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Design of Fin-Tube Heat Exchanger for Control of Overheating In Hydraulic Power Pack

Vaishal J. Banker[1]
[1]Mechanical Engineering Department, A D Patel Institute of Technology

***Abstract*: The aim of this work is to control the overheating problem in Hydraulic Power Pack which can cause irregular functioning of any component of the power pack. The effects are like the oil seals get damaged, the wear and tear of the cylinder, hose pipes occurs. Hydraulic power packs are used at many places and in industrial circuits. The calculations are done on the basis of general design equations. Finally, 2D and 3D models are developed using AutoCAD software for the calculated dimensions of fin tube heat exchanger.**
***Keywords*: Hydraulic Power Pack, overheating, fin tube heat exchanger, design, CAD model**

I. INTRODUCTION

Hydraulic power packs are used at many places and in industrial circuits. The main purpose of them is to transmit forces and motion to the assembly attached to the end of the cylinder or end of the circuit. A hydraulic power pack is a unit or a system used to transmit hydraulic power or motion. It consists mainly of a Hydraulic Motor, Hydraulic pump, DCV, Pressure Relief Valve, Oil tank or Reservoir. The hydraulic oil is stored in the Oil reservoir which is basically a tank. The oil is the only medium in the circuit that is flowing continuously on working conditions.

A. There are many problems found in hydraulics like

1) Overheating in hydraulic power packs
2) Troubleshooting in pressure flow and fluid flow in hydraulic power pack
3) Designing and functioning of hydraulic power pack

The overheating problem is a main problem which can cause irregular functioning of any component of the power pack. A hydraulic power pack can malfunction if overheating happens in the hydraulic circuit. Some of the major effects of Overheating can be described in following lines. As a result of low viscosity oil the temperature of the unit increases. So there is a pressure drop found in the unit. So the oil does not flow at the required pressure [1]. The pressure gauge will not indicate the required pressure. The DCV i.e. Directional Control Valve would not function properly as a result of overheating. The efficiency of the unit would decrease. The life of hydraulic motor and hydraulic pump would become less [2]. The oil seals in the hydraulic cylinder would get damaged due to the excess temperature in the power pack. The internal and external leakages in the cylinder would occur more. Another important effect is that the power pack will start to mal-function [3]. That is, the parts will not get required pressure and as a result the system will not work properly. Mal-function means the cylinder will not open at required time, pump will do more work, oil flow in DCV will be affected [4].

In the present work, a fin-tube heat exchanger is designed by use of empirical corelations for controlling of overheating of a hydraulic power pack. Further, a 2D and 3D model is created in AutoCAD using the dimensions found by calculations.

II. EXPERIMENTAL METHOD

A Hydraulic Power Pack having the following specifications is used in the present work. Table 1 and Table 2 list the test conditions for hydraulic oil and air. The hydraulic circuit for the power pack is shown in Figure 1. Figure 2 shows the previous circuit having a new addition of fin tube heat exchanger. The pressure to be exerted is 110 N/mm^2. The flow of heat exchanger is taken as counter-flow and heat exchanger will be air cooled.

TABLE 1
PARAMETERS OF THE OIL FROM HYDRAULIC POWER PACK

Specific Heat	4.2 kJ/kg^{0}C
Inlet Temperature	70^{0}C
Outlet temperature	45^{0}C
Mass flow rate	1.5 kg/s

