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ANALYSIS FOR NATURAL CIRCULATION LOOP

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ABSTRACT

A natural circulation system operates on the basis of natural laws like gravity and buoyancy. Although natural circulation is a benign gift of nature for applications to several heat removal systems due to their simplicity in design, elimination of hazards related to pumps, better flow distribution, cost reduction, etc. however, the potential threat of flow instabilities still eludes for its wide applications. Although addition of local losses (orificing) may suppress instabilities, however, it is accompanied by significant flow reduction which is detrimental to the natural circulation heat removal capability. A rectangular single-phase natural circulation loop, which consists of two horizontal copper tubes (heat transfer sections) and two vertical copper tubes (legs), connected by means of four bends. The lower heating section consists of an electrical heating wire made of nichromel on the outside of the copper tube, the upper cooling system consists of a coaxial cylindrical heat exchanger with a water set at controlled temperature and flowing through the annulus. The parameters investigated during the experiments power transferred to the fluid and effectiveness of the heat transfer and determine the velocity of the copper pipe and power raised the loop.

Key words: Natural circulation loop, heat exchanger, effectiveness.

I.INTRODUCTION

In this chapter the topic of this investigation is introduced. Some background is provided while the purpose, scope and deliverables of the investigation are discussed. Natural circulation is a simple phenomenon which occurs in a fluid in presence of temperature and density gradients in a force field. In natural circulation systems there are a heat source and a heat sink, with the former placed lower than the latter, both in contact with a portion of fluid. As consequence of the heat fluxes, the heated part of the fluid becomes lighter and rises, while the cooled part becomes denser and is dropped down by gravity. These combined effects establish circulation. As it does not need any moving mechanical part, like pump or fan, natural circulation is characterized by high reliability and low costs of maintenance. On the other hand, it is very important the design of the systems which use natural circulation as primary heat transfer mechanism in order to optimize the thermal performances and to avoid unwanted dynamic behaviours, such as flow instabilities or flow reversals.

Various models were proposed in literature. One of the first models was introduced by in figure and consisted in a loop made by point heat source and point heat sink, both with imposed wall temperature, connected by two adiabatic legs. We lander explained the presence of instabilities and flow reversal with the “hot pocket” theory: when buoyancy forces and shear stresses are not in phase, in case of specific boundary conditions, the presence of a temperature discontinuity (hot pocket) may cause an initial velocity oscillation, which grows in time and amplitude, eventually producing a flow reversal.

In rectangular loops also the influence of the aspect ratio, defined as the height to width ratio, was systematically analysed. By analytically found that the stability of the system has a minimum for aspect ratio approaching the unity (square loop). Therefore our loops were designed with aspect ratios between 0.84 and 1.47, very close to the critical one. Also the influence of different materials of the pipes was analysed. In particular it was found that the more conductive are the pipes the more stable is the system, because the temperature discontinuities are smoothed by the heat transfer between fluid and tube walls as shown in figure 1.

