# PERFORMANCE ANALYSIS OF TUBE IN TUBE HEAT EXCHANGER USED FOR LIQUID DESICCANT BASED AIR CONDITIONING CYCLE

**Nishadevi N. Jadeja[1] & Jay Solanki[2]**
[1]Asst. professor, Mechanical department, Government engineering college, Bhavnagar.

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**ABSTRACT** *There is dire need to develop alternative technologies for air conditioning, which can use renewable or waste energy sources rather than electricity. Liquid desiccant dehumidification technology is a promising alternative to conventional air conditioning systems, particularly for reducing power consumption for fresh air dehumidification.LD can be used to remove latent heat from air in a component called dehumidifier. In present work performance of tube in tube heat exchanger was investigated in a liquid desiccant air conditioning system.For tube in tube heat exchanger, the mass flow rate on hot side was kept constant (22-24 kg/h) and on cold side was varied, which was 25, 50 and 75 kg/h. Maximum heat transfer rate achieved was 0.256 kW at flow rate 76 kg/h and maximum rise in temperature was 8.1oC at flow rate 22 kg/h.*

***Keywords****: tube and tube heat exchanger, liquid desiccant based air conditioning cycle, heat transfer rate.*

### 1.1 Energy Scenario and Air Conditioning

At Present, Cooling represents the third leading use of energy and source of carbon emissions in commercial buildings (after lighting and space heating) and the fourth leading use of energy and source of carbon emissions in the residential buildings (after space heating, water heating and lighting).So, one can not underestimate energy consumed by air conditioning. The most common system used for air conditioning is VCR system, runs on electrical energy. It consumes a big part of total electrical energy used. So, increasing demand of air conditioning will also increase energy scarcity. A fundamental problem with vapor compression air conditioners is that they do a great job at sensible cooling, but provide a very inefficient system for latent cooling. A typical airconditioner provides 80% of its total cooling as sensible cooling (decrease in temperature) and only 20% as latent cooling (removing moisture). Also it is very poor in removing moisture. If desired humidity ratio is very low, to remove moisture by conventional HVAC system air needs to be overcooled upto dew point temperature and then it should be reheated after removing desired moisture. So, there is wastage of energy and also it uses high grade electrical energy. One thing is sure that we must consider change in design of buildings so that comfortable conditions can be maintained with less active cooling and dehumidification and to develop air conditioners that run on renewable energy sources, to continue benefits of air conditioning for long period.

### 1.2 Objectives and Methodology

The objectives of present work is to investigate performance of plate type and tube in tube type heat exchangers for 1 TR LDAC system. This would involve following steps:

A. Prepare heat exchanger test set up with instrumentation, which can provide conditions similar to actual 1 TR LDAC system.

B. Investigate performance and determine efficacy of plate heat exchanger and tube in tube heat exchanger for heat exchange between two streams of LD in LDAC system.

C. Analyse the results, draw conclusions and make recommendation for future.

## Experiments on Tube in Tube Heat Exchanger

### 2.1 Test set up, Instrumentation and Experiment Procedure.

Here, the same test set up used for plate heat exchanger was used. All the arrangement for test set up, instrumentation and experiment procedure is same as it is in plate heat exchanger. Tube in Tube Heat Exchanger.

The tube in tube heat exchange is shown in fig.2.1. It contains one ethylene plastic tube (inside) and a stainless steel tube (outside) with dimensions shown below. Of course, the Page | 47 inside tube material ethylene plastic is not a good conductor of heat. But, the same was used because of costing limitation.