TABLE II
PARAMETERS OF THE AIR

Specific Heat	1 kJ/kg^{0}C
Inlet Temperature	20^{0}C
Outlet temperature	25^{0}C

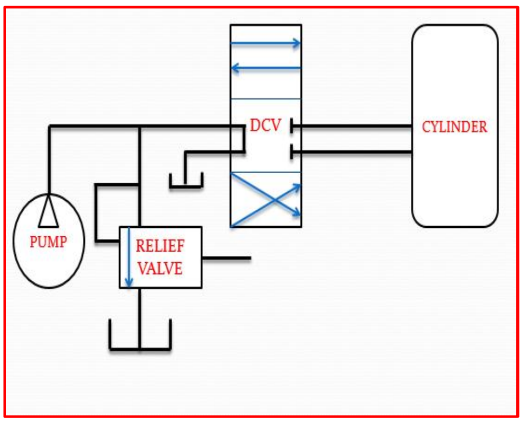

Fig. 1 Hydraulic Circuit for Power Pack

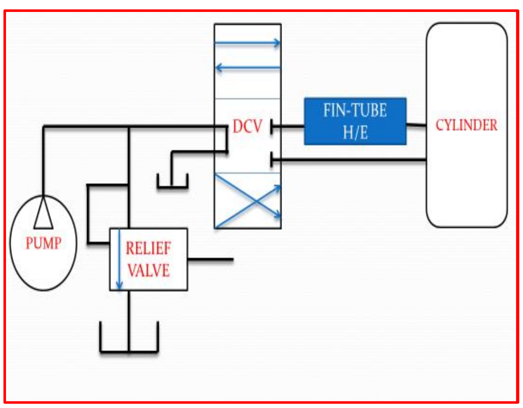

Fig. 2 Proposed Design

III.DESIGN OF FIN-TUBE HEAT EXCHANGER

A. Assumptions

The Overall Heat Transfer Co-efficient (U) is constant. The flow of oil is steady. There is no loss of heat to the surroundings. There is no change of phase of hydraulic oil. The changes in the potential and kinetic energies are negligible. Specific heats and mass flow rates of oil and air is constant.

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B. Thermal Design

1) Heat Transferred (Q)

Heat transferred, Q=Heat generated by oil

By using formula, $Q = (\dot{m}_h) \times C_{ph} \times \Delta T$ (Hot side)

$Cp_h = 4.2$ kJ/kg^{0}C, $\dot{m}_h = 1.5$ kg/s

Q= 4.2 x 1.5 x (70 - 45)

Q=157.5 KW

2) Calculate the mass-flow rate of air($\dot{m}_c$).

$Q = (\dot{m}_h) \times C_{ph} \times \Delta T$ (Hot side)= $(\dot{m}_c) C_{pc} \Delta T$ (Cold side)

$(\dot{m}_h) \times C_{ph} \times \Delta T$ (Hot side)= 157.5 kW

Cpc = 1kJ/kg

157.5=m_c x 1 x (25-20)

Mass flow rate of air , m_c =31.5 kg/s

3) Calculate Logarithmic Mean Temperature Difference

$$\text{LMTD} = \frac{\theta 1 - \theta 2}{\ln(\theta 1 - \theta 2)}$$

where, $\theta_1 = (t_{h1} - t_{c2})$

$\theta_2 = (t_{h2} - t_{c1})$

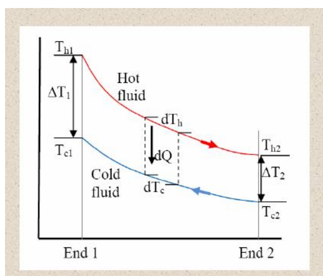

Fig. 3. CounterFlow[2]

- $t_{h1} = 70^0$C $\quad$ $t_{h2} = 45^0$C
- $t_{c1} = 20^0$C $\quad$ $t_{c2} = 25^0$C
- Flow is considered to be counter-flow as it is suitable for cooling of liquids.

LMTD= 34.02℃

4) Calculate heat flow area (A)

From the data tables the overall heat transfer coefficient U i.e.,

$$U_{DO} = 50 \text{ W/m}^2 ℃$$

The value of overall heat transfer coefficient is for liquids and fluids used as industrial oils.

Now, Q= U_{DO} x A x LMTD

157.5=50 x A x 34.02

Net flow heat area, A=0.092 m^2

5) Decide O.D, I.D, wall thickness of pipe. State the grade of the pipe to be used.

