Design and Development of Heat Pump for Large Capacity Fish Storage Tank

Assistant Professor: S.K.Malave

Neha Kokate[#1], Sagar Shimpi[#2], Sayali Bachhav[#3]

Department of Mechanical Engineering, Savitribai Phule Pune Univercity, India

.**Abstract** – This paper presents the overall design and development of heat pump for large capacity fish storage tank. Temperature is the most important factor with regard to quality and shelf life of fish. Temperature of the fish storage tank can be maintained upto28◦C using some water heating technologies available in the market. The proposed system will be able to maintain water temperature within 28-40◦C optimum range with reduced cost of electricity. The aim of this paper is to summarise the methodology to develop heat pump as per the requirements to enhance the life of the fish.

Keywords – Heat pump, Water temperature, Fish, large capacity.

1 Introduction

With the growing demand of seafood products across the world, the dynamic of seafood business is changing fast. For this industry, fish storage is the most important aspect. Optimum storage temperature is the most important factor to maintain the quality of fish as long as possible. Even slight variations from proper storage temperature can result in shorter shelf life and weaken immune system.

1.1 Need to mainatain optimum temperature

Most aquatic animals are cold blooded. Their metabolism speeds up and slows down according to surrounding temperature. Sometimes temperature variation may lead to white spot disease. Each organism is adapted to survive best at a given range of temperatures. If the tempearature changes too drastically, their metabolism will not function, decreasing their ability to survive and reproduce. Hence, the very basic requirement for fish storage is maintaining a temperature which is comforatable for them.

1.2 Competing technologies

There are several competing solutions to attain optimum temperature within the system such as solar heaters, boilers, electric heaters. However, out of these water heating technologies, heat pump is the most efficient and cost saver technology.

1.3 Heat pump

Heat pump is a machine that can pump heat in both directions. Heat pump can produce about 3kW thermal energy for every 1kW of energy consumed, giving an effective efficiency of 300%. High efficiency results in lower electric bills. It has lower maintainance cost and short payback period.

1.4 Environmental benefits

The environmental benefits, as lower carbon dioxide emissions, with heat pump will depend on how the electricity is generated and how the heat supply is provided before the implementation. Because heat pump systems do not burn fossil fuels for heat production, they generate far fewer greenhouse gas emissions than a conventional heating technology

.

1.5 Present scenario

Now-a-days, in almost all the aquariums electric heaters are used. These heaters are of different capacities. If these electric heaters are used for large capacity storage tank, electric bill will be higher. Whereas, in case of heat pump, as its COP is 2-3, electricity consumption is reduced to upto 60-67% for same capacity storage tank.

2 POTENTIAL LOADS

The first law of thermodynamic is applied for the tank system when an ideal mixing is considered in eqn (1.a) :
Considering a prototype model of 1000 litre capacity.

$$m\,cp\,dT/d\theta = Q_{net} - W \quad (1.a)$$

There is no net work done in tank, hence W=0. It is given in eqn (1.b):

$$Q_{net} = m\,cp\,dT/d\theta \quad (1.b)$$

And Q_{net} is represented in eqn (1.c)

$$Q_{net} = \sum\{Q_{in} - Q_{out}\} \quad (1.c)$$

Then

$$m\,cp\,dT/d\theta = \sum\{Q_{in} - Q_{out}\} \quad (1.d)$$

In this expression, $\sum Q_{out}$ represents all of the heat losses. The left hand side of equation (1.d) corresponds to the change of internal energy of the water tank. In the present study, the heat added to the tank through the heat pump is presented by (Q_{in}). This formula will be used in present study for the thermal analysis of water tank.

