How to Cite:
Vairavel, D. K., Suresh, M., Kumar, C. S. R., Rajamurugan, T. V., Ali, S. B. R., & Malathi, V. (2022). Power filling process in tube heat exchanger. *International Journal of Health Sciences*, *6*(S3), 696–720. https://doi.org/10.53730/ijhs.v6nS3.5424

Power Filling Process in Tube Heat Exchanger

Dinesh Kumar Vairavel
Assistant Professor, Department of Mechanical Engineering, Kalasalingam Academy of Research and Education, India
Email: autodinesh@gmail.com

M. Suresh
Assistant Professor, Department of Mechanical Engineering, Karunya Institute of technology and Sciences, Coimbatore, Tamilnadu, India
Email: thillaisuresh@gmail.com

C. Selin Ravi Kumar
Department. Civil Engineering, College name and address. Malla Reddy Engineering College (Autonomous), Hyderabad

T.V.Rajamurugan
Assistant Professor, Government College of engineering, Srirangam, Trichy

S. B. Riswan Ali
Assistant Professor of chemical engineering, Annamalai University, Annamalai Nagar

Malathi V.
Research scholar, Department of Electronics and Communication Engineering, JAIN Deemed to be University

***Abstract*---** Fin tube heat exchangers are used in many heating and cooling devices for the purpose of heat transfer from one element to other by conduction and convection. Due to their designs fin tube heat exchangers also provide a control space or volume for carrying out chemical reactions by extracting heat from the surrounding or liberating heat to the surrounding while undergoing an endothermic or exothermic reaction respectively. The Thermally Activated Cooling (TAC) system developed by Thermax Ltd. is a device one of its kind. It comprises of metallic powder which undergoes endothermic and exothermic reactions at different temperature. When arranged in a proper cycle desired cooling effect can be obtained via this process, The fin tube heat exchangers comprise of metallic powder and provide a control volume for reactions and a conductive surface for heat transfer. The amount of metallic powder in the heat exchanger is

International Journal of Health Sciences ISSN 2550-6978 E-ISSN 2550-696X © 2022.
Corresponding author: *Suresh, M.; Email: thillaisuresh@gmail.com*
Manuscript submitted: 18 Nov 2021, Manuscript revised: 09 Feb 2022, Accepted for publication: 27 March 2022

proportional to its performance, which is directly proportional to the COP of refrigeration system. The agenda of this project is to enhance the material filling process which is a crucial stage and requires a lot of time. Currently the tilling process is done manually with conjunction to vibrating system which requires at least 5 hours to achieve the 90% of theoretical weight of powder in the heat exchangers, but when moving to mass production this time duration is not appreciable and has to be reduced to as low as possible for higher production rate. The project focuses on the process enhancement and its execution on large scale.

***Keywords*---**filling process, heat exchanger, endothermic, exothermic, process enhancement.

Introduction

Theory

1. Taking an example of filling flour in a box. A normal tendency is filling and frequent tapping so that the material gets compact and more material can be stored in a given volume.
2. Similar is the concept for the material filling, the manual process of tapping is done by a vibrating machine. But unlike the example, the actual setup has a very small internal diameter and also an internal filter and foam assembly which hinders the solid flow of powder to a great extent.
3. So to increase the solid flow rate we need to force the particles in the tube. This can be achieved by either pushing the particles with a compressed air flow or to pull the particle inside by creating vacuum/ suction inside the tube.
4. We choose pulling or suction method over compressed air flow due to the following reason:
   a) The stagnation due to filter which will increase the turbulence inside and won't Let the particles settle.
   b) The integration of compression system with the vibration is very complex and will result into a bulky system.
   c) A compression system will be at the top side above the hopper, which will reduce the scope of further improvement with regard to automatic filling system.
5. The vacuum pump can be connected to a hollow chamber which can be mounted on the vibrating machine thus integrating them,
6. This leaves the above setup for hopper arrangement and regarding automation leaves room for automatic weighing machine and different control systems.
7. Maximum vacuum which can be achieved ideally is -1 bar (absolute zero pressure), so for increasing the pull force, flow rate of the pump has to be increased.

8. Also when working with wire mesh filter of micron size, the higher the Reynold's number lower will be the friction factor. Hence high flow rate pumps are preferable.

Following is the schematic arrangement for vacuurn-vibra.tion integral material filling system.

Literature survey and selection of pump

Employing finned-tubes with different configurations [1–9] and adding particles with higher thermal conductivity than the PCM [1, 10–19] enhance the effective thermal conductivity of PCM used in PCM-based thermal systems

The factors to be considered while selecting a pump are as follows:

1. Maximum Vacuum which pump can achieve,
2. Flow rate of the pump.
3. Power rating of the pump.
4. No. of units to be loaded.

To compare different pumps, their specifications were collected from the vendor. The following table show types of pump and their parameters:

Table 1
Types of pumps and their different parameters

Sr. no.	Pump Type	Parameters		
		Power (HP)	Flow rate (cu. m/hr)	Max. vacuum (mm of Hg)
1	Rotary vane vacuum pumps	0.25 – 7.5 HP	5 -120 m3 / hr	600 to 710 mm Hg
2	Dry type rotary vane vacuum pump	0.25 - 7.5 HP	3-120m3/hr	240 to 450 mm Hg
3	Rotary vane oil sealed vacuum pumps	0.25 - 20 HP	3-600 m3 /hr	<759 mm Hg
4	Liquid Ring Vacuum Pumps	1 - 150 HP	18 - 2000 m3/hr	720 to 730 mm Hg
5	Rocking piston vacuum pumps	0.25 HP	1.5 to 4.5 m3/ hr	680 MM Hg

a) The rocking piston vacuum pump has very low flow rate and low vacuum in comparison to others so is it has to be ruled out.
b) Out of the remaining 4, the dry rotary vane vacuum pump has moderate flow rate for high powers but the vacuum of pump is low.
c) The maximum power (7.5 HP) rotary vane vacuum pump has high flow rate of 120 m^3/hr and a vacuum of 710 mm of Hg, but due to high power rating the cost of this pump is high. It will also increase the size of the overall system.