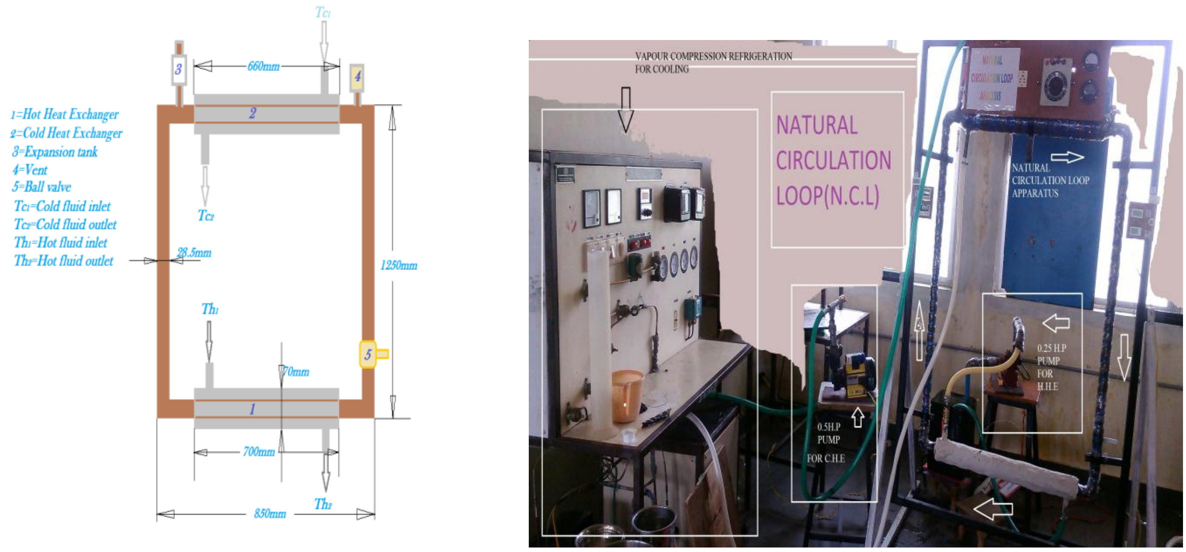

Fig. 1 Experimental setup

1. LITERATURE SURVEY:

A natural circulation system operates on the basis of natural laws like gravity and buoyancy. Although natural circulation is a benign gift of nature for applications to several heat removal systems due to their simplicity in design, elimination of hazards related to pumps, better flow distribution, cost reduction etc. however, the potential threat of flow instabilities still eludes for its wide applications. Although Addition of local losses may suppress instabilities. The purpose of this research paper is concerned with the use of Al2O3nanofluids to suppress the instabilities in a single-phase natural circulation loop induced by a heating–cooling system. Experiments were demonstrated by that with Nano fluids at different concentrations[1].

Single-phase natural circulation inside rectangular loops is studied. The large scale loop presented different dynamical behaviours (stable and unstable flow with or without mass flow rate oscillations), while both mini-loops showed always steady-state conditions at regime. Flow velocities were evaluated by means of frequency analysis or with an enthalpy balance in case of steady state flow and analysed as function of the boundary conditions. Natural circulation is a simple phenomenon which occurs in a fluid in presence of temperature and density gradients in a force field. In natural circulation systems there are a heat source and a heat sink, with the former placed lower than the latter both in contact with a portion of fluid. As consequence of the heat fluxes, the heated part of the fluid becomes lighter and rises, while the cooled part becomes denser and is dropped down by gravity [2].

A Natural circulation, closed loop thermo syphon can transfer heat over relatively large distances without any moving parts such as pumps and active controls. Such loops are thus considered suitable for nuclear Reactor cooling applications where safety and high reliability are Paramount importance [3].

The study presents a numerical investigation of the dynamical behaviour of a rectangular natural circulation loop with horizontal heat exchanging sections. The study has been developed in a two dimensional domain considering uniform wall temperatures and thermally insulated vertical legs as thermal boundary conditions. The analysis has been performed for a fixed geometry of the loop and for various Rayleigh numbers, separating the values of Rayleigh for which the system manifests stable and unstable dynamics [4].

2. PROPOSED WORK:

2.1 Experimental Setup:-

Experiments were conducted in a natural circulation loop with geometry.The test facility resembles rectangular in geometry with circular flow cross-section area. The pipes are made of Copper with inner diameter of around 26.6 mm. Important dimensions of the loop are shown in Figure. The loop consists of a hot heat exchanger (tube in tube type) at the bottom of the horizontal leg through which hot water continuously flows with the help of a 0.25hp pump. It was cooled at the top through a tube-in-tube type heat exchanger with tap water flowing through the annulus by the help of a 0.5hp pump. An expansion tank was provided at the topmost elevation to accommodate the volumetric expansion of the fluid. It also ensures that the loop remains full of water. Thermocouples were installed at different positions in the loop to measure the instantaneous local temperature. The flow rate was measured using a differential pressure transducer installed in the horizontal leg of the loop. The instruments were connected to a data acquisition system which could scan all the channels in less than 1s. The secondary side cooling water flow rate was measured with the help of a rotameter. The loop was insulated to minimize the heat losses to the ambient.

2.2 Experimental Procedure

This experiment is done with the help of a circulation of water to vapour with bottom to top. The experiment is analysis with a practical and theoretical value depends on the different parameter and showing different inclination position.