2.1 Evaporation process

2.1.1 Evaporation Mass Loss

Many evaporation rate empirical correlations are available with margin of errors, which depends on the range of the implemented experimental data in their formulation. The evaporation rate represents the principal component of the predicted heating load that is required for the water tank, it comoses about (65-70) % of total heat loss from a tank. Hence, a careful consideration has to be presented in the selection of such correlations for the design needs. Tanks typically lose more than (50 %) of the heat placed in the tank by the heater. The amount of evaporated water can be predicted from available empirical equation (2)

$$\dot{m}\,h_{fg} = A_s\{(30.6 + 32.1\,u_{wind})(p_{w,sat} - p_{a,Dew})\} \quad (2)$$

$$\dot{m}\,h_{fg} = 3.92\ \text{kW}$$

The pressure difference in the above equation has (mm. Hg) units. It is obvious that the evaporation rate is proportional to the environment and climate conditions such as temperature, humidity and wind speed.

2.1.2 Evaporation Heat Loss (Q_{evap})

The heat load supplied to the pool is required to compensate among others the heat rejected away from the surface due to evaporation. Most of the heat required for the evaporation is taken from the water itself. To maintain the water temperature heat must be supplied. The required heat supplied to cover the evaporation loss can be calculated from:

$$Q_{evap} = \dot{m}\, h_{fg} \quad (3)$$
$$= 3.92$$

Evaporation latent heat of water was taken at the pool temperature.

2.2 Convection Loss ($Q_{surface}$)

The heat loss from water surface due to convection should also be compensated as a part of the total design requirements. It mainly depends on the ambient air heat transfer coefficient and potential temperature difference between the ambient climate and tank value. The temperature difference is a time dependent variable, but for conservative steady state heat loss estimation, it may be considered at its highest possible value to maximize the losses. This heat loss was estimated from the following eqn (4)

$$Q_{surface} = \alpha_s \Delta T_{aw} A_s \quad (4)$$
$$=23.6 * 23 * 1$$
$$= 0.542 \text{ kW}$$

The heat transfer coefficient (α_s) for an outdoor tank was found by eqn (5)

$$\alpha_s = 3.1 + 4.1\, u_{wind} \quad (5)$$
$$= 3.1 + (4.1 * 5)$$
$$=23.6$$

2.3 Inertia Heating-up Load ($\dot{Q}_{Heat\text{-}up}$)

This component of total heat load represents the amount of energy rate to be added to raise the temperature of the pool from its initial to the set point value. This load can be calculated from eqn (6):

$$Q_{Heat\text{-}up} = (v\, \rho\, cp\, \Delta T_{water})/\Delta\theta \quad (6)$$
$$= 1ooo*4.19*(28\text{-}5)$$
$$= 26.7 \text{ kW.....for 3 hours}$$
$$= 9 \text{ kW ...for 1 hour}$$

The heating-up time (Δθ) is a major factor which controls the amount of energy required for the water tank during occupancy and preheating for the next day. Hence, a considerable attention should be paid for this part of the design load demand when a water tank is to be designed.

2.4 Side Walls Convection-Conduction ($Q_{Conv, Cond}$)

The heat loss through the walls of the tank is composed of two heat transfer modes. The convection and conduction modes are the predominant and estimated from eqn (7):

$$(Q_{Conv,Cond}) = U_{wall} A_{wall} \Delta T_{aw} \quad (7)$$
$$= 11 * 1 * 23$$
$$= 0.253 \text{ kW}$$

The overall heat transfer coefficient U_{wall} is determined by the material and thickness of the composite wall structure.

2.5 Ground Conduction (Q_{Gro})

Conduction between the water tank and ground is in most circumstances accounts for less than (1) % of the total energy loss from the tank. This component is usually ignored when the tank is sitting firmly on the ground due to the small temperature difference between the tank bottom and ground, hence (Q_{Gro}) =0.

2.6 Radiation Heat Loss (Q_{rad})

The long-wave radiation heat lost from the tank water surface is usually estimated from the general radiation formula in eqn (8.a) as:

$$Q_{rad} = A_s\, \varepsilon\, \sigma\, \{T_s^4 - T_{sky}^4\} \quad (8.a)$$
$$= 1 * 0.9 * 5.67 * 10^{-8} (301^4 - 278^4)$$
$$= 0.11408 \text{ kW}$$

Where, ε is given in eqn (8.b)

$$\varepsilon = 0.9 \quad (8.b)$$

And, σ is given in eqn (8.c)

$$\sigma = 5.67 * 10^{-8} (\text{W/m}^2\text{K}^4) \quad (8.c)$$

The sky temperature (T_{sky}) depends on the condition of the atmosphere, cloudy or clear and day or night time.