d) This leaves us to oil sealed and liquid ring vacuum pumps. For a given flow the power rating of oil sealed pump is higher than that of the liquid ring pump. The vacuum of oil sealed pump is higher than liquid ring by 30-40 mm of Hg, But the operating cost of oil seal is very high because it requires continuous supply of special grade oil at the rate of 4-6 LPM for its working. Whereas the liquid ring pump uses normal water for is operation,

e) Liquid ring pump has high vacuum of 720 mom Hg and high flow rate 18 m3*1* hr for 1 HP power rating which makes it an ideal choice of pump.

f) 1 HP pump has been used over higher power rating because of loading condition. During testing phase of this idea we decided to load the pump with only 1 heat exchanger at a time for tilling operation. So, as the load is less we choose lower power rating pump to reduce the capital cost.

g) Also a calculation of ReynoId's number and Euler's number is show below which supports the decision.

Screen models

The deviations between the measured pressure loss and the theoretical prediction are based on the problem to determine and to consider the real pore geometry. Formerly investigations have shown that the main geometry parameters are the screen porosity and the effective pore diameter, however both parameters have to he estimated and cannot be measured directly. In this study the geometry influence is considered using the screen porosity E, which is defined as the ratio of void volume (Vvoid) and total volume *(Vtorai),* and a characteristic diameter *DM* which is defined by the ratio of void volume and wire surface area *(Awire):*

$$\varepsilon = \frac{\mathrm{Vtotal} - \mathrm{Vwire}}{Vtotal}$$

$$Dch = \frac{Vtotal - Vwire}{Awire}$$

Wire volume and surface area are computable with four definite manufacturer properties: number of shute *(try)* and warp wires per inch (nw) and the shute (d5) and warp wire diameters (dw). A weave is characterized by these properties and they do not vary between different manufacturers. No further information is needed or has to be determined,

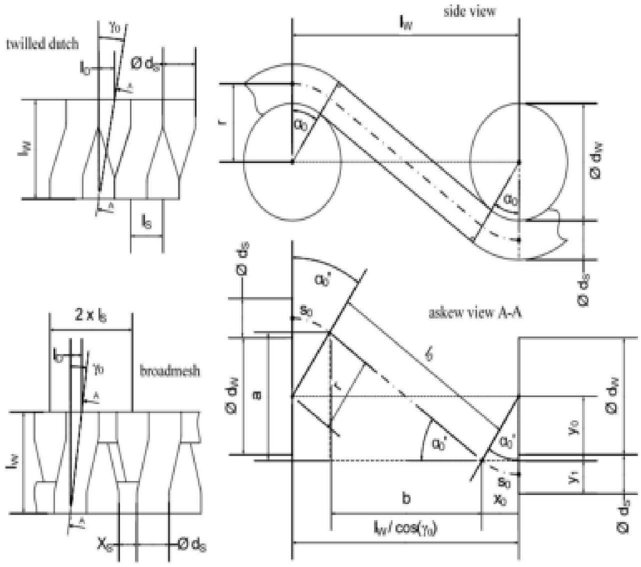


Figure 1. Geometry of metal screen

A description of the screen geometry is needed to calculate the wire volumes and surface areas of shute and warp wires. There is no deformation of the warp wires assumed, and they can be treated as straight cylinders. On the contrary the shute wires are deformed in two directions: they are passing over and under the warp wires alternately and changing direction through displacement due to the close packing of shute wires. A major influence is due to the deformation angle *a& it is* assumed that the askew angle nearly equals the straight angle:

Following geometric values are defined:

Shute wire radius of curvature: $r = \frac{ds+dw}{2}$

From the manufacturer we have input of:

ds = 40 micron

dw = 70micron

Therefore,

$$r = \frac{40+70}{2}$$

$$r = 55\ micron$$

$$r = 0.055mm$$

Now, the weave is $200\ \times 1400$ per inch (per 25.4 mm)

Therefore, $ls = \frac{25.4\times 2}{1400}$

$$ls = 0.03628\ mm$$

And, $l_W = \frac{25.4}{200}$

$$l_W = 0.127\ mm$$

Average gap between shute wires: $Xs = 2(ls - ds)$

$Xs > 0$ for broad mesh weaves,

$Xs = 0$ for typical dutch twill weave.

Calculating, Xs we get

$$Xs = -8.64 \times 10^{-3}$$

Therefore, Xs is taken to be 0.