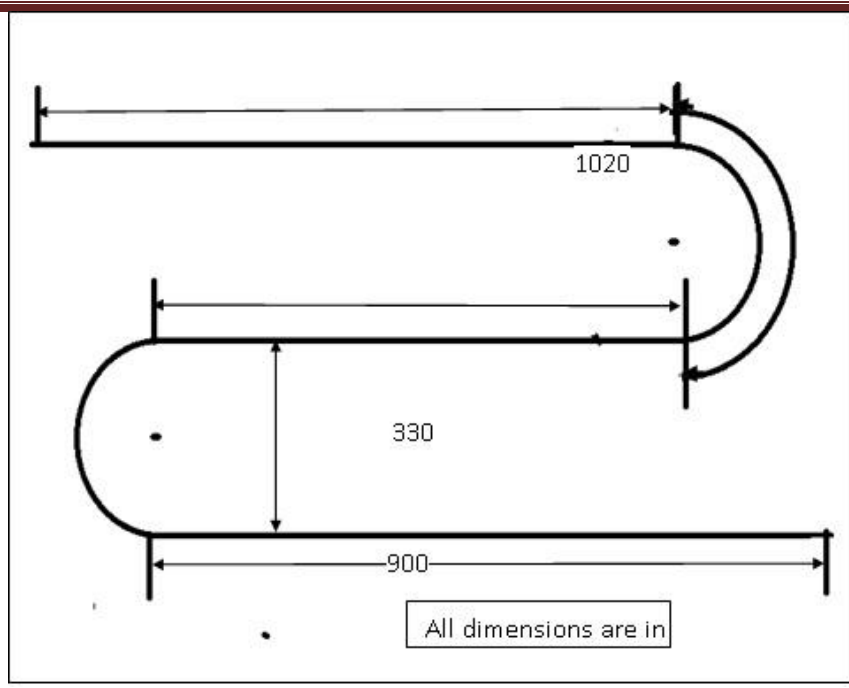


Figure 2.1: Tube in Tube Heat Exchanger Dimensions

The configurations of tube in tube heat exchanger is as follow

| Inside Tube | | Outside Tube | |
|---|---|---|---|
| **Material** | Ethylene Plastic | **Material** | Stainless Steel |
| **Outside Diameter** | 6.35 mm | **OutsideDiameter** | 28.5 mm |
| **Inside Diameter** | 4.38 mm | **Inside Diameter** | 25.4 mm |
| **Thickness** | 1 mm | **Thickness** | 1.5 mm |
| **Thermal Conductivity** | 0.2 W/mK | **Thermal Conductivity** | 16 W/mK |

Table 2-1Configurations of tube in tube heat exchanger

- Total length – 1020+330+900+330+1000 = 3580 mm

The experiments were carried out. The parameters such as temperatures, mass flow rate and specific gravity are directly measured using the measuring instruments. The heat lost rate by hot LD (Qh), heat gain rate by cold LD (QC), rate of heat lost to atmosphere from heat exchanger surface (QLOST), maximum possible heat transfer rate (QMAX) effectiveness of heat exchanger (ε) and LMTD are calculated from these values (same as in plate heat exchanger).

Experiments were carried out with constant mass flow rate (22-24 kg/h) on hot side (which was in inside plastic tube) and on cold side (which was in outside SS tube) it was varied, values - 25, 50 and 75 kg/h and same values for inlet temperatures of hot (70oC) and cold LD (30oC)

The results for experiments with PHE at different mass flow rates are summarized in table 2.1

| | Nomenclature | Unit | 1 | 2 | 3 |
|---|---|---|---|---|---|
| Mass flow rate of LD on cold side | $M_C$ | kg/h | 22.30 | 49.08 | 76.08 |
| Mass flow rate of LD on hot side | $M_H$ | kg/h | 24.70 | 22.56 | 22.36 |
| Temperature of hot LD at inlet | $T_{HI}$ | $^oC$ | 75.9 | 75.9 | 73.3 |

| | | | | | |
|---|---|---|---|---|---|
| Temperature of hot LD at outlet | $T_{HO}$ | $^oC$ | 62.8 | 60.6 | 57.7 |
| Temperature of cold LD at inlet | $T_{CI}$ | $^oC$ | 35.5 | 33.6 | 31.8 |
| Temperature of cold LD at outlet | $T_{CO}$ | $^oC$ | 43.5 | 38.7 | 34.9 |
| **Heat lost rate by hot LD** | **$Q_H$** | **kW** | **0.234** | **0.248** | **0.256** |
| Heat gain rate by cold LD | $Q_C$ | kW | 0.133 | 0.187 | 0.169 |
| Heat lost rate from HE surface | $Q_{LOST}$ | kW | 0.101 | 0.058 | 0.086 |
| Maximum possible heat transfer rate | $Q_{MAX}$ | kW | 0.576 | 0.616 | 0.592 |
| Logarithmic mean temperature difference | LMTD | $^oC$ | 29.7 | 32.1 | 31.5 |
| **Effectiveness** | ε | - | **0.41** | **0.41** | **0.43** |

Table 2-2: Observation and results of experiment for different mass flow rates

## 2.2 Sample Calculation

**Heat lost rate by hot LD (QH):**

Mass flow rate of LD on hot side (MH) = 22.36 kg/h =0.006211kg/s
Inlet temperature of LD on hot side (THI) = 73.3 ◦C
Outlet temperature of LD on hot side (THO) = 57.7 ◦C
Specific heat of LD at X=0.33 & TAVG = $THI+THO/$ 2 = 65.5 ◦C is CP = 2819.06 J/kg K
So, Heat lost rate by hot LD (QH) = MH * CP * (THI-THO) = 0.256 Kw

**Heat gain rate by cold LD (QC):**

Mass flow rate of LD on cold side (MC) = 76.08 kg/hour = 0.0211 kg/s
Inlet temperature of LD on cold side (TCI) = 31.8 ◦C
Outlet temperature of LD on cold side (TCO) = 34.9 ◦C
Specific heat of LD at X=0.33 & TAVG = $TCI+TCO$ 2 = 33.35 ◦C is CP = 2702.17J/kg K
So, Heat gain rate by cold LD (QC) = MC * CP * (TCI-TCO) = 0.169 kW