2.3 Basic Procedure

The primary loop was filled with tap water. To drive out the air bubbles, the filling flow was continued for some more time with pulsing. To remove the dissolved gases in the water, the loop will run under natural circulation condition at a small power for some time. This procedure will be followed in all the tests to drive out the dissolved gases. Before the experiment, the secondary cooling water flow rate was set at the required value. Sufficient time was given for the test loop to stabilize at room temperature without heating. Then power was switched on and actual recording of data began.

2.3.1 Steady-State Test Procedure

When steady initial conditions were reached, the reading was noted with zero power. Then the heater was put on and set at the required power using the DC controlled heating Variac. At the steady-state condition (which can be observed from the trend of the pressure drop data and temperature variation in the graphical display mode on the data acquisition system), all the temperatures and pressure drop data were collected. Then the power was increased in steps of 50W and allowed to attain the steady state. The same procedure was repeated till the power reached 450W. Repeatability of the test results was also checked.

2.3.2 Procedure for The Stability Tests

The instability threshold was found to be dependent on heat addition paths. The scenarios for the following heat addition paths were studied:

1. start-up from stagnant conditions;
2. sudden power raising from an initial stable steady-state conditions;
3. Instability behaviour during a power step back process.

During all the tests the cooling water flow rate (1l/min) and its inlet temperature (around 30°C) were kept constant. The fluctuation in the controlled coolant flow rate was within ±3% of the set value. A brief experimental procedure is given below for each of the tests.

2.3.3 Start-Up From Stagnant Conditions

The experimental procedure consisted of suddenly switching on the heater when the fluid was stagnant at uniform initial temperature throughout the loop. To achieve uniform initial temperature, the cooling water was valved-in at least half an hour before the actual test. The system was allowed to operate until the flow was stable. The primary objective of these tests was to obtain the threshold power below which the system can be started up without encountering instability with sudden increase in power from initial stagnant condition. These experiments were conducted by water

Sl No	Time Interval In Minutes	Hot Water inlet	Hot Water Outlet	T1	T2	T3	T4	T5	T6	Cold Water Inlet	Cold Water Outlet	Cold Water Discharge m^3/S	Hot Water Discharge m^3/S
		Temperature in ℃											
1	45	90.2	88.6	42	42	38	38	38	51	6	13	0.00023	0.00019
2	90	91.1	89	45	44	42	40	39	51	6.3	14.2	0.00022	0.00019
3	135	91.8	89.5	49	46	47	43	42	48	6.5	14.8	0.00022	0.00019
4	180	92.5	87.3	52	53	53	46	45	47	6.8	15.6	0.0002	0.0001
5	225	93	88	54	55	56	48	47	46	7	16	0.0002	0.0001

The values of cold and hot water discharge as shown in table.1 w.r.to the time intervals and change in temperature are obtained by conducted the experiment.

3. PROBLEM ANALYSIS AND CALCULATIONS:

The experimental approach of NCLs' performances and behaviour is commonly preferred to CFD analysis: it is not easy to perform an extensive and reliable numerical analysis without the help of experimental data, due to the problem chaotic features. The latter approach gives more information about a lot of parameters which are more difficult to measure such as flow rate, local fluid velocity, pressure drops and instantaneous heat flow rate. The CFD is helpful to evaluate the three-dimensional effects, which cannot be deduced from the measurement of the temperatures. To check the validity of the numerical code used for solving the full elliptical governing equations, the numerical scheme was tested on cases close to the problem under consideration, in order to compare the calculated data with the published data. The comparisons were made for natural convection flow of air on two cases: a square cavity with differentially heated vertical walls and adiabatic horizontal wall.

With the demand for energy continues to grow globally, there is a need to make heat transfer equipment more energy efficient. At one hand, the exponential growth of electronics, communication and computer technology and their choice to go for miniaturization has put added pressure on the designer to create efficient thermal management devices for these systems.