Length of displacement: $l_D = \frac{-ls}{4ds}\ Xs + \frac{ls}{2}$

$$l_D = l_S/2 \qquad \text{for general dutch twill,}$$

$$l_D = \frac{0..03628}{2} = 0.01814\ mm$$

Displacement angle: $\gamma_o = \tan^{-1}(\frac{l_D}{l_W})$

$$\gamma_o = \tan^{-1}(\frac{0.01814}{0.127})$$

$$\gamma_o = 8.1288°$$

Auxiliary geometrical lengths: $x_0(\propto_0) = r\ \times \sin(\propto_0)$

$$y_0\,(\propto_0) = r \times \cos(\propto_0)$$

$$y_1\,(\propto_0) = \frac{d_W}{2} + d_S - y_0\,(\propto_0)$$

$$a_0(\propto_0) = d_w + 2d_S - 2y_1\,(\propto_0)$$

$$b_0\,(\propto_0) = \frac{l_W}{\cos\gamma_0} - 2x_0(\propto_0)$$

$$tan(\propto_0) = \frac{a_0(\propto_0)}{b_0\,(\propto_0)}$$

Now by combining the above equations we get,

$$\sin(\propto_0) = \frac{2\, r\ \cos \gamma o}{l_W}$$

$$\propto_0 = 1.03027\, radians = 59.03°$$

Therefore,

$$x_0(\propto_0) = r\ \times \sin(\propto_0) = 0.055 \times \sin 59.03$$

$$= 0.04716 \text{ mm}$$

$$y_0\,(\propto_0) = r \times \cos(\propto_0) = 0.055 \times \cos 59.03$$

$$= 0.028302 \text{ mm}$$

$$y_1\,(\propto_0) = \frac{d_W}{2} + d_S - y_0\,(\propto_0)$$

$$= \frac{0.070}{2} + 0.040 - 0.028302$$

$$= 0.046698 \text{ mm}$$

$$a_0(\propto_0) = d_w + 2d_S - 2y_1\,(\propto_0)$$

$$= 0.070 + 2 * 0.040 - 2 * 0.046698$$

$$= 0.056604 \text{ mm}$$

$$b_0\,(\propto_0) = \frac{l_W}{\cos \gamma_0} - 2x_0(\propto_0)$$

$$= \frac{0.127}{0.9899} - 2 * 0.04716$$

$$= 0.03397 \text{ mm}$$

Section lengths of shute wires: $l_0(\propto_0) = \sqrt{(a_0(\propto_0)^2 + b_0\,(\propto_0)^2)}$

$$= \sqrt{(0.056604^2 + 0.03397^2)}$$

$$l_0(\propto_0) = 0.066014 \text{ mm}$$

$$s_0(\propto_0) = r \times \propto_0$$

$$= 0.055 * 1.03027$$

$$s_0(\propto_0) = 0.056664 \text{ mm}$$

Total length of a shute: $L0s = \frac{2\,l_w}{\cos\gamma o} + 2(2\,s_0 + l_0)$

$$= \frac{2*0.127}{0.9899} + 2(2*0.056664 + 0.066014)$$

$$L0s \quad = 0.\ 61528 \text{ mm}$$

Clinched shute wire diameter: $d_S = \sqrt{{d_{S0}}^2 \times \frac{4\,l_W}{L0_S}}$

$$= \sqrt{0.040^2 \times \frac{4*0.127}{0.61528}}$$

$$= 0.03635 \text{ mm}$$

ds_0 = manufacturers diameter; $ds < ds_0$

It is now possible to calculate the unit volumes and surface areas of shute and warp wires.

Total unit volume:

$$V_{total} = 4\,l_W \times 2l_S \times (d_W + 2d_S)$$

$$= 4 * 0.127 \times 2 * 0.03628 \times (0.07 + 2 * 0.03635)$$

$$V_{total} = 5.26 \times 10^{-3}\,mm^3$$

Total wire volume:

$$V_{wire} = 4\left(\frac{\pi}{4} \times d_S{}^2 \times L0s\right) + 4\left(\frac{\pi}{4} \times d_w{}^2 \times 2l_s\right)$$

$$V_{wire} = 3.671 \times 10^{-3}\ mm^3$$

Total wire surface area:

$$A_{wire} = 4(\pi \times d_S \times L0s) + 4(\pi \times d_w \times 2l_s)$$

$$= 4(\pi \times 0.03635 \times 0.61528) + 4(\pi \times 0.07 \times 2 * 0.03628)$$

$$A_{wire} = 0.3449\ mm^2$$

Now using the above values, we can calculate the porosity and the characteristic diameter of the pores, as following:

1. Porosity,

$$\varepsilon = \frac{\text{Vtotal} - \text{Vwire}}{Vtotal}$$

$$\varepsilon = \frac{5.26 \times 10^{-3} - 3.671 \times 10^{-3}}{5.26 \times 10^{-3}}$$

$$\varepsilon = 0.3021$$

$$\%\varepsilon = 30.21$$

2. Characteristic diameter,

$$Dch = \frac{Vtotal - Vwire}{Awire}$$

$$Dch = \frac{5.26 \times 10^{-3} - 3.671 \times 10^{-3}}{0.3449}$$

$$Dch = 5.03 \times 10^{-3}\ mm$$

Experimental references for optimization

For the same 200 x 1400 per inch dutch twilled weave, many experiment. computational and analytical activities were carried out to measure the friction factor r, Euler number Eu against Reynold's number *Re* (change in

flow rate through filter). A comparison of all the methods with the experimental results was done by *Alexander Fischer and Jens Gersimann* at German Aerospace Center (DLR) and published in their paper "Flow Resistance of *Metallic* Screens in Liquid, Gaseous and Cryogenic Flow". Using the comparison, we can decide an optimized range of Reynold's number against pressure loss and power requirement for operation.
The experimental data and theoretical results calculated by different scientist are compared. From the graph, The dimensionless correlations for the DTW 200x1400 screens are given by Armour & Cannon. They derived a general prediction for the pressure loss of the Eve common weave types. Their equations are applicable to calculate the influencing geometry parameters and to estimate the flow resistance. Only for dutch weave types the effective pore diameter is not provided and the information has to be determined by experiment or taken from manufacturers information. Fig. shows the comparison of the considered prediction formulations by (Blatt, Cady. Belov. Ergun and Erhardt) compared to literature data for the 200x1400.

The predictions of Belov and Ergun show no agreement with the experimental data. Cady's correlations fit well with the experimental data, but the coefficients are valid only for two dutch twilled and one broad mesh weave. Erhardt defined a pressure loss prediction based on flow properties. He did not consider the screen and the oefficients determined for a single screen are varying for different manufacturers. The general correlation from Armour & Cannon predicts higher friction factors than measured in the experiments, Whereas the correlation of Blatt predicts lower values for *fr*.

Looking at both the graphs. we can say that relations developed by Armour, Cady and Erhardt are close to the results. We will go with the relations given by Cady as those formulas agree with experimental result for more number of meshes than others. Following are formulas and constants given by Cady,

$$f_r = \frac{\propto}{R_E} + \beta$$

Where,

β and $\propto$ constants and have values as - $\propto = 4 \ and \ \beta = 0.2$

Going again to the pump data,
Flow rate of pump = 18 m3ihr For a square inch of filter sheet,
Area of sheet = 64-5.16 mm' Using equation for flow.

