)\right] \quad (5)$$

Conservation of energy equation:

$$\frac{\partial}{\partial t}(\rho cT)+\frac{\partial}{\partial x}(\rho ucT)+\frac{\partial}{\partial y}(\rho vcT)+\frac{\partial}{\partial z}(\rho wcT)= \frac{\partial}{\partial x}\left[\frac{\partial}{\partial x}(kT)\right]+\frac{\partial}{\partial y}\left[\frac{\partial}{\partial y}(kT)\right]+\frac{\partial}{\partial z}\left[\frac{\partial}{\partial z}(kT)\right] \quad (6)$$

Considering the computer resource and time the problem has been divided into sub-domain such as refrigerant, oil, and heat conduction of compressor shell domain. The electircal motor of any refrigeration compressor is in the oil because of cooling requirement of the motor windings. Therefore, oil domain is explicitly modelling as a thermal boundary condition. The refrigerant and oil are assumed as steady state and no heat transfer. In addition, every walls were assumed smooth and unnecessary small components which were not affect fluid flow such as fillets were removed.

Following assumption were made for the analysis:

- Steady state flow conditions.
- The refrigerant properties were considered as piecewise linear function and value taken ASHRAE [9].
- The heat losses from stator and motor winding assumed Neumann boundary condition.
- The radiation heat transfer neglected.
- The refrigerant assumed as an incompressible fluid.

The fluid domain has very complexity shape, this reason discretized with tetrahedral elements as shown in Figure **1**. The elements skew and aspect ratio was checked. Bounday conditions were choosen velocity inlet and pressure outlet. These values were taken from experimental results. For grid independence, the grid size increased until the converging criterias which was average temperature and pressure drop at inlet (taken from experiments).

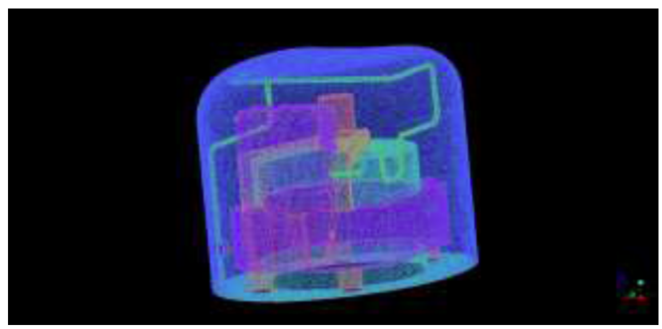

Figure 1: Tetrahedral Grid Elements.

The standart k-e turbulence model selected for widelly using engineering application and reasonable accuracy. The coupled formulation solved the governing equations simultaneously. In addition, Realizable k-e turbulence model was chosen for complex shear flow involving rapid change and compare the two turbulence model with experimentals results. Second order upwind scheme was used for whole iteration. The solution converged predicted average temperature at compressor shell and pressure drop at compressor inlet.

RESULTS AND DISCUSSION

The validation of the numerical study was compared the experimental results. The experimental set up consist of two type of the compressors such as a prototype DC (manufactured EBB Energy Co.) and a commercial DC (BD50 manufactured by Danfoss Inc). The temperature measurements on the solar powered compressor surface shown in Figure **2**. These two compressors were compared experimentally. Cooling capacity of the compressors were 1/10 hp and volumetric compression capacities were 13 cm^3.

Figure 2: Experimental setup for the compressor.

The results of the experimental study given in Table **1**. The temperature, pressure, mass flow rate and power consumption values are measured. However, the numerical results were compared with the experimental mean temperatures at the compressor surface.

The measured temperature at the inlet of the compressor was -4.5 ^{o}C, at the outlet was 66.2 ^{o}C. In addition, the temperature of the compressor surface (top) was 30.8^{o}C. The Figure **4** shows numerical results of temperature distribution on the compressor surface. The basic system components of a solar powered cooling system were summarized in Daut *et al*. [10].

The most important parameter of the reciprocating compressor efficiency are the mas flow rate for the volumetric efficiency, power consumption per unit refrigerant capacity, discharge temperature and performance under part load conditions. Therefore, in this section discussion of pressure drop at inlet of the compressor, temperature at the suction muffler inlet, temperature distribution on the shell, and critical velocity isosurface were realized.