3. THERMAL DEMAND ASSESSMENT

There are many design schemes and procedure philosophies available in the open literature to estimate the energy management of a water tank with a scatter of accuracy limits. A simple procedure was suggested at the present work for preliminary heating load estimation for an over-ground outdoor water tank.

The evaporation process reduces the temperature of the tank surface due to the latent heat of vaporization drawn from the tank water body. The convective process cools or warms the liquid surface according to value of the ambient air. It cools if the air temperature is lower and warms if the air temperature is higher. Therefore, it depends mainly on the direction of heat flow, in or out of the water tank. The possible minimum temperature that the water surface can attain is the wet-bulb temperature of the ambient air due to equilibrium conditions.

The total design load to heat and maintain the tank at set point temperature may be expressed in eqn (9) as:

$$\dot{Q}_{Design} = \dot{Q}_{Heat\text{-}up} + \dot{Q}_{evap} + \dot{Q}_{Conv,Cond} + \dot{Q}_{surface} + \dot{Q}_{rad} - \dot{Q}_{rad,sw} \quad (9)$$

The evaporation component is the predominant among other losses mechanisms. It is inevitable and significant during the tank occupancy and composes almost the principal heat loss source.

Since the tank water and ambient air temperatures vary with time, hence the convection-conduction component could be a heat gain or loss with respect to atmosphere. This adds another complication to the assessment task of thermal mechanism during heating up of the tank. During night time the ambient temperature falls below the tank and heat loss is evident to the ambient. During the initial heating up, the tank temperature passes through both modes of loss and gain with time. For conservative heating load evaluation, the heat transfer through the tank walls was considered as a heat loss to the ambient temperature. Further, the short- wave heat gain ($Q_{rad,sw}$) due to radiation was neglected to be within the safety factor for the conservative thermal load analysis, hence eqn (10) gives:

$\dot{Q}_{Design} = \dot{Q}_{Heat\text{-}up} + \dot{Q}_{evap} + \dot{Q}_{Conv,Cond} + \dot{Q}_{surface} + \dot{Q}_{rad}$ (10)

= 9+ 3.92+ 0.542 +0.253 +0.11408

= 13.83 kW

4 METHODOLOGY

The energy and load assessment of the storage tank is quite difficult task to e handled in full details due to several different parameters which should be considered in their design. These variables experience a time dependent fluctuation during the day and night time. The heat balance is also being affected by the place of tank installation. For outdoor tank, the heat balance of the tank is governed by the weather conditions such as solar radiation, air and sky temperatures, humidity and wind speed.

Superheat 1-1'= 5°C

Subcooled 3'-3 = 5°C

$Sg_1 + C_{pv} \ln (T_{1sup}/T_1) = Sg_2 + C_{pv} (T_{2sup}/T_2)$

$1.7274 + 1.218 \ln (278/273) = 1.7059 + 1.218 (T_{sup}/T_2)$

$T_{2sup} = 334.77K$

$T_2 = 61.77°C$

Now,

$h_1 = hg_1 + C_{pv} (Tsup_1' - T_1)$

$= 398.7 + 1.218 (5)$

$= 404.79$

$h_2 = hg_2 + C_{pv} (Tsup_2 - T_2)$

$= 423.6 + 1.218(5)$

$= 429.69$

$h_3 = h_4 = h_3' - C_{pf} (T_3' - T_3)$

$= 271.6 - 1.569(5)$

$= 263.755$

Condensing effect = h_2-h_3 = 165.935

Mass flow rate of refrigerant = (Total heat load)/(h_2-h_3)

= (2.84*211)/(165.935)

= 2.6112 kg/min

4.1 Condensor calculation method:

Condensor side: - As the condenser side is in contact with main system, the desired output is obtained at the condenser side. The condenser system is nothing but the heat exchange taking place between refrigerant and water tank side. To get the the efficient rate of exchange of heat between water and between refrigerant side a suitable heat exchanger should be designed. So the parallel flow shell and tube heat exchanger is chosen as per the demand of the system. This arrangement gives rise totemperature increase of 23°C which is required.