Flow rate = area of sheet $\times$ *porosity* $\times$ velocity through screen

$$\frac{18}{3600} = \frac{645.16 \times 0.3}{10^6} \times u_{screen}$$

$$u_{screen} = 25.833\, m/s$$

Reynold,s number,

$$R_e = \frac{u_{screen} \times Dch}{\nu}$$

$$R_e = \frac{25.833 \times 5.03 \times 10^{-6}}{16.97 \times 10^{-6}}$$

$$R_e = 7.657$$

Euler's number,

$$E_u = \frac{A}{R_e} + B$$

$$A = 68.01$$

$$B = 8.01$$

$$E_u = \frac{68.01}{7.657} + 8.01$$

$$E_u = 16.892$$

So for 1HP and 18 m3ihr the value calculated for an inch square of filter sheet the value of Reynold's number is high, [tiler's is low, therefore the friction factor is loss, thus, the pressure loss and power loss will be. less.
Also from the graph, we can say that while moving to higher flow rate pumps for same inch square filter sheet the Reynold's number will increase but the Euler's number will not decrease to that extent so for practical purposes the Reynold's number should be kept between 0.1 to 10 for optimization.

So when loading only heat exchanger of 16 tubes 1HP pump is sufficient and give the required suction. And as the loading increases for usual working with 4 heat exchangers, the pump capacity should be increased as the increased number of tubes will compensate for increased flow rate and keep the ReynoId's number in the optimum range.

Equipments

1 Pipes - IA inch pipes: length - 6m and 2.-1 inch pipes: length - 8m and 2m
2 Liquid Ring Vacuum Pump: I nos.
3 Vibrating machine: 1 nos.

4 Holding fixture: I. nos.
5 Aluminium connecting link: 1 nos. (200 ruin)
6 Connectors: 112" male BSP connectors - 1 nos 1/2" female BSP connectors - l nos. 1" to 112 "female BSP adapter - I nos.
7 I/2" Float valve.
8 Drum: Capacity = 250 litre

9. The holding fixture is bolted onto vibrating machine using two M6 x 25 bolts.
10. A 1" hose pipe is connected to suction inlet of the pump and other end is connected to a 1" to ½" BSP male type converter which is connects with the adapter of holding fixture thus forming a complete link.
11. The discharge of the pump is connected to the 1" pipe whose outlet goes into the drain.

Set up
Schematic layout

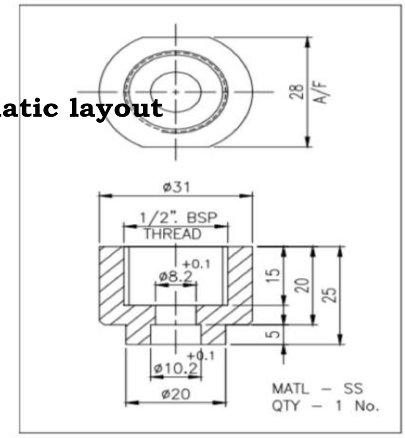
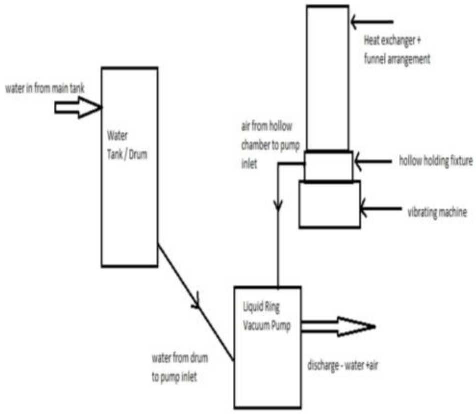


Fig.7 10mm to ½" BSP female adapter



Description

- The main head tank is connected to the intermediate tanIddrurn of capacity of 250 lit, by a 1/2" hose pipe. The hose pipe has a ih" BSP female which is connected to the male side of the float valve
- At an approximate height of 314th of the height of tank a 1/2" hole is made to attach a 1/2" float valve and at the foot of the drum a similar W hole is made to connect a ball valve.
- The float valve is used to maintain a constant water level in the drum so that. A constant flow rate of water can be a maintained as the pump requires a continuous supply of 4-5 LPM tbr its working.
- The other end of the ball valve is connected to a 1/4" inch pipe and the other end of the pipe is connected to a 1/2" BSP male

type connector which locks into female thread of water inlet of vacuum pump.

- The pump has an inlet and outlet section of l",
- The holding chamber fixture a hollow and a40 l 0.2 mm (earlier/5 6,5 nun, increased due to unavailability) hole on its vertical side, which gives and opening to connect the chamber to the inlet side of the pump.

7. A $\phi 10 \times 20$ mm aluminium tube is welded at the above mention hole.
8. The other end of the tube is welded to a 10mm to ¼" BSP female adapter.

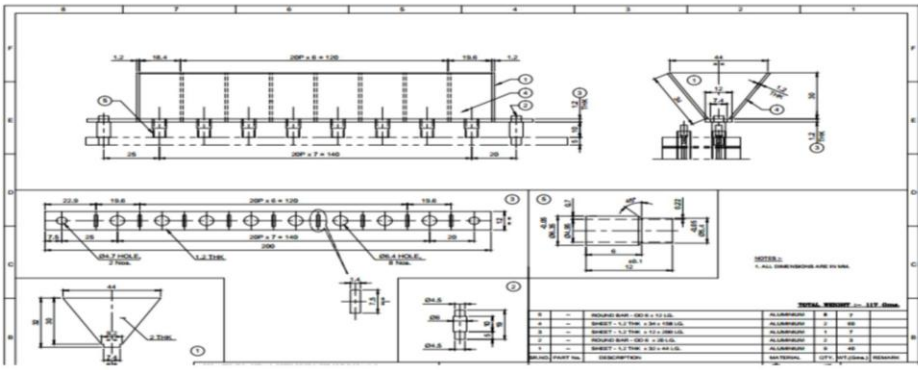


Fig.8 eight-hole hopper arrangement.