Figure **3** shows refrigerant flow path in the compressor domain. The refrigerant as superheated gas contacts suction muffler, motor, stator, and discharge line that occur heat transfer from electrical motor with adjoining satator and discharge line. Raja

Table 1: Experimental Results

Sensor #	Temperature-^{o}C	EBB Co. Prototype	Danfoss BD50
1	Inlet temperature of the compressor	-4.5	-1.2
2	Discharge temperature of the compressor	66.2	64.05
3	Surface temperature at condenser	48.8	36.35
4	Discharge temperature of the condenser	43.8	35.67
5	Inlet temperature of the evaporator	-2.9	-11
6	Refrigerator cabinet temperature	3	-4
7	Discharge temperature of the evaporator	-2.0	-10.1
8	Surface temperature of the evaporator	-4.3	-4.98
9	Top surface temperature of the compressor	30.8	47
10	Ambient temperature	24	24
Sensor #	**Pressures-bar**		
11	Inlet pressure of the compressor (abs)	2.6	1.61
12	Discharge pressure of the compressor	21.3	11.3
13	Discharge pressure of the condensor	21.0	11.05
	Pressure ratio	8.1	7.0
Sensor #	**Flow rate- l/min**		
14	Refrigerant flow rate	0.18	0.10
Sensor #	**Electrical**		
15	Compressor current-A	4.9	3.8
16	Compressor voltage-VDC	24	24

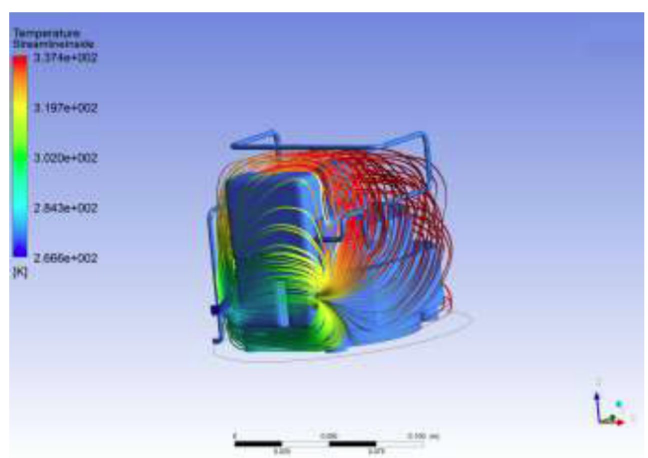

Figure 3: Stream line in entire compressor domain.

et. al.(2003) present different numerical domain that suction muffler located behind the stator. In this case, the suction muffler inlet was faced the compressor inlet. This design enhanced the volumetric efficieny of the compressor, which is related refrigerant temperature and mass flow rate in equation 1. The suction muffler inlet temperature also determines to improve efficiency. Furthermore, the temperature and termal stress were given in Figure **4** for the the critical component.

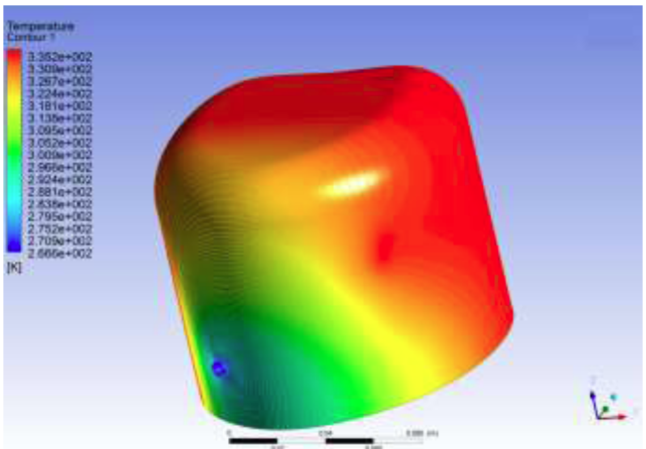

Figure 4: Temperature distribution on the compressor shell.

The refrigerant temperature is important as well as flow regime. Therefore, Figure **5** shows that the temperature distribution at above 0.001 m/s flow velocity and Figure **6** shows temperature distribution at above 0.0005 m/s flow velocity. The heat transfer phenomena depends upon refrigerant flow velocity The efficieny of the compressor component design also depend on pressure drop.

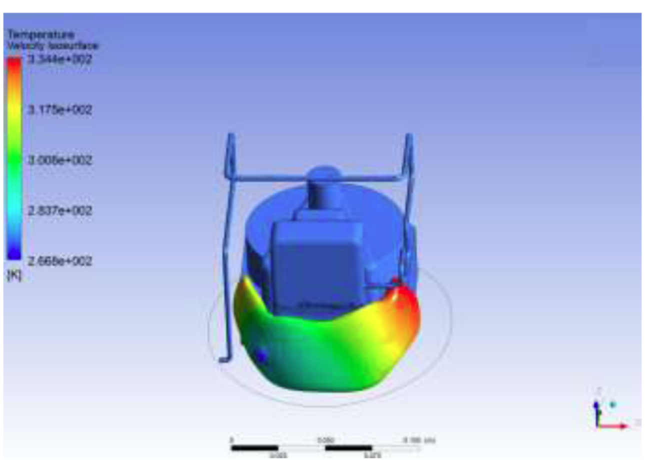

Figure 5: Temperature distribution with 0.001 m/s.