3.1. Effectiveness Calculation For Hot Heat Exchanger

Experimental discharge of .25hp pump, $Q_{discharge}=0.0001 \frac{m^3}{sec}$

Experimentally calculated values:

T_{h1} in ^oC	T_{h2} in ^oC	T_{c1} in ^oC	T_{c2} in ^oC
93	**88**	**46**	**54**

We know that:

$\Theta_1=t_{h1}-t_{c2}=93-54=39^OC$ $\Theta_2=t_{h2}-t_{c1}=88-46=42\,^OC$

Therefore: $\Theta_m=\frac{(\theta1-\theta2)}{\ln(\theta1/\theta2)}=40.4814747^oC$

Mass flow rate of hot water,

$\dot{m}_h$=Density x discharge=978.1 x 0.0001=0.09781$\frac{kg}{sec}$

Specific heat at constant presser :

At 90^oC, C_{ph}= $4.205\frac{KJ}{Kg\,K}$ At 50^oC, C_{pc}= $4.179\frac{KJ}{Kg\,K}$

Heat transfer rateof hot heat exchanger,

$Q=\dot{m}_h x\ C_{ph} x\ (T_{h1}-T_{h2})=2.05645525$ watt

Mass flow rate of cold water,

$\dot{m}_C=\frac{(mh\ x\ Cph)x\ (Th1-Th2)}{Cpc(Tc2-Tc1)}=0.06151158\frac{kg}{sec}$

Heat capacity:

Heat Capacity of hot fluid, $C_h=m_h x C_{ph}=0.41129105\frac{KJ}{K}$

Heat Capacity of cold fluid, $C_c=m_C x\ C_{ph=}0.257056906\frac{KJ}{K}$

$C_{min}=0.257056906\frac{KJ}{K}$ $C_{max}=0.41129105\frac{KJ}{K}$

$Q_{max}= C_{min}\ x\ (T_{h1}-T_{c1})=0.257056906\ x\ (47)=12.08167458$ watt

Effectiveness:

$\mathcal{E}_h=\frac{Q}{Qmax}=\frac{2.05645525}{12.08167458}=0.17021$ or (17%)

3.2. Effectiveness calculation for cold heat exchanger:

Experimental discharge of .5hp pump, $Q_c=0.0002\ \frac{m^3}{sec}$

Experimentally calculated values :

T_{h1} in ^oC	T_{h2} in ^oC	T_{c1} in ^oC	T_{c2} in ^oC
56	**48**	**7**	**16**

$\Theta_1=t_{h1}-t_{c2}=56-16=40^OC$ $\Theta_2=t_{h2}-t_{c1}=48-7=41\ ^OC$

Therefore:

$\Theta_m=\frac{(\theta1-\theta2)}{\ln(\theta1/\theta2)}=\frac{(40-41)}{\ln(40/41)}=40.4979423^oC$

Mass flow rate of cold water,

$\dot{m}_c$=Density x discharge=978.1 x 0.0002=0.19562 $\frac{kg}{sec}$

Specific heat at constant presser

At 90^oC, C_{pc}= $4.179\frac{KJ}{Kg\,K}$ At 50^oC, C_{ph}= $4.193\frac{KJ}{Kg\,K}$

Heat transfer rate of cold heat transfer,

$Q=\dot{m}_h x\ C_{ph} x\ (T_{h1}-T_{h2})=7.357464$ watt

Mass flow rate of hot water,

$\dot{m}_h = \frac{(mh \times Cph) \times (Th1-Th2)}{Cpc(Tc2-Tc1)} = 0.1733039 \frac{kg}{sec}$

Heat capacity:

Heat Capacity of hot fluid, $C_h = \dot{m}_h x\ C_{ph} = 0.726663093 \frac{KJ}{K}$

Heat Capacity of cold fluid, $C_c = \dot{m}_C x\ C_{ph=} 0.81749598 \frac{KJ}{K}$

$C_{min} = 0.726663093 \frac{KJ}{K}$ $C_{max} = 0.81749598 \frac{KJ}{K}$

$Q_{max} = C_{min}$ x $(T_{h1}-T_{c1}) = 0.726663093$ x$(56-7) = 35.60649$ watt

Effectiveness:

$\mathcal{E}_h = \frac{Q}{Qmax} = \frac{7.357464}{35.60649} = 0.20663266$

Calculation Driving Force (Change In Mass Flow Rate)

3.3. Mass flow rate in Hot Heat Exchanger section:

Experimental discharge of .25hp pump, $Q_{discharge} = 0.0001 \frac{m^3}{sec}$

Experimentally calculated values:

T_{h1} in ^oC	T_{h2} in ^oC	T_{c1} in ^oC	T_{c2} in ^oC
93	**88**	**46**	**54**

$\Theta_1 = t_{h1}-t_{c2} = 93-54 = 39^OC$ $\Theta_2 = t_{h2}-t_{c1} = 88-46 = 42\ ^OC$

Therefore:

$\Theta_m = \frac{(\theta1-\theta2)}{\ln(\theta1/\theta2)} = 40.4814747^oC$

Mass flow rate of hot water,

$\dot{m}_h$=Density x discharge=978.1 x 0.0001=0.09781 $\frac{kg}{sec}$

Specific heat at constant presser :

At 90°C, $C_{ph} = 4.205 \frac{KJ}{Kg\ K}$ At 50°C, $C_{pc} = 4.179 \frac{KJ}{Kg\ K}$

Heat transfer rate of hot heat exchanger,

$Q = \dot{m}_h x\ C_{ph} x\ (T_{h1}-T_{h2}) = 2.05645525$ watt

Mass flow rate of cold water,

$\dot{m}_{Ch} = \frac{(mh \times Cph)x\ (Th1-Th2)}{Cpc(Tc2-Tc1)} = 0.06151158 \frac{kg}{sec}$

3.4. Mass flow rate in Cold Heat Exchanger section:

Experimental discharge of 0.5hp pump, $Q_c=0.0002\ \frac{m^3}{sec}$

Experimentally calculated values :

T_{h1} in °C	T_{h2} in °C	T_{c1} in °C	T_{c2} in °C
56	48	7	16

$\Theta_1=t_{h1}-t_{c2}=56-16=40^OC$ $\Theta_2=t_{h2}-t_{c1}=48-7=41\ ^OC$

Therefore:

$\Theta_m=\frac{(\theta1-\theta2)}{\ln(\theta1/\theta2)}=\frac{(40-41)}{\ln(40/41)}=40.4979423^oC$

Mass flow rate of cold water,

$\dot{m}_c$=Density x discharge=978.1 x 0.0002=0.19562 $\frac{kg}{sec}$

Specific heat at constant presser :

At 90°C, C_{pc}= 4.179 $\frac{KJ}{Kg\ K}$ At 50°C, C_{ph}= 4.193 $\frac{KJ}{Kg\ K}$

Heat transfer rate of cold heat transfer, Q= $\dot{m}_h$x C_{ph}x $(T_{h1}-T_{h2})$=7.357464watt

Mass flow rate of hot water,

$\dot{m}_{hc}=\frac{(mh\ x\ Cph)\ x\ (Th1-Th2)}{Cpc(Tc2-Tc1)}=0.1733039\frac{kg}{sec}$

Mass flow rate of inside the loop fluid in hot heat exchanger is 0.06151158 $\frac{kg}{sec}$and mass flow rate of inside the loopfluid in cold heat exchanger is 0.1733039$\frac{kg}{sec}$.The difference of mass flow rate of this two heat exchanger is 0.11179232$\frac{kg}{sec}$

As there is change in mass flow rate in the bottom and upper heat exchanger so it is concluded that there is fluid flow inside the loop. The variation between mass flow rate and heat transfer rate for hot and cold water can be observed as shown in figure 2,3 and 4.

