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Thermal Design Characterization of Heat Exchangers for 3 Stages Turbo Centrifugal Geared Air Compressor

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***Abstract*— *A single or multi stage turbo centrifugal geared compressor is widely used in various industrial application such as oil & gas, chemical, power, and air separation plants. It is consist of compression part including air scroll, impeller, and diffuser, gear and bearing, cooling, oil, sealing, and control systems. Shell and tube type heat exchanger commonly used to cool down compressed gas or air between stages or after final stages, and supply oil temperature as a cooler. On compressor's cooling system strongly required to compact (miniaturization) and high heat transfer capability. Increases of heat transfer capability can result reduce compression aero-power and pumping-power, enhance system stability, and compact.*

Recently, nanofluid, which is a colloidal dispersion of nano-sized particles in basefluid such as water, ethylene-glycol, propylene-glycol and their mixtures etc., exhibit a significant enhancement in their thermal conductivity. Thus, using nanofluid expected to be an advanced coolant material in compressor cooling system.

In this study, HTRI Xist used to thermal design of heat exchanger, nanofluid's thermal conductivity used to coolant material property. The thermal conductivity of the nanofluid increased by 9.3% compared to that of water. The heat exchanger's tube side overall heat transfer rate is increased by ~7% compared to that of basefluid.

Keywords*— *HTRI, Nanofluid, Shell and tube heat exchanger, Turbo centrifugal geared compressor

I. INTRODUCTION

A compressor is essential energy equipment to supply high quality air or composite gas where oil & gas, power, chemical, and air separation plant. There are three kinds of types which are reciprocating, screw, and turbo. Among them, turbo type compressor has many advantages such as flow rate, high pressure ratio, low noise, and easy maintenance capability. And it can possible to supply to high purity compressed gas cause oil-free type [1, 2]. It is consist of aero-compression (scroll, impeller, and diffuser), gear, bearing, sealing, cooling, and control systems. Cooling system is one of the vital needs in aero-compression and oil supply system. It strongly required to compact (miniaturization) and high heat transfer capability. Increases of heat transfer capability can result in reduce compression aero-power and cooling system's supply pumping-power.

It is considerable several ways to enhance heat transfer coefficient or heat transfer rate.

At first glance, it may think that for a given system, it may remove more and more energy by increasing the coolant flow rate (means need high pumping-power). Unfortunately, current designs of thermal management systems have already adopted this technology to its limits. Generally, the advance heat transfer given by [3]

$$\boldsymbol{h = (\mathrm{Nu}\ k_f) / D}$$

Where, h is heat transfer coefficient, Nu is Nusselt number, k_f is thermal conductivity of coolant, and D is system size. From the heat related formula, it is considerable way to be improved heat transfer coefficient is increasing the thermal conductivity of coolant. If thermal conductivity of coolant can increase, it is possible to obtain higher heat transfer coefficient of a given system.

Basically, heat exchanger use water, ethylene-glycol, propylene-glycol or their mixture as a coolant. The thermal conductivity of the conventional coolants is essentially fixed and difficult to achieve high-performance cooling [4].

Masuda et al. reported that a suspension of ultra-fine particles in a basefluid enhances the thermal conductivity of the fluid [5]. Also Choi reported that a suspension of nanoparticles in a basefluid, which is so called "nanofluid", enhances the thermal conductivity of the fluid [6]. Successfully employment of nanofluid should support the required facilities compact (miniaturization) by enhancing the design of much smaller lighter systems, higher energy-efficiency [7].

Recently, several researchers have been reported heat transfer characterization of shell and tube type heat exchangers [8~12]. Most of report is based on theoretical correlation, CFD analysis, and cost optimization.

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However, the effectiveness of nanofluid in compression cooling system application and its design approach is lack. In this study, 3 stages turbo centrifugal compressor designed using S. Hall's equation [13], 2 intercoolers and 1 aftercoolers designed using HTRI (heat transfer research institute Inc.). HTRI has developed a useful thermal design tool of commercial heat exchanger. A nanofluid fabricated by pulsed laser ablation in liquid (PLAL) method as a single-step method. The thermal conductivity measured by transient hot-wire method. The intercoolers and aftercooler designed using measured thermal conductivity as a fluid material property. The thermal design result of intercooler and aftercooler based on HTRI input, output and geometry arrangement.

