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# STUDY ANALYSIS OF EFFECT OF DIFFERENT PARAMETERS ON DESIGN ASPECTS OF CRYOGENIC WIRE MESH HEAT EXCHANGER

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## ABSTRACT

In this paper the study is emphasized on the cryogenic heat exchanger of helium liquefaction system operating with three-stage Gifford McMahon refrigerator, as cryogenic heat exchanger is the most critical component which can make the any liquefaction system efficient or inefficient. Here, wire mesh packing matrix is used in recuperative type heat exchanger, as it has quite high effectiveness comparable to regenerators. With the help of computerized design, results are produced for (1) various wire mesh diameter of packing matrix and (2) various mesh size of packing matrix and their effect on design criteria of cryogenic heat exchanger is studiedResults are plotted on the graphs. By analyzing those graphs some conclusions are made:

1. For economical and satisfactory performance of wire mesh heat exchanger; porosity should be as high as possible.
2. For constant mesh size, as the wire diameter increases, length of H.E. decreases and pressure drop increases.

**Keywords:** Heat Exchangers, Joule-Thompson expansion valve, Inner tube diameter, Outer tube diameter

## 1. INTRODUCTION

For the study, the wire mesh heat exchanger used for helium liquefaction system operating with three stage Gifford – McMahon refrigerator is selected, which is shown in fig. (1).

Referring to figure 1, the helium gas is initially compressed to 20 atm. and 300 K. It passes through the H.E.-1 (design analysis of which is to be carried out) giving its heat to the cold gas stream, which is coming in at 80 K in counter flow direction from liquid receiver. By using Gifford Mc-Mahon refrigerator, cooling of helium gas is achieved below its inversion temperature (i.e. 45 K) [2] in three stages. So that it can be liquefied after J-T expansion. After the first stage of refrigeration, hot stream is allowed to flow through H.E. – 2. In H.E. – 2, it is cooled by a cold stream, entering at temperature around 35 K in counter flow direction. Then it enters the second stage of refrigeration, the hot incoming stream passes through the H.E. – 3, where it is further cooled by cold stream at 14K and then enters into the third stage of cooling. After that, incoming stream passes through the H.E. – 4 and is cooled by return cold gas stream at lower than 4.2.K coming from liquid receiver directly.

Finally the incoming gas is allowed to expand through the J – T expansion valve in which isenthalpic expansion takes place. As the temperature of incoming helium gas

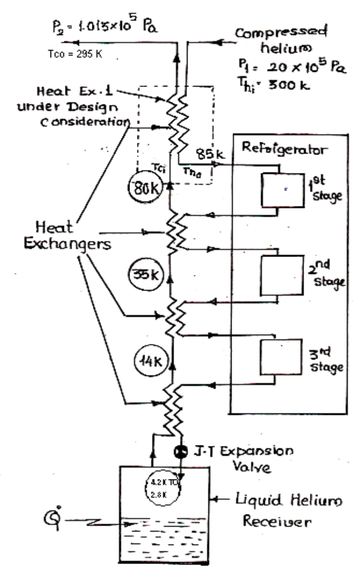



Fig 1: A Helium Liquefaction System Containing Three Stage Gifford McMahon Cooling.