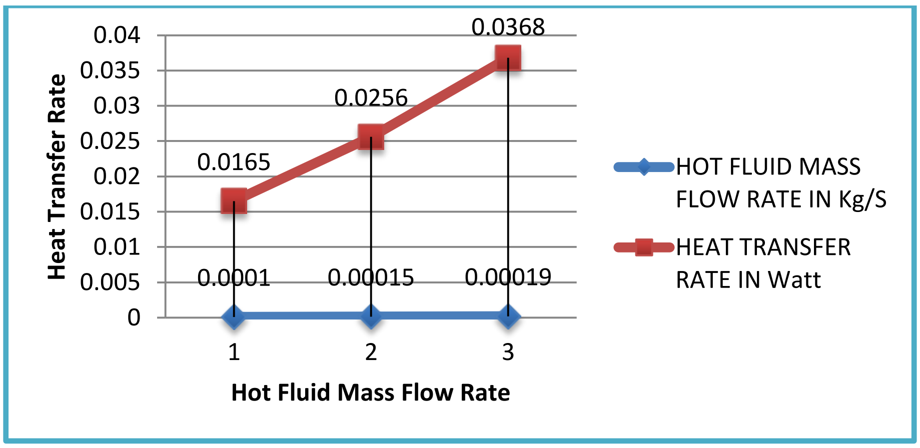

Fig.2 Hot fluid mass flow rate vs heat transfer rate

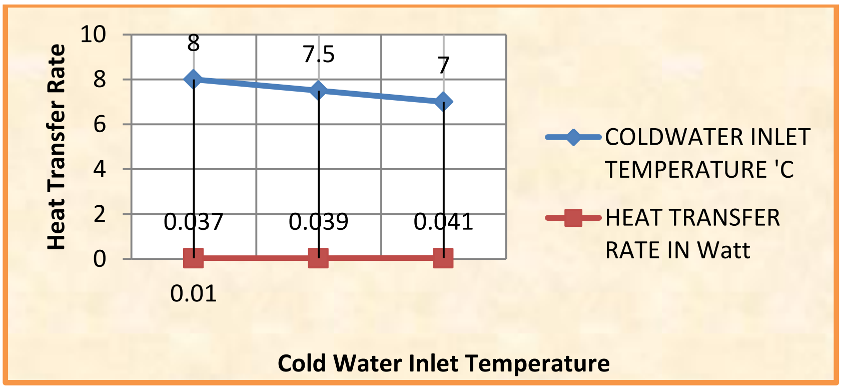

Fig.3 Cold inlet temperature vs heat transfer rate

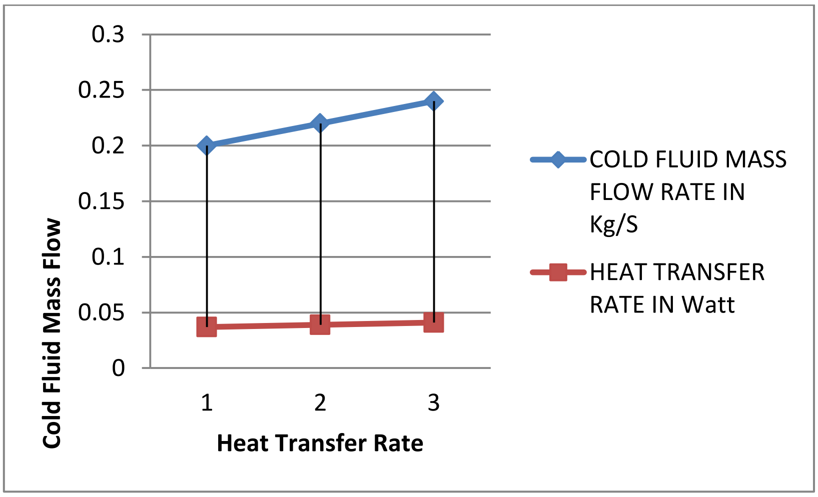

Fig.4 Cold fluid mass flow vs heat transfer rate

4. CONCLUSION:

The Natural Circulation Loop was successfully studied and fabricated. Experiments are conducted with the distilled water and after some time the temperature difference with in the loop has been observed by the temperature indicator. We conclude that the fluid flow has been taking place within the loop by density difference. And we plot the graphs of heat Transfer rate to hot flow inlet temperature, heat Transfer rate to Mass flow rate of hot fluid and heat Transfer rate to cold water inlet temperature. We have observed that Heat Inlet Temperature and Mass Flow rate of hot fluids directly proportional to Heat Transfer rate and cold inlet temperature and Mass Flow rate of cold fluid is directly indirectly proportional to Heat Transfer rate.

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