II. THERMAL DESIGN

2.1 Compressor design

Table 1 summarized design requirement and Table 2 summarized supply cooling systems information.

Table 1 Compressor design requirement

Item		Content	Unit
Fluid		Air	-
Suction	Flow rate	1.50	kg/s
	Temperature	20.00	C
	Pressure	1.013	barA
Discharge	Temperature	42.00	C
	Pressure	6.70	barA
Humidity		60.00	%

Table 2 Supply cooling system

Item		Content	Unit
Fluid		Water Nanofluid	-
Inlet	Temperature	32.00	C
	Pressure	5.00	barA
Outlet	Temperature	42.00	C

Based on given design requirement and information, the compressor designed. Table 3 summarized design result. The overall pressure ratio is 6.61, to optimize each stages pressure ratio divided by 2.2, 2.0, and 1.5 respectively. As the result discharge pressure and temperature and aero-power are obtained.

Table 3 Compressor design result

Item		Content	Unit
Overall pressure ratio, p_r		6.61	-
Stage pressure ratio	1st	2.20	-
	2nd	2.00	-
	3rd	1.50	-
Gas constant, *R*		8,314.47	
Mole weight		28.96	
1st stage			
Discharge pressure		2.22	barA
Discharge temperature		122.71	C
Polytropic aero-power		136.32	kW
2nd stage			
Discharge pressure		4.45	barA
Discharge temperature		137.24	C
Polytropic aero-power		145.38	kW
3rd stage			
Discharge pressure		6.70	barA
Discharge temperature		94.94	C
Polytropic aero-power		80.62	kW
Mechanical loss		10.56	kW
Total power		372.88	kW

2.2 Nanofluid thermal conductivity

Fe_2O_3 nanofluid fabricated by pulsed laser ablation in liquids (PLAL) method as a single-step method [14]. A Q-switched Nd:YAG laser (wavelength: 532 nm) was used to produce Fe_2O_3 nanofluid by varying an irradiation time 18hours. Fig. 1 shows the thermal conductivity of base fluid and nano fluid measured by transient hot-wire method.

As shown Fig. 1, the thermal conductivity of the nanofluid increased by 9.3% compared to that of basefluid.

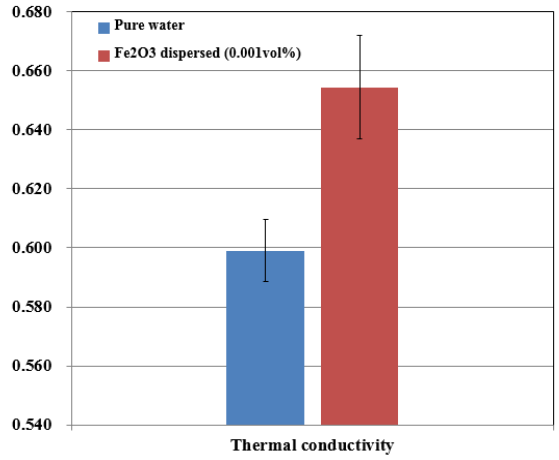

Fig. 1: Thermal conductivity of basefluid and nanofluid

2.3 Heat exchanger design condition

Table 4 summarized heat exchanger design requirements with API 672 std. [15]. Basic supply information is shown in Table 2.

Table 4 Heat exchanger design requirement

Item	Content	Unit
Coolant velocity	1.2 ~ 2.5	m/s
Fouling resistance	0.00018	m^2K/W
Max. pressure drop	1.00	bar

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TEMA class		C	-
Tube side	Material	Cu 90/10 or Cl220 Cu	-
	Out-diameter	Not less than 15	mm
	Wall-thickness	1.25	mm
	U-bend	Not permitted	-

Fig. 2 shows heat exchanger design condition as HTRI input. 2nd intercooler and aftercooler's hardware condition is also same only different compression requirement as shown Table 3.