- The heat exchanger sits on the holding fixture and its extruded connectors go into the shoulder of the fixture.
- The shoulder ID is 010,2 mm and the OD of the connector is 06 mm, so to make the arrangement air tight rubber bushes are used,
- The rubber bush is a silicon rubber having shore hardness of 30A and dimensions
  are shown in the sketch of holding fixture.
- At top of the heat exchanger a 8 hole hopper is attached where in the material is poured.

Electrical requirements

1. Electrical connections for pump — 240V, 50 Hz, 34) supply.
The pump is connected to the main supply using a power contactor for voltage control and safety of pump in case of overloading.
2. Single phase supply for vibrating machine.

Fig.9 (b)

Procedure

- The heat exchanger is clamped as tightly as possible on the holding fixture without damaging its fins. Before doing so, the empty weight of the heat exchanger is measured.
- Operating the system is very easy, the pipe linking owlet of drum and water inlet ofvacuum pump is removed from drum end and the ball valve is adjusted to get flow rate of 4-5 LPM, A I lit. beaker and a stopwatch are used to measure the flow rate. Once the flow rate is achieved the pipe is connected back to the drum end without changing the position of ball valve,
- Now for starting the pump two people are needed. One gradually opens the water inlet valve of pump and other one at the same time switches on the pump motor and then the water inlet valve is opened completely,
- The pump has now started and the vibrating machine is also started and its Frequency is kept to maximum,
- The 8 hole hopper/funnel is now, put into half 8 side of the heat exchanger and the other half 8 holes of the heat exchanger are closed using plastic caps,
- Measured amount of powder is taken into a beaker and using a spatula is feed into each funnel compartment slowly till all the tubes are filled with powder and the process is repeated with the other 8 half portion,
- The starting time and ending time of powder filling is noted.
- Then the heat exchanger is undamped and the final weight of the heat exchanger is measured. The weight of the catalyst put inside is calculated by subtracting the empty weight from final weight.

- The vibration machine and pump motor are switched off and the water inlet valve is closed.
- The weight of the powder inside the heat exchanger is now known and is compared with theoretical weight and average weight using earlier method for dimensional similar heat exchanger. Also the time required data of earlier procedure is compared with the new developed method,

Observation

During the time of operation, we encountered and unexpected problem which changed the whole direction of our focus from material tilling towards the problem, that's why we call it 'tangent to the project'.

Toneent to the project

A heat exchanger of following dimension was manufactured following the SOP: Fin length =. 290 mm
Total length = 310 nun Number of tubes = 16 Empty weight = 665.6 g
The heat exchanger was clamped onto the holding fixture and by following the procedure everything was done properly. With everything working fine, the motor running and with the vibration machine on, we started pouring the powder material in the hopper. A measured amount of 250g. of powder was taken_ Expected weight in the 8 tube half portion was 220g., so ideally we should be left with 30g. at the end, even if taking 10g. for general losses due to handling the beaker should be left with 20g in it, But in the initial I5 min. the half portion consumed about 240g. which is not even possible ideally. So we checked the discharge, the water was dark grey in colour, the colour of powder. It happened that, the was powder flowing through the filters, came into the pumps and was out from discharge. The filters were leaking.

The reason for leakage through filter could be as follows:

- The powder particle size: if the size of powder particles was less than
  probability distribution function which showed that less than 10% of particles had size greater than 10 microns. So this could be one of the reason but not solely responsible for leakage.
- LASER welding: the seam of the filter is joined through LASER welding. So when inspecting randomly picked few filters from the batch under optical microscope. The size of these holes were measured using the microscope and is magnification, it turned out that the average size of these holes was 200 microns which is twice the size of maximum size of the powder. Also such holes were observed to be present at every 10 to 15 min distance, so larger the size of the filter, more will be the numbers of holes and more will be the loss of Material. Off the above mention possible reasons, the contribution of the tatter in the problem

occurred is very high compared to the former. This can be said on the basis of past experiences.

This problem was also checked in the ongoing material filling process for prototype manufacturing using only vibration for material filling. This problem also occurred there but the loss of material was less. The loss in vacuum-vibration integrated system is comparatively more because the suction aids the leakage, but this should not be considered as a disadvantage of the system as blow holes originally are undesirable and even if there, then the size of hales should be less than 5 microns. Also the latter problem can also cause difficulties during further stage of activation process of material where a heat exchanger is pressurised and vacuumed in a cycle several times.

Problems due to leakage

1. Loss of material: the powder is a composition on several different elements (cannot mention the names due to company rules and confidentiality reasons), but can surely that the overall cost of production of the powder is high and is also the key element of the cooling system. The powder coming out the discharge or fell on the floor for that matter is contaminated and has to be discarded completely as it cannot be reused. So with the high cost and no scope for reuse of powder, its loss not affordable and appreciable.
2. Insufficient/inefficient filling: as the solid flows out continuously with a rate, so isit difficult to fill it with desired quantity for material. One way to minimize the loss of material was to decrease the vibration frequency but again it will lead to less compactness of powder in the tubes, lower tap density and less amount of material than desired,

Two trials were conducted, one 290L and other 5551, heat exchanger for filling on vacuum- vibration integrated system. The observation and results of the trials are as follows:

Test for 2901

Date - 7(11/17
Fin length = 290 mm Total length = 310 mm Empty weight = 665.5 g
Total weight of powder taken for filling = 500 g
Ideal capacity of heat exchanger = 456 g (as per drawings) Average practical amount filled in heat exchanger = 410 g Time duration of trial = 30 minutes
After 30 min. the beaker was empty and all the 500 g powder was consumed by the heat exchanger, but this is not possible as it has an ideal capacity of only 456 g. Final weight of heal exchanger .= 1002.5
Weight of material in heat exchanger = final weight - initial weight = 1002.5 - 665.5
Weight of material in heat exchanger = 337 g

Amount of material lost .7 amount taken in beaker- material wt. in HX = 500 - 337
Amount of material lost 7. I 63 g

Test for 5551

Date - 09/11117
Fin length =. 555 mm
Total length 570 mm
Empty weight = 1206.5 g
Total weight of powder taken for filling = 900 g
Ideal capacity of heat exchanger = 860 g (as per drawings)
Average practical amount filled in heat exchanger = 794,24 g Time duration of trial = 45 minutes
After 30 min. the beaker was empty and all the 900 g powder was consumed by the heat exchanger, but this is not possible as it has an ideal capacity of only 860 g.

Final weight of heat exchanger = 1826 g
Weight of material in heat exchanger = final weight - initial weight 1826.0 - 1206.5
Weight of material in heat exchanger = 620 g
Amount of material lost due to leakage = amount taken in beaker - material wt. in HX
= 900 - 620
Amount of material lost due to leakage = 280 g
From the results of the two trials conducted we can say that, loss of material increases with increase in size of heat exchanger. Also loss in a single unit is very high, which is not appreciable and has to he minimize which can only be achieved by reducing the number of blow holes or by completely eliminating them.

Solution to the problems

We came up with three possible solutions for minimizing blow holes occurred during welding of filter mesh. Every solution with a justified test and its advantages, disadvantagesi difficulties associated with its execution are mentioned as follows:

Soldering

Instead of welding, soldering can also be used as a joining process for filter manufacturing. As soldering uses filler material, the formation of blow holes can he eliminated and a leak proof joint can be achieved.
To check this, a heat exchanger with filter manufactured using soldering technique was made and tested for leakage. Test:

- As material to be soldered is stainless steel, normal solder filler and flux cannot be used.
- Lead free solder AC305 (Tin (Sn) - 96.5%, Silver (Ag) - 3.0%, Copper

- (Cu) - 0.5%) was used as filler and the flux used was acidic. The flux used was SOLMUX-303, an acidic flux with appropriate mixture of Hydrochloric acid, Ammonium chloride and Zinc chloride. The solder alloy wz.v., supplied by "Harsh solders, Waluj, Aurangabad" and the was purchased from "Mathure fluxes, Thane".
- The joints of filter assembly, the seam and connectors at both ends were soldered. Eight such filters were manufactured. Also the process of brazing of connectors to heat exchanger tubes was replaced by brazing.
- Before brazing these filters were checked for holes on seam under optical microscope, A few were found and repaired immediately,
- The heat exchanger the was set up for material filling by following the method given in the procedure section.

The test of the result was noted and is as follows.: Test for 2901- soldered heat exchanger:

Date — 15/11/17
Fin length = 290 mm Total length = 310 mm Empty weight = 354 g
Total weight of powder taken for filling .= 250 g
Ideal capacity of heat exchanger .= 228 g (as, per drawings)
Average practical amount filled in heat exchanger = 205 g Time duration of trial = 30 minutes
During the trial. discharge of the pump was monitored continuously. The coicar of water was clear transparent and no leakage or loss of material was observed.
After 30 mitts the tubes were full and the heat exchangers final weight was measured.
Final weight of heat exchanger = 542 g
Weight of material left in the beaker = 58 g
Weight of material in heat exchanger = final weight — initial weight
= 542 — 354
Weight of material in heat exchanger = 188 g
Amount of material lost = amount taken in beaker — material wt, ire 1-IX — material left
= 250 — 188 — 58
Amount of material lost = 4g
This loss of 4g gram is general loss occurred during material filling due to handling and spillage.
Therefore, can be used as an alternative to LASER welding, Advantages:
a. The blow holes are eliminated completely_
b. Solder is very cheaper compared to LASER welding.
c. Loss of material is controlled.

Dis advantages

1. Soldering with SAC305 cannot be used as its melting point is 220°C and as the next process is brazing. The conduction of heat from brazing zone to filter will meh the solder. So hard

soldering will have to applied with filler of silver copper or silver-tin alloy.

2. LASER welding with filler As observed in soldering that as the use of a tiller rod reduces the formation of blow holes. So the current welding can be process should employ use of a filler rod of same material as filter mesh. Wire diameter of rod should be around 0.7-0.9 mm.
3. Change in parameters of current welding technique:On discussing with LASER welding vendor about the problem, we came to know that the thickness of filter strips we've been sending to vendor were less than 7mm_ When measured this dimension for 5 randomly selected strips, the average strip thickness came out to be 6. mm. Less than requirement of 7mm. This could also be a possible reason for formation of blow holes, as the strip won't curve into a complete circle leaving space in between two longitudinal ends.

For a proper weld, the ends should overlap each other which is possible for strip thickness > 7mm. Therefore, the liability of this issue can be traced back to both the ends — I) at the LASER vendor and 2) at our end. But first we asked the vendor to change the welding parameters (current and beam thickness etc_) and weld the same strips to manufacture 8 new filters of 2851_ for a new trial. These new filters with changes parameters were checked under microscope for holes. There were still blow holes, but their number had reduced to great extent. The vendor was asked to note the new parameters and to manufacture the upcoming lot using these parameters only unless told otherwise. Then an 8 tube heat exchanger heat exchanger was manufactured using these new filters, the setup was arranged to fill material in this heat exchanger using vacuum-vibration integrated system.