**Rate of heat lost to atmosphere from heat exchanger surface (QLOST):**

Rate of heat lost to atmosphere, QLOST = QH-QC = 0.086 kW

**Maximum possible heat transfer rate (QMAX):**

Heat capacity on hot side = MH * CP = 17.501 J/K s
Heat capacity on cold side = MC * CP = 57.04 J/K s
So, less is on cold side.
And, THI- TCI=41.5 ◦C
So, Maximum possible heat transfer rate (QMAX) = 0.592 kW

**Effectiveness of heat exchanger (ε):**

Effectiveness = *A*ctual heat transferred/Maximum possible heat transfer
= 0.43 OR 43%

**LMTD**

Θ1 = THI-TCO = 38.4◦C
Θ2=THO-TCI = 25.9◦C
So, LMTD = Θ1-Θ2/ln(Θ1/Θ2) = 31.5◦C

## 2.3 Analysis

Here, we find that the change in temperature of LD (TCI-TCO) was decreasing with increase in mass flow rate. Fig. 2.2 shows the same. Here, the heat transfer rate was increasing with increase in mass flow rate of LD. It is shown in fig. 2.3. We can see that the increase in heat transfer rate is not as much as it is in plate heat exchanger.

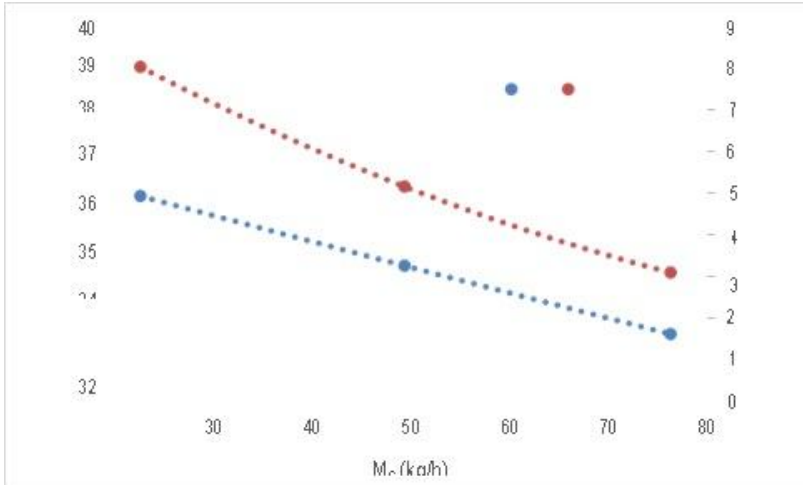


Figure 2.2: Change in temperature of cold LD& inlet temperature of cold LD with MC

### 2.4 Prediction of Heat Transfer Rate with SS Tube

The heat transfer rate is very poor in tube in tube heat exchanger, because inside tube was made of plastic and it has very poor conductivity. However, it must be good, if there is a metal tube inside the heat exchanger. So, theoretical calculation is done here to find the heat transfer rate for the same, if a steel tube is used in place of plastic tube. Here, the conductivity of plastic was replaced by same of steel.
The results for SS tube inside are calculated theoretically and summarized in table 2.3

| | Nomenclature | Unit | 1 | 2 | 3 |
|---|---|---|---|---|---|
| Mass flow rate of LD on cold side | $M_C$ | kg/h | 22.30 | 49.08 | 76.08 |
| Mass flow rate of LD on hot side | $M_H$ | kg/h | 24.70 | 22.56 | 22.36 |
| Temperature of hot LD at inlet | $T_{HI}$ | $^oC$ | 75 | 75 | 75 |
| Temperature of cold LD at inlet | $T_{CI}$ | $^oC$ | 30 | 30 | 30 |
| **Heat transfer rate** | **Q** | **kW** | **0.406** | **0.470** | **0.495** |
| Maximum possible heat transfer rate | $Q_{MAX}$ | kW | 0.748 | 0.748 | 0.748 |
| **Effectiveness** | ε | - | **0.54** | **0.62** | **0.66** |

Table 2-3: Calculated results for SS tube

#### 2.4.1 Sample Calculation

Mass flow rate of LD on cold side, MC =22.30 kg/h
Mass flow rate of LD on hot side, MH = 24.70 kg/h
Temperature of hot LD at inlet, THI = 75 oC
Temperature of cold LD at inlet, TCI = 30 oC
Inner diameter of inside tube di = 6.35 mm
Outer diameter of inside tube do = 4.38 mm
Inner diameter of outside tube D = 25.73 mm
Length of tube l = 3580 mm

**Desiccant Properties**

Following properties were calculated by for LD using formulas described in Appendix.