Shell type is BEM and shell inner diameter is 350mm with 1 pass. Tube type is continuous fin without baffle, tube material is Cu/Ni 90/10, fin material is Al. Tube out-diameter is 15.875mm, wall-thickness is 1.245mm with 30 degree layout angle (21.166mm of pitch) and 4 tube passes. Tube count is 40.

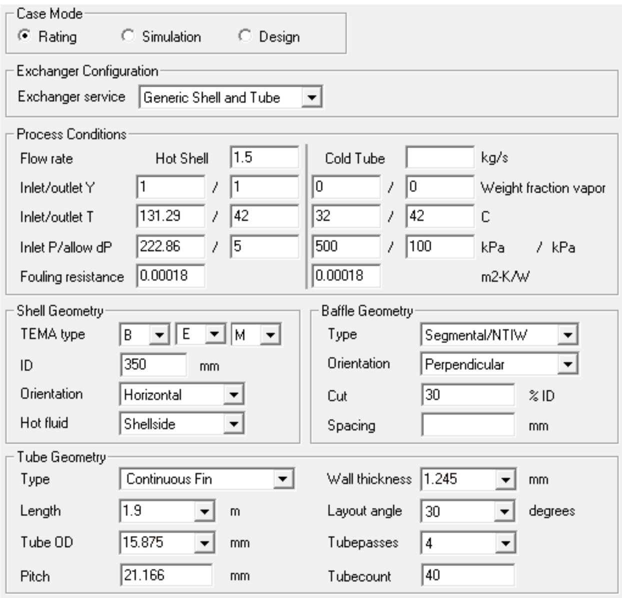

Fig.2: HTRI input condition (1st intercooler)

HTRI has three thermal design modes. There are rating, simulation, and design. Among them most thermal engineer using rating mode cause checking design margin based their input size and design requirement and quick design checking. In this study, we using rating mode.

Fig. 3 shows heat exchanger's sketch layout with in/out nozzle of shell side and tube side. Shell side in/out nozzle location is basically same, however, 2nd intercooler is different cause the aero-compression location (2nd aero-part location).

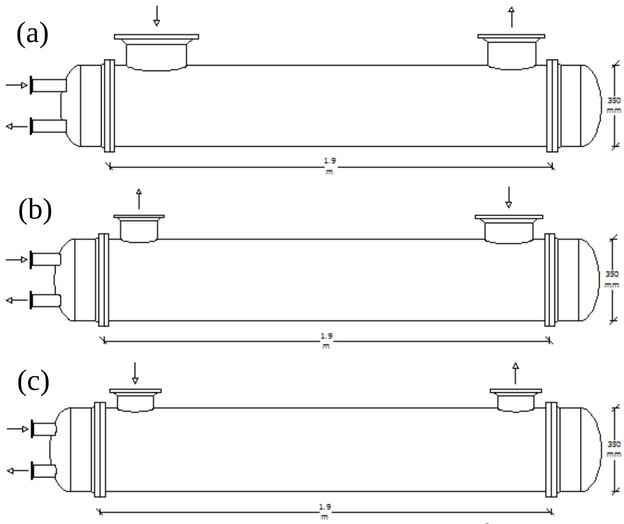

Fig.3: Sketch layout of 1st intercooler (a), 2nd intercooler (b), and aftercooler (c)

III. THERMAL DESIGN RESULT

Table 5 summarized thermal design result using basefluid. The thermal design is satisfied, based on compression requirement and API standard. The coolers are optimized design as shown in Table 5 (overdesign ratio). To take a benefit of cost saving and compressor arrangement, aftercooler using same configuration.