The observation of this trial are as follows: Test for 2901, with new filters:
Date — 25/11/17
Fin length = 290 mm Total length = 310 mm Empty weight = 392.5 g
Total weight of powder taken for filling = 250 g
Ideal capacity of heat exchanger = 22S g (as per drawings) Average practical amount filled in heat exchanger = 205 g Time duration of trial = 90 minutes
After 20 rnins,
The heat exchanger was undamped and its weight was measured to be 564,5 g Therefore, amount of powder in = 172.0 g
Then again it was clamped and left to vibrate so that the powder settles and room is
made for addition of more material.
During this the discharge was monitored regularly, there were losses but very less compared to earlier, as the turbidity of discharge was reduced and it was only lightly polluted.
Material was poured in as soon as some space was made in tubes of heat exchanger. After 70 mins from start time
The heat exchanger was undamped and its weight was measured to be 595 g Therefore, amount of powder in = 202.5 g
Then again it was clamped a,nd left to vibrate so that the powder settles and mom is made for addition of more material.
After 90 mins from start time,

The final weight of heat exchanger was measured to be 597 g
Therefore, amount of powder in = 204.5 g
Material left in beaker accounted for = 4 g
Loss of material = initial amount — amount of powder in FIX — final amount left
=. 250 — 2045 — 4
Loss of material =, 413 g

Table 3.2 Experiment 2 - (Tube + Filter + Copper foam)	
Length of tube	313 mm
Effective length	310 mm
I.D of tube	5.6 mm
Filter Length	285 mm
Volume of copper foam	77.723
Number of copper foam	5
Volume of tube	7635.327
Total Volume of Copper foam	388.615
Volume of Filter	910.859
Available voume	6335.853
Initial weight	31.4 g
Final weight	58.4 g
Weight of material	27.0 g
Tap density	4.261 g/cc

2.11 Calculation for tap density:

Table 2. COPPER FOAM DATA				
Sr. no.	O.D	Thickness	Length	Avg. Weight
1	5.12	1.03	59.48	0.6964
2	5.1	0.96	58.66	
3	5.11	1.06	59.54	
4	5.13	0.88	58.78	
5	5.1	1.03	58.66	
Avg.	5.112	0.992	59.024	
Density of Copper		8.96 g/cc		
Average volume of Copper foam		77.723		

Table 3. Experimental Data	
Table 3.1 Experiment 1 - With empty tube	
Length of tube	599 mm
I.D of tube	5.45 mm
Volume of tube	13973.645
Weight of tube after connector brazing	24.0 g
Weight after material filling	88.9 g
Weight of material filled	64.9 g
Tap density (max.)	4.644 g/cc

Using the above data, calculation for 2901, heat exchanger can be done. Effective length of tube = 300 mm

Total volume of tube =4
= 3,14 x 5,62 x 300 x 0.25
= 7389.026 nim3
Available volume of tube = total volume - foam volume - filter volume
= 7389.026 - 388.615 - 910.859
= 6089552 mm3
Total available volume = no. of tubes x available volume in one tube
= 8 x 6089.522
= 48716.176 mm3= 48.7162 cm3
Tap density = 4.2 g/cc

Therefore, we can say that packing efficiency of the powder compact is also high 52.17%, which is only 5.53% lesser than maximum packing efficiency,

Result

1. The solder method improves the joining process to great extent and eliminates blow holes but for its application to manufacturing, demands change in the other standard fabrication process in the chain. Solder method has the potential to reduce the cost to one-fourth of that required in LASER welding.
2. The best solution For now, is to change the parameters of LASER welding and control the width of the filter strips.
3. As it can be seen in the latest trial, that change in few parameters of welding the number holes reduced drastically and the loss of material for dimensionally similar heat exchangers reduced form 81.5 g to 415 g which is nearly the half.
4. Also with these new filters, it was possible to achieve the practical average weight target of 205 g for 290L heat exchangers. No need of reducing vibration or pump was to be reduced.
5. Also the time required to fill a heat exchanger of 290L was reduced from 5 hour's average (only with vibration) to 1,5 hrs (with vacuum-vibration integrated system) for filling 205 g of material in the heat exchanger.
6. The packing efficiency is also powder compact is also high.

$$\% \text{ Max. packing efficiency} = \frac{\text{max.Tap density}}{density\ of\ material}$$

$$= \frac{\text{max.Tap density}}{density\ of\ material}$$

$$= 57.7\%$$

Large scale execution

- As mentioned above, the system uses a vacuum or suction method for material filling which is installed from the bottom side. This leaves room for arrangement of automatic hoppers which can weigh and discharge proper amount powder material in the hopper which is goes into the tubes of the heat exchangers. Depending upon the size of the heat exchangers the amount of powder can be varied by using proper programming and sensors and actuators for data collection and controlling the parameter.
- Also for large scale execution, for example, filling 4 units at a time the pump
- capacity has to be increased, the flow rate has to be increased, to keep the Reynold's number and Euler's number in the optimum range for operation.
- The vibrating machine used in the trail was a small one, Therefore, when working with four units at a time, a bigger size and high maximum frequency vibration machine will be needed.
- Also dimensionally similar heat exchangers have to clamped simultaneously toavoid any mismatch in flow rate through different tubes.

Conclusion

The study of 200 x 1400 per inch. Dutch twilled weave was carried out and important parameters like porosity, characteristics diameter etc. were calculated and verified, The flow for fluid through dutch twill weave was studied, factors influencing the pressure drop like flow rate and dimensionless quantities Reynold's number and Euler's number were studied.Their effect on pressure loss and friction factor was measured.
Using the above study an optimum range of Reynold's number was set for filling operation. The study also helped to choose a suitable pump for operation.

Solution to the problem faced due to leakage were suggested and tested successfully. Loss of material was reduced by improvement in welding parameters. Also the main objective of the project of time reduction was also achieved. The average time of 5 hour's was reduced to 1,5 hour's (90 minutes), which is only the 30% percent of earlier time i.e. the process time reduced by 70%. Also the system was developed keeping in mind the scope for future development, therefore it has room for further improvement and automation.