Design of the shell and tube heat exchanger is done by nusselt no. method given by [1]:

$Nus = 0.36\, Res^{0.55}\, Prs^{1/3}$ [1]

Where *Prs* is the dimensionless group Prandtl and *Res* is the Reynolds number. The Nusselt no. and the Reynolds no. are defined by [2] and [3] respectively:

$Nus = (hs * Deq)/ks$ [2]

$Res = (Deq * vs * ps)/\mu s$ [3]

The equivalent diameter is a function of the outer diameter (dte) and the tube pitch (ltp), and also depends on the layout of heat exchanger given by [4]:

$Deq = (4\, ltp^2) / (\pi\, dte)$ [4]

The flow velocity is given by [5]:

$Vs = (ms) / (ps\, Ar)$ [5]

Where ms is the shell mass flow rate and Ar is the flow area between adjacent baffles, which can be described by [6]:

$Ar = Ds\, FAR\, lbc$ [6]

Where *Ds* is the shell diameter, lbc is the baffle spacing and *FAR* is the free area ratio, that is given by [7]:

$FAR = (ltp - dte) / ltp = 1 - (dte/ltp) = 1 - (1/rp)$ [7]

4.2 Tube side: - The tube side Nusselt no. is given by [8] and [9]:

$Nut = 0.023\, Ret^{0.11} Prt^n$ [8]

$Nut = (ht\, dti) / kt$ [9]

The Reynolds no. is given by [10]:

$Ret = (dti\, vt\, pt) / \mu t$ [10]

The pressure drop in the tube side flow (ΔPt), considering constant physical properties is given by [11]:

$(\Delta Pt/pt\, g) = (ft\, Npt\, l\, vt^2/ 2\, g\, dtl) + (K\, Npt\, vt^2/ 2g)$ [11]

The expression of the overall heat transfer coefficient (U) is [12]:

$U = 1/[(dte/dti\ ht) + (Rft\ dte/dti) + (dte\ \ln\ \{dte/dti\}/2\ ktube) + Rfs + (1/hs)]$ [12]

The logarithmic mean temperature difference (ΔTlm) described by [13]:

$\Delta Tlm = [(Thi - Tco) - (Tho - Tci)]/\ln[(Thi - Tco)/ (Tho - Tci)]$ [13]

The heat transfer rate equatin is given by [14]:

$Q = U\, Ar\, \Delta Tlm\, F$ [14]

Where Q is the heat load and Ar is the required area

4.3 Evaporator Calculations:

As evaporator side is in direct contact with the surrounding, the efficient way is to choose fin tubed heat exchanger so that maximum heat is absorbed from the surrounding.

$A_{root} = \pi Dr$

$Do = Dr + 2H$

$A_{f1} = (Do^2 - Dr^2)*(\pi/2)$

$Ar = [1 - (N_f * Y)]*A_{root}$

Dimensions:

Dr = 3/8 inches = 0.375 inches

Do = Dr + 2H = 0.768

H =5mm, Y = 2mm (0.0787 inch)

No of fins/square inches (Nf) = 5.64 fins/inches

No of tubes required:

$A_{root} = \pi Dr = \pi*0.375 = 1.178$ inches

$A_{f1} = (0.763^2 – 0.375^2)*(\pi/2) = 0.7056$

$A_{f2} = N_f *A_{f1} = 3.97$ inch^2

$Ar = [1- (N_f * Y)]*A_{root} = 0.5561*1.178 = 0.6551$ inches^2

$A_{f3} = (\pi* N_f*Y*Do)/2 = 0.5754$ inch^2

$APF = (A_{f1}+Ar+A_{f3})/12 = 0.1580$ inch^2/ inch length

= 101.93528 mm^2/ inch

= 40.13 mm^2/cm

= 4013 mm^2/m (Assuming tube size = 1m)

Area calculated from evaporator load = 1.6089 m^2

= $1.6089*10^5$ mm^2

No of tubes = 400.922

4.4Performance of R134a as an Alternate to R22-

Ozone Depletion and Global Warming has always been prime environmental alarming factor with major repercussion. Ozone layer is quite helpful in cleaning all the harmful ultraviolet rays of the sun through the absorbing maximum of the damaging ultraviolet radiation.