Let size of tube be 1” BWG 8 GRADE

Outer Diameter O.D	: 1”	= 25.4 mm
Inner Diameter I.D	: 0.670”	= 17 mm
Wall thickness	: 0.166”	= 4 mm
Flow Area	: 0.355”	= 230 mm^2

6) *Calculate heat transfer coefficient of pipe. Select a suitable fouling factor for hydraulic oils.*

Fouling Factor for Oil= Rs= 0.0001

Heat Transfer co-efficient for pipe $= \frac{1}{0.0001}$ = 10000 W/m^2°C

7) *Design fins. State the shape of fins, width of fins, no. of fins, length of tube, ratio of fin height to spacing.* Assume Circular fins of 2” diameter. Circular fins are selected because the pressure distribution and heat dissipation capacity of the circular surfaces is high as compared to rectangular fins.

a) O.D : 2” dia
b) Width : 3 mm
c) No. of fins : 3 fins/inch
d) Ratio of fin height to spacing $(\frac{F_h}{S_f})$: 2.6
e) Length of tube : 12”
f) No of fins : 36

8) *NTU Method*

C_c = ṁcCpc= 31.5 kW

C_h=ṁh C_{ph} = 6.3 kW

Now, $C_c > C_h$

Effectiveness , $\varepsilon = \frac{Q}{Q_{max}} = \frac{t_{h1} - t_{h2}}{t_{h1} - t_{c1}}$ = 0.9

Area Required:

$C_{min} = C_c$ =6.3 kW
$C_{max} = C_h$= 31.5 kW
R=$\frac{C_{min}}{C_{max}}$ = 0.2

Effectiveness , $\varepsilon = \frac{1-\exp(-NTU(1-R))}{1-R\exp(-NTU(1-R))}$

From the above equation we can calculate NTU. It is found that NTU=2.6306

NTU= $\frac{UA}{C_{min}}$

A= 0.33 m^2

9) *Heat transfer with and without fins*

Heat Transfer without fins = Q = 157.5 kW
Heat transfer with Fins:

a) Q = k x A_{cs} x m x $(t_o - t_a)$

b) m = $\sqrt{\frac{hP}{kA_{cs}}} = \sqrt{\frac{50(.0159)}{207.6\,(2.02*10^{-3})}}$ = 18.95

$$P = \pi \text{ D} = 50.8 * 3.14 = .0159 \text{ m}$$
$$A_{cs} = \frac{\pi}{4} D^2 = 2.02 * 10^{-3} m^2$$

Now, Q = 207.6 x $2.02\, x 10^{-3}$ x 18.95 x (70 - 25)

$$Q = 357.6 \text{ kW}$$

This value is greater than the rate of actual heat transferred without fins. So our design of fins is safe.

10) Calculation of Re, Nu, Pr.

Viscosity	$v = 8 \times 10^{-6} m^2/s$
Specific Heat	C_p = 4200 J / kg
Thermal Conductivity	k = 0.3950 W/m °C
Density	$\rho = 1050\ kg/m^3$
Velocity	U= 2.5 m/s

a) Reynolds' Number

$$\text{Re} = \frac{U\,D}{v}$$
$$= \frac{2.5\,(.0254)}{8 \times 10^{-6}}$$
$$\text{Re} = 793$$

b) Prandtl number

$$\text{Pr} = \frac{\rho v \text{Cp}}{\text{k}}$$
$$= \frac{1050\,(8 \times 10^{-6})4200}{.3950}$$
$$\text{Pr} = 89.31$$

c) Nusselt Number

$$\text{Nu} = 0.023\,(Re)^{0.8}(\text{Pr})^{\frac{1}{3}}$$
$$= 0.023\,(7937)^{0.8}(89.37)^{\frac{1}{3}}$$
$$\text{Nu} = 135$$

d) So the flow is turbulent flow from the values of Re, Nu, Pr.

C. CAD Model

The 2D and 3D models as shown in Figure 4 and Figure 5 are developed using AutoCAD 2009 software for the above calculated dimensions of fin tube heat exchanger [4].