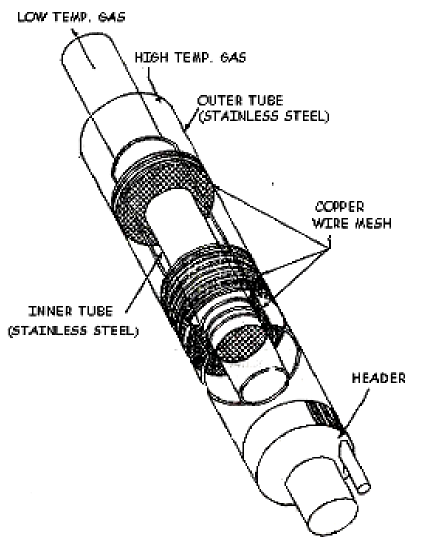



Fig 2: Concentric Tube Wire Mesh Heat Exchanger

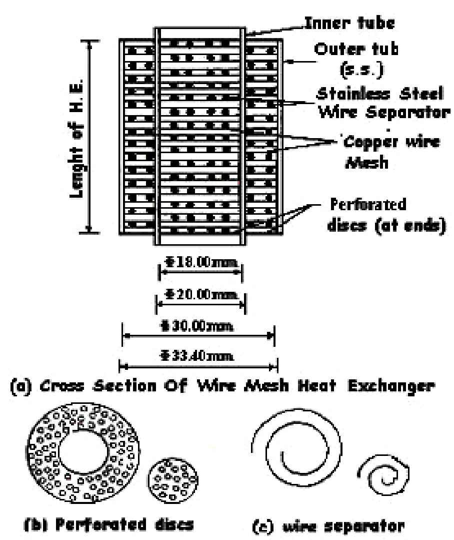



Fig 3: Details of Wire Mesh Heat Exchanger

entering into the J – T expansion is below the inversion temperature (45 K), it will be liquefied and liquid yield will be collected in liquid receiver.

Among the four heat exchangers of liquefaction system, the effect of variation of different parameters is studied on design aspects of first heat exchanger, H.E.-1.

Wire mesh heat exchanger is a matrix type heat exchanger in which wire mesh screens are used as a packing matrix. These exchangers are used in cryogenic systems because of its quite high effectiveness comparable to regenerators of cryo – coolers.

From some experiments and basic calculations it is concluded that "where volume or surface area with respect to NTU is of prime importance, the wire mesh screens have been proved as an excellent medium of heat transfer for a high effectiveness heat exchangers at low temperatures".

In helium liquefaction system, heat exchanger effectiveness is the most important parameter for liquefaction i.e. cooling effect and heat transfer area is of prime importance. Therefore, I have selected wire mesh as a packing material and NTU/volume approach [2] for configuration of heat exchanger.

With the above consideration in mind, an exchanger configuration selected is as shown in fig. 2 and fig. 3.

## 2. DESIGN PARAMETERS

For liquefaction system shown in figure – 1 following parameters are taken constant.

Let,

Inlet pressure of compressed gas, $P_1$ = $20.26 \times 10^5$ Pa

Inlet temperature of hoot stream, $T_h$ = 300 K

Outlet temperature of hot stream, $T_{ho}$ = 85 K

Inlet temperature of cold stream, $T_{ci}$ = 80 K

Outlet temperature of cold stream, $T_{co}$ = 295 K

Now, effectiveness of Heat exchanger,

$$\varepsilon = \frac{\mathbf{m_h \cdot Cp_h \ (T_{hi} - T_{ho})}}{\mathbf{(m_* Cp)_{min} \ (T_{hi} - T_{ci})}} \quad \text{--------- (1)}$$

$$\varepsilon = \frac{\mathbf{m_c * Cp_c \ (T_{co} - T_{ci})}}{\mathbf{(m_* Cp)_{min} \ (T_{hi} - T_{ci})}} \quad \text{---------- (2)}$$

For our heat exchanger, no liquid yield is withdrawn from the liquid receiver, i.e. the liquefier is used for refrigeration purpose and therefore,

$m_h = m_c$ and $C_{ph} = C_{pc}$ and so,

$$\varepsilon = \frac{\mathbf{(T_{hi} - T_{ho})}}{\mathbf{(T_{hi} - T_{ci})}} = \frac{\mathbf{(T_{co} - T_{ci})}}{\mathbf{(T_{hi} - T_{ci})}} \quad \text{-------- (3)}$$

$$= \mathbf{0.977}$$

Hence, the effectiveness of H.E. – 1 is 97.7% which is quite acceptable.