| Property | Nomenclature | Unit | Hot side (Temp.- 75 $^oC$) | Cold side (Temp.- 30 $^oC$) |
|---|---|---|---|---|
| Density | ρ | $kg/m^3$ | 1273.96 | 1301.07 |
| Specific heat | C | J/kg K | 2851.34 | 2688.54 |
| Thermal conductivity | K | W/m K | 0.62 | 0.58 |

| | | | | |
|---|---|---|---|---|
| Dynamic viscosity | ν | $m^2/s$ | 0.000001491 | 0.000003387 |

**Convective heat transfer coefficient on inner side of inside tube – hi**

Reynolds No. Re.=ρ∗v∗di/ μ =1210.62

Prandtl No. Pr. = μ∗C /K = 6.928

Nusselt No.

- Hydrodynamic entry length l1 = 0.058*Re*di = 0.3873 m
- Thermal entry length l2 = 0.058*Re*Pr*di = 2.6842 m
- Nu for simultaneously developing laminar flow, Nu1 = 1.953($Re*Pr*di/l1$ ) 0.33 = 8.898
- Nu for only thermally developing laminar flow,

  Nu2 = 4.364 + [0.086($Re*Pr*di/l2$ ) 1.33]/[ 1+0.1$*Pr*$($Re*di/l2$ ) 0.83] = 5.7993
- Nu for fully developed flow laminar flow, Nu3 = 4.34
- Nu (weighted AVG of all entry length) = {$Nu1*l1$+$Nu2*l2$+$Nu3*(l-(l1+l2)]/ l$ =5.7421

Convective heat transfer coefficient on inner side of inside tube.

hi = $Nu*K/di$ = 822.11 W/m2K

**Convective heat transfer coefficient on outer side of inside tube – ho**

Reynolds No. Re.=ρ∗v∗(D−$do$)/μ = 104

Prandtl No. Pr. = μ∗C/K = 19.52

Nusselt No.

- Hydrodynamic entry length l1 = 0.058*Re*D = 0.10097 m
- Thermal entry length l2 = 0.058*Re*Pr*D = 1.9713
- Nu for simultaneously developing laminar flow, Nu = 1.953($Re*Pr*d$ /$l1$ ) 0.33 = 14.27
- Nu for only thermally developing laminar flow,

  Nu2 = 4.364 + [0.086($Re*Pr*d/l2$ ) 1.33]/[1+0.1$*Pr*$($Re*d/l2$ ) 0.83] = 4.37
- Nu for fully developed flow laminar flow, Nu3 = 4.34
- Nu (weighted average of all entry length) = [$Nu1*l1$+$Nu2*l2$+$Nu3*(l-(l1+l2)]$ /$l$ = 21.27

Convective heat transfer coefficient on inner side of inside tube

ho = $Nu*K/di$ = 638.96 W/m2K

**Overall heat transfer coefficient for inner tube (for outside)**

Uo = 1/[$do/di$∗hi+ $do/2$ $*KSS*lndo/di$ + 1/$ho$] = 292.114 W/m^2K

**Heat transfer rate**

Heat capacity on hot side = MH* CH = 18.0025 W/K

Heat capacity on cold side = MC* CC = 17.978 W/K

Maximum heat capacity, CMAX = 18.0025 W/K (on hot side)

Minimum heat capacity, CMIN = 17.978 W/K (on cold side)

Heat capacity ratio, R = $CMIN/CMAX$ = 0.9986

Surface Area of outer side of inner tube, Ao = π*d0*l = 0.07339 m^2

NTU = Uo∗Ao/CMIN = 1.1924

Effectiveness for counter flow arrangement

ε = [1−$e$ ^−(1−$C$)$NTU$] /[1−$C*e$^−(1−$C$)$NTU$] = 0.5441

Maximum possible heat transfer rate, QMAX = CMIN *(THI-TCI) =746.99 W

Actual heat transfer rate Q = QMAX* ε = 406.43 W = 0.406 Kw

**Conclusion and Scope for Future Work**

**3.1 Conclusion**

For, tube in tube heat exchanger the mass flow rate on hot side was kept constant (22-24 kg/h) and on cold side was varied, which was 25, 50 and 75 kg/h. The heat transfer rate (according to heat lost by hot LD) was 0.234, 0.248 and 0.256 kW for mass flow rates 25, 50 and 75 kg/h respectively. The temperature rise was 8, 5.1 and 3.1oC for mass flow rates 25, 50 and 75 kg/h respectively.

**3.2 Scope for Future Work**

In future, the heat exchangers may be used in actual LDAC system. So, that more practical conclusions can be made

The set up may be used for more values of mass flow rates, other desiccants and different configurations of heat exchangers such as

- Other sizes of plate heat exchangers
- Tube in tube heat exchanger with metal tubes of various dimensions

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