As per Montreal Protocol, R22 is going to be phase out due to its negative impacts on environment e.g. ozone depletion potential (ODP) and global warming potential (GWP). R-134a has zero ODP and considerably GWP as compared to R22.

5. RESULTS AND DICUSSION

The heating load estimation for a water tank depends mainly on the philosophy of its use. The limitations of its usage are in regard of outdoor or indoor, private or public, size, season and mostly the climate condition.

6. CONCLUSION

There are many ways of heating water. In this study large capacity of water needs to be heated with the help heat pump. A heat pump system can provide both heaing and cooling. It is least expensive method of heating. It is also the most energy efficient.

ACKNOWLEDGEMENT

It is indeed a great pleasure and moment of immense satisfaction for us to present the project report on “Design and Development of Heat Pump for Large Capacity Fish Storage Tank” amongst a wide panorama that provided us inspiring guidance and encouragement, we take the opportunity to thanks to those who gave us their indebted assistance. We wish to extend our cordial gratitude with profound thanks to our internal guide Prof. S.K. Malave Sir for his everlasting guidance. It was his inspiration and encouragement which helped us in completing our project. Our sincere thanks and deep gratitude to Head of Department,Dr .N. P. Sherje sir and other faculty member; but also, to all those individuals involved both directly and indirectly for their help in all aspect of the project. At last but not least we express our sincere thanks to our Institute’s Principal Dr.A. V. Deshpande for providing us infrastructure and technical environment.

REFERENCES

[1] Fabián Correa, Cristian Cuevas:“Air-water heat pump modelling for residential heating and domestic hot water in chile”.

[2]. YuexiaLv[a,b,*], PengfeiSi[c,d,†], XiangyangRong[d], JinyueYan[e,f]:“An Optimization Method for CCHP and River Water Source Heat Pump Combined System ”.

[3] Young-JinBaik[1], Minsung Kim*, Ki-Chang Chang, Young-Soo Lee, Ho-Sang :“Potential to enhance performance of seawater source heat pump by series operation”.

[4] Helge Averfalk[a,*], Paul Ingvarsson[b], Urban Persson[a], Mei Gong[a], Sven Werner[a]:“Large heat pumps in Swedish district heating systems ”.

[5] Juan M. Hernandez, Miguel Leon-Santana, Carmelo J. Leon: “The Role of the Water Temperature in the Optimal Management of marine Aquaculture

[6] Roger Coggan: “Seasonal and Annual Growth Rates In

Antarctic Fish *NototheniaCoriiceps*

[7] Susmita Dasgupta, Mainul Huq, Md Golam Mustafa, Md. IstiakSobhan, David Wheeler: “The Impact of Aquatic Salinization on Fish Habitats and Poor Communities in a Changing Climate: Evidence from Southwest Coastal Bangladesh.

[8] JozsefNyers, Laszlo Garbai, Arpad Nyers: “A Modified Mathematical Model of Heat Pumps Condenser for Analytical Optimization”

[9] UgurCakir, Kemal Comakh: “Exergetic Interrelation between a Heat Pump and Components”

[10] FarazAfsari, Omer Comakli, Sendogan K Aragoz and hadiGhasemiZavaragh: “Heat pump and refrigeration device using several refrigerants”

[11] Ali H. Tarrad’s: “Heating mechanism And energy analysis for over ground outdoor swimming pool technology”

[12] Julia C. Lemos, André L.H. Costa, Miguel J. Bagajewicz “Linear Method for the Design of Shell and Tube Heat Exchangers Including Fouling Modeling”.

[13] Wangb,*, Xinlei Mab, Qiang Wangc “Study on the performance of small tube diameter R-290 fin tube evaporator”.