References

1. R. Velraj, R. V. Seeniraj, B. Hafner, C. Faber, and K. Schwarzer, “Heat transfer enhancement in a latent heat storage system,” Solar Energy, vol. 65, no. 3, pp. 171–180, 1999.View at: Publisher Site | Google Scholar
2. R. Velraj, R. V. Seeniraj, B. Hafner, C. Faber, and K. Schwarzer, “Experimental analysis and numerical modelling of inward solidification on a

finned vertical tube for a latent heat storage unit," Solar Energy, vol. 60, no. 5, pp. 281–290, 1997.View at: Publisher Site | Google Scholar

3. K. A. R. Ismail, C. L. F. Alves, and M. S. Modesto, "Numerical and experimental study on the solidification of PCM around a vertical axially finned isothermal cylinder," Applied Thermal Engineering, vol. 21, no. 1, pp. 53–77, 2001.View at: Publisher Site | Google Scholar
4. E. M. Sparrow, E. D. Larson, and J. W. Ramsey, "Freezing on a finned tube for either conduction-controlled or natural-convection-controlled heat transfer," International Journal of Heat and Mass Transfer, vol. 24, no. 2, pp. 273–284, 1981.View at: Publisher Site | Google Scholar
5. U. Stritih, "An experimental study of enhanced heat transfer in rectangular PCM thermal storage," International Journal of Heat and Mass Transfer, vol. 47, no. 12-13, pp. 2841–2847, 2004.View at: Publisher Site | Google Scholar
6. A. Trp, "An experimental and numerical investigation of heat transfer during technical grade paraffin melting and solidification in a shell-and-tube latent thermal energy storage unit," Solar Energy, vol. 79, no. 6, pp. 648–660, 2005.View at: Publisher Site | Google Scholar
7. A. Trp, K. Lenic, and B. Frankovic, "Analysis of the influence of operating conditions and geometric parameters on heat transfer in water-paraffin shell-and-tube latent thermal energy storage unit," Applied Thermal Engineering, vol. 26, no. 16, pp. 1830–1839, 2006.View at: Publisher Site | Google Scholar
8. A. Erek, Z. Ilken, and M. A. Acar, "Experimental and numerical investigation of thermal energy storage with a finned tube," International Journal of Energy Research, vol. 29, no. 4, pp. 283–301, 2005.View at: Publisher Site | Google Scholar
9. A. Erek and M. A. Ezan, "Experimental and numerical study on charging processes of an ice-on-coil thermal energy storage system," International Journal of Energy Research, vol. 31, no. 2, pp. 158–176, 2007.View at: Publisher Site | Google Scholar
10. L. C. Chow, J. K. Zhong, and J. E. Beam, "Thermal conductivity enhancement for phase change storage media," International Communications in Heat and Mass Transfer, vol. 23, no. 1, pp. 91–100, 1996.View at: Publisher Site | Google Scholar
11. J. P. Bédécarrats, F. Strub, B. Falcon, and J. P. Dumas, "Phase-change thermal energy storage using spherical capsules: performance of a test plant," International Journal of Refrigeration, vol. 19, no. 3, pp. 187–196, 1996.View at: Publisher Site | Google Scholar
12. J. C. Mulligan, D. P. Colvin, and Y. G. Bryant, "Microencapsulated phase-change material suspensions for heat transfer in spacecraft thermal systems," Journal of Spacecraft and Rockets, vol. 33, no. 2, pp. 278–284, 1996.View at: Publisher Site | Google Scholar
13. J. M. Marín, B. Zalba, L. F. Cabeza, and H. Mehling, "Improvement of a thermal energy storage using plates with paraffin-graphite composite," International Journal of Heat and Mass Transfer, vol. 48, no. 12, pp. 2561–2570, 2005.View at: Publisher Site | Google Scholar
14. L. F. Cabeza, H. Mehling, S. Hiebler, and F. Ziegler, "Heat transfer enhancement in water when used as PCM in thermal energy storage," Applied Thermal Engineering, vol. 22, no. 10, pp. 1141–1151, 2002.View at: Publisher Site | Google Scholar

15. H. Mehling, S. Hiebler, and F. Ziegler, “Latent heat storage using a PCM—graphite composite material,” in Proceedings of the 8th International Conference on Thermal Energy Storage (Terrastock '00), pp. 375–80, Stuttgart, Germany, 2000.View at: Google Scholar
16. X. Py, R. Olives, and S. Mauran, “Paraffin/porous-graphite-matrix composite as a high and constant power thermal storage material,” International Journal of Heat and Mass Transfer, vol. 44, no. 14, pp. 2727–2737, 2001.View at: Publisher Site | Google Scholar
17. M. Xiao, B. Feng, and K. Gong, “Thermal performance of a high conductive shape-stabilized thermal storage material,” Solar Energy Materials & Solar Cells, vol. 69, no. 3, pp. 293–296, 2001.View at: Publisher Site | Google Scholar
18. W. Wang, X. Yang, Y. Fang, and J. Ding, “Preparation and performance of form-stable polyethylene glycol/silicon dioxide composites as solid-liquid phase change materials,” Applied Energy, vol. 86, no. 2, pp. 170–174, 2009.View at: Publisher Site | Google Scholar
19. W. Wang, X. Yang, Y. Fang, J. Ding, and J. Yan, “Enhanced thermal conductivity and thermal performance of form-stable composite phase change materials by using β-Aluminum nitride,” Applied Energy, vol. 86, no. 7-8, pp. 1196–1200, 2009.View at: Publisher Site | Google Scholar
20. M. J. Hosseini, A. A. Ranjbar, K. Sedighi, and M. Rahimi, “A combined experimental and computational study on the melting behavior of a medium temperature phase change storage material inside shell and tube heat exchanger,” International Communications in Heat and Mass Transfer, vol. 39, no. 9, pp. 1416–1424, 2012.View at: Publisher Site | Google Scholar