Table 5 Thermal design result using basefluid

Item		1st	2nd	After	Unit
Heat load		135.5	144.6	80.2	kW
Area		17.3	17.3	15.5	m^2
Overdesign		0.54	-0.59	2.47	-
Overall *U*		26	265	250	W/m^2K
Tube	Re	44,306	48,888	29,128	-
	Pr	4.66	4.49	4.66	-
	U	10,639	11,204	7,629	W/m^2K
	ΔP	0.63	0.71	0.28	bar
	v	2.32	2.48	1.52	m/s
Shell	Re	8,194	8,140	8,495	-
	Pr	0.7	0.7	0.7	-
	U	374	376	372	W/m^2K
	ΔP	0.04	0.03	0.07	bar
	v	21.07	9.31	2.14	m/s

To compare with basefluid, used nanofluid thermal conductivity as shown Fig. 1. Table 6 summarized thermal design result using nanofluid.

Table 6 Thermal design result using nanofluid

Item	1st	2nd	After	Unit
Heat load	135.5	144.6	80.2	kW
Area	17.3	17.3	15.5	m^2

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Overdesign		1.48	0.29	3.74	-
Overall U		266	268	254	W/m^{2}K
Tube	Re	44,989	49,935	29,577	-
	Pr	4.13	3.95	4.13	-
	U	11,419	12,026	8,187	W/m^{2}K
	ΔP	0.62	0.71	0.28	bar
	V	2.32	2.47	1.52	m/s
Shell	Re	8,194	8,140	8,495	-
	Pr	0.7	0.7	0.7	-
	U	374	376	372	W/m^{2}K
	ΔP	0.04	0.03	0.07	bar
	V	21.07	9.31	2.14	m/s

IV. DISCUSSION

To understand the effectiveness of nanofluid in heat exchangers, compared heat transfer coefficient of overall, tube, and shell side, calculated Re, and Pr.

Heat transfer coefficient equation is below [16].

$$\frac{1}{U} = \left(\frac{1}{h_t}\right) + \left(\frac{1}{h_s}\right) + \left[\left(\left(tube\ OD \times \ln\left(\frac{tubeOD}{tubeID}\right)\right)/\left(2 \times k_{tube}\right)\right) + \left(\left(\frac{tubeOD}{tubeID}\right) \times (2 \times flowrate)\right.\right.$$

Where, U is overall heat transfer coefficient, h_t and h_s is heat transfer coefficient of tube and shell side, respectively, k_{tube} is thermal conductivity of tube material.

Reynolds number Prandtl number equation is below [16].

$$Re = \frac{(\rho * u * di)}{\mu} \qquad Pr = \frac{(C * \mu)}{k}$$

Where, ρ is density, u is linear velocity, di is tube inner diameter, μ is dyanmics viscosity, C is specific heat capacity, k is thermal conductivity.

Fig. 4 shows tube side heat transfer coefficient comparison. Using nanofluid, the heat transfer coefficient is increased by ~7% compared to that of basefluid. However, as shown is Fig. 5, shell side heat transfer coefficient is not changed cause compressed air property is unchanged.

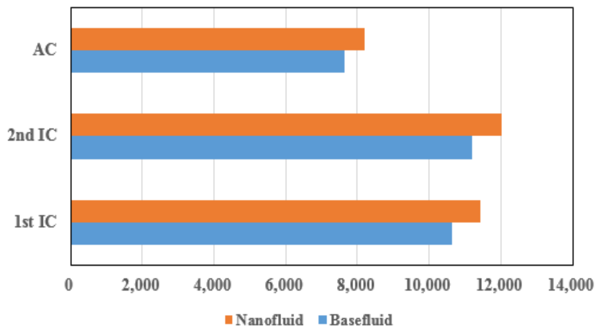

Fig. 4: Tube side heat transfer coefficient comparison

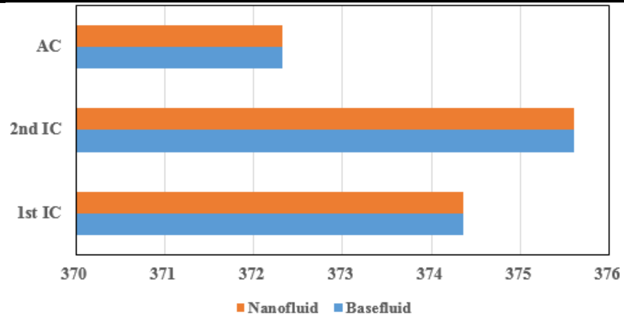

Fig. 5: Shell side heat transfer coefficient comparison

Fig. 6 shows overdesign (%) comparison in rating mode. Using nanofluid, the overdesign (design margin) is increased by 8~30% compared to that of basefluid. From this result, it is possible to obtain higher design margin and energy-efficiency than that of basefluid of a given system and lower pumping power.