The comparsion were realized between the temperature above 0.001 m/s and 0.0003 m/s fluid velocity. The heat transfer enhanced with fluid velocity since the higher velocity descrease the motor winding temperature. On the other hand, superheating of the refrigerant causes the lower compressor volumetric efficiency.

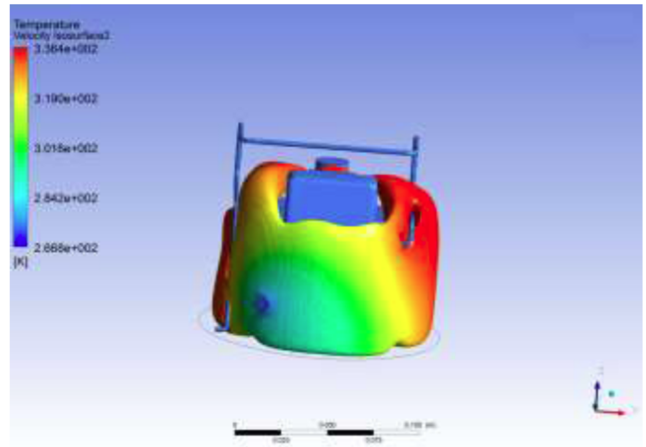

Figure 6: Temperature distribution with 0.0003 m/s.

The angular position was choosen 0^o at compressor inlet and rotates towards clock-wise direction. The temperature in the fluid varies from 295 K to 335 K. The minimum and maximum temperatures were occured at the bottom and top of the refrigerant domain, respectively. The maximum temperature was observed at the adjoining motor at the reference angle 180^o.

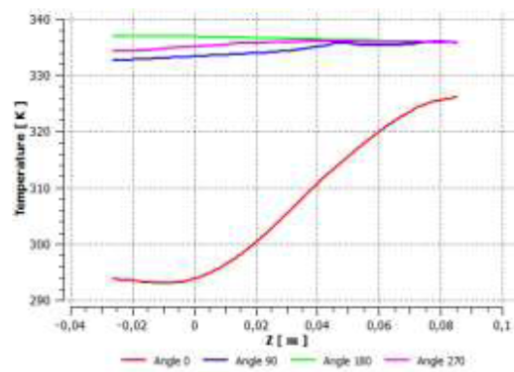

Figure 7: Variation of temperature at adjoinnig motor.

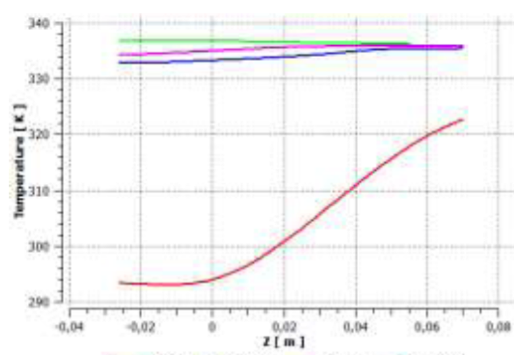

Figure 8: Variation of temperature at adjoining shell.

CONCLUSION

In this paper, an experimental comparison of two type of DC compressors such as a prototype DC (manufactured EBB Energy Co.) and a commercial DC (BD50 manufactured by Danfoss Inc) was realized. Furthermore, the numerical results were validated by using the experimental results. In the numerical study, the hermetic sealed reciprocating compressor was one of the critical component. Therefore, the heat transfer between the refrigerant and electrical windings of the compressor were investigated since it affects the compressor volumetric efficiency. Thus, the thermal profile inside the solar powered compressor described using numerical analysis and experimental results. As a results, the solar powered compressor can be solution of higher energy bills of coolers.

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NOMENCLATURE

$\dot{m}$ = mass flow rate

v_e = specific volume at inlet of compressor

$\dot{V}_{SW}$ = volumetric flow rate in the cylinder

ρ = density

∇ = Gradient operator

t = time

V = velocity vector

u = x-component of velocity

υ = y-component of velocity

w = z-component of velocity

p = Pressure

τ = shear stress

e = lattice energy

k = thermal conductivity

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