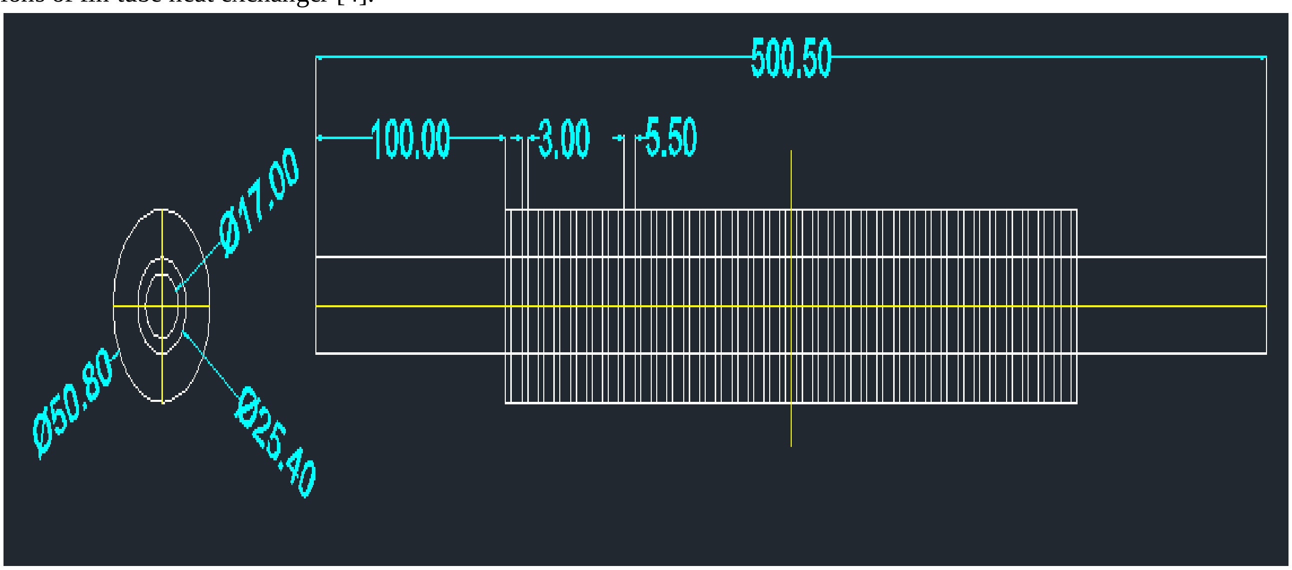

Fig. 4. 2D Model

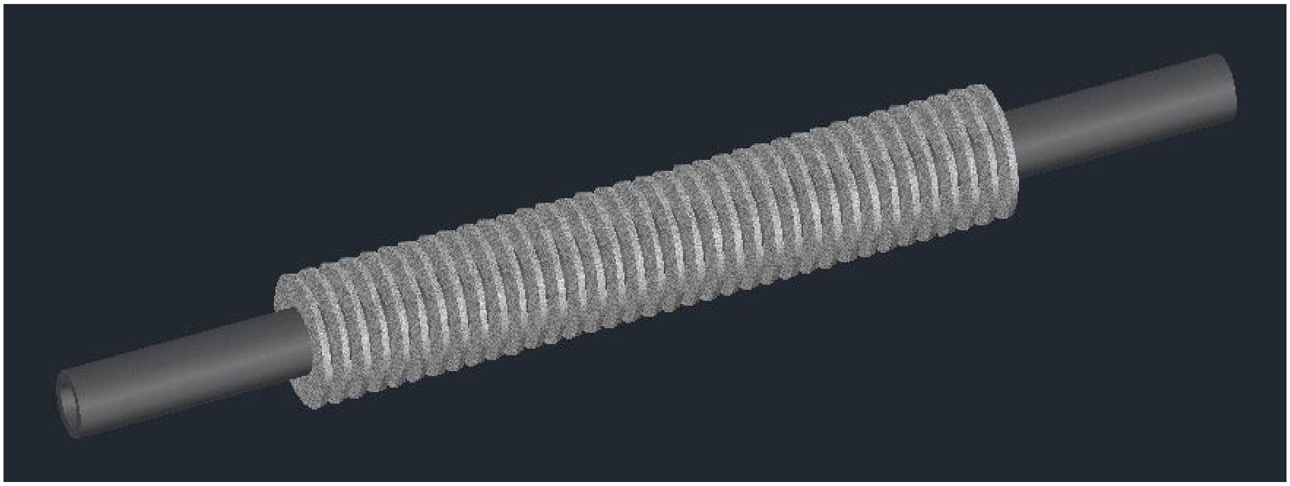

Fig. 5. 3D Model

IV. CONCLUSIONS

A detailed design was created by using primary equations for afin-tube heat exchanger. Heat transfer with fins was calculated as 357.6 kW which was greater than the rate of actual heat transferred without fins. So proposed design of fin tube heat exchanger is safe. With this the oil running period would increase and the components would function properly.