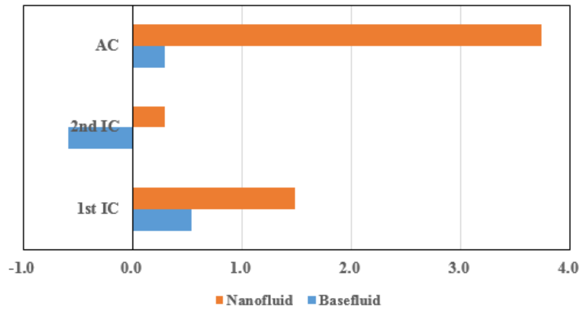

Fig. 6: Overdesign (%) comparison

Fig. 7 and 8 shown tube side Re and Pr comparison. Re is dimensionless number that is used to help similar flow patterns in different fluid flow situations [17]. From analysis result, the tube side flow shows turbulent flow. As shown in Fig. 7, there are no different with two fluid types.

On the other hand, using nanofluid, Pr is decreased by 13% compared to that of basefluid as shown in Fig. 8. Pr is dimensionless number defined as the ratio of momentum diffusivity to thermal diffusivity [18]. Small values of Pr means the thermal diffusivity dominates, the other hands large values of Pr means momentum diffusivity dominates it behavior. Decreased by 13% of nanofluid, it means that the heat diffuses quickly compared to the velocity. This means that for nanofluid thickness of thermal layer is larger than basefluid. It is clearly match to increased thermal conductivity of nanofluid.

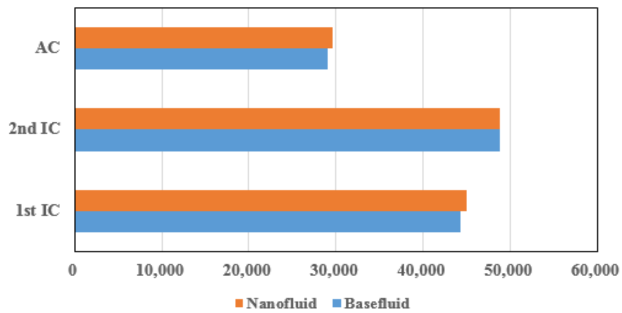

Fig. 7: Tube side Re comparison

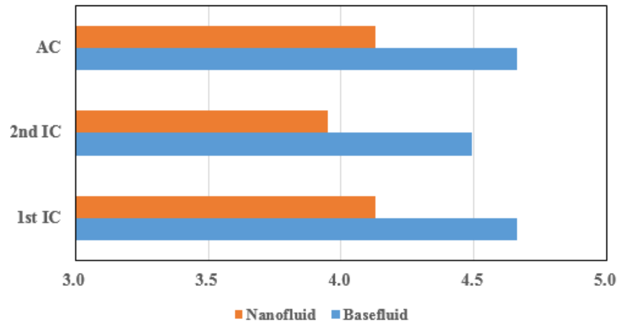

Fig. 8: Tube side Pr comparison

V. CONCLUSION

We have thermal design characterization of heat exchanger for compression cooling system using nanofluid property. The tube side heat transfer coefficient of heat exchanger using nanofluid was increased by ~7% compared to that of basefluid. Tube side Pr number of heat exchanger using nanofluid was decreased by ~ 13% compared to that of basefluid.

We believe that the present design approach is a useful for compression system cooling design. More research will be carried out to find thermal effectiveness of nanofluid in order to understanding the heat transfer mechanism.

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