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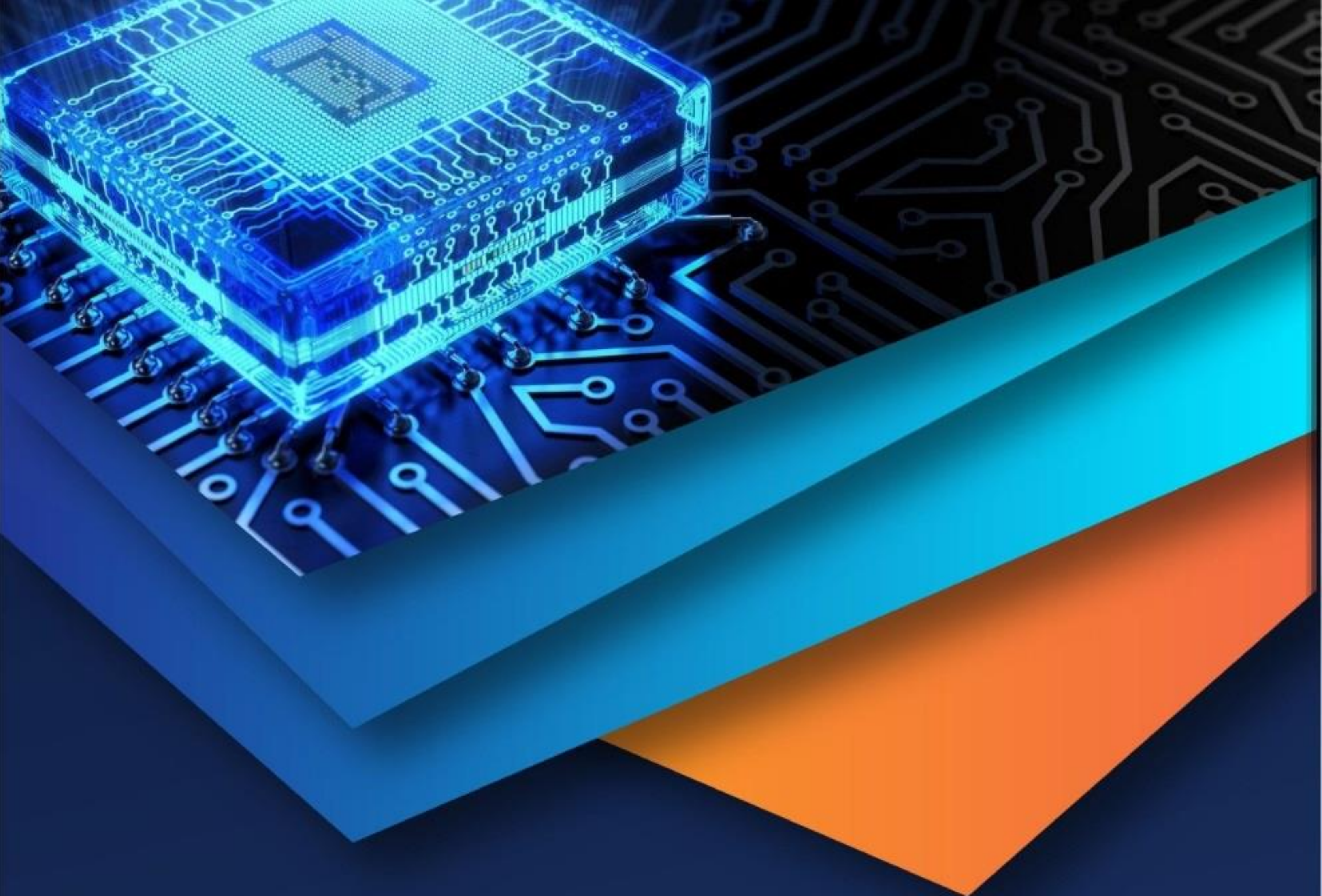

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