Now it is decided that hot stream will flow in the annulus (passage between outside of inner tube and inside of outer tube) and the cold stream will flow in the inner tube.

## 3. SPECIFICATION TAKEN FOR THERMAL DESIGN CONSIDERATION

- *Type of heat exchangers*:
  - i. recuperative heat exchangers
  - ii. Matrix type heat exchangers with wire mesh packing.
    - Gas to gas type (single phase )

- Tubular type.
- *Flow arrangement* :
  - counter flow
- *Heat transfer mode :*
  - combined conduction and convection
- *Packing material :*
  - copper wire mesh screens
  - Stainless steel spiral wire separator.
- *Working fluid :*
  - helium
- *Inlet pressure P1 :*
  - $20.265\text{X}10^5$ Pa
- *Mass flow rate m :*
  2.00 Kg/hr

## 4. RESULTS AND ANALYSIS

For economic consideration, pressure drop and size (length) of H.E. are important parameters. Pressure drop on cold side (low pressure circuit) is especially significant [4] and may be cost in terms of compression power requirement. So, during analysis, attention is given to pressure drop on cold side, ΔPc particularly.

With the help of computerized thermal design, results were obtained for the following cases:

a) Effect of variation in wire diameter of mesh.
b) Effect of variation in mesh size of packing matrix.

For the analysis purpose, for different mesh size like 30, 40, 50 and 60 important parameters like pressure drop & length of H.E. is calculated.

In addition to this, some other parameters like porosity p, hydraulic diameter $D_h$, overall fin effectiveness $\eta_{oh}$ (hot side) and $\eta_{oc}$ (cold side) for various wire diameter and for various mesh sizes are also obtained.

From those results, graphs are drawn for different parameter as shown in fig. 4 to 7.

Fig. 4, 5, 6, and 7 represents the graphs drawn for mesh sizes 30, 40, 50, and 60 respectively.

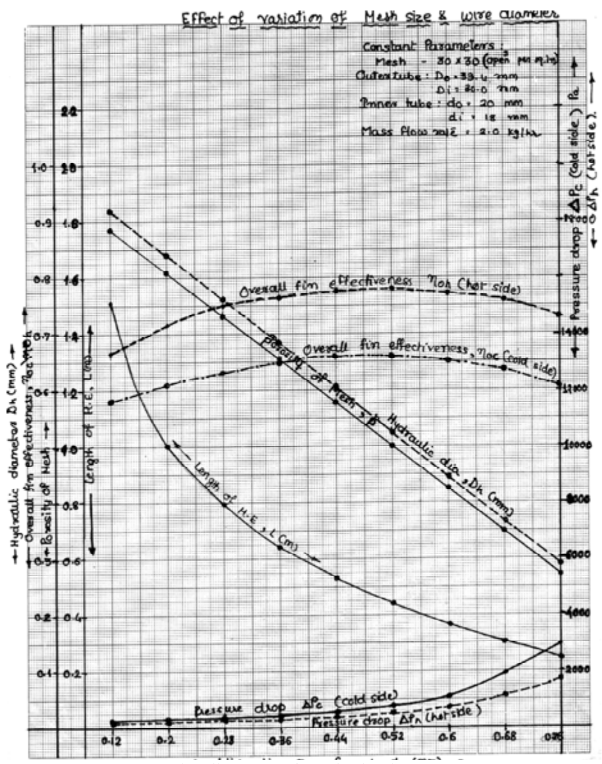
Each of these graph represents curves for pressure drop within H.E., length of H.E. L, porosity of mesh p, hydraulic diameter $D_{h.}$, overall fin effectiveness $\eta_{oh}$ and $\eta_{oc}$ for various wire diameter and for different mesh sizes.

- <u>From the graphs following observations are noted:</u>

a) For particular mesh size as the wire diameter dw increases, porosity of mesh decreases linearly. Also length of heat exchanger decreases and decrease is non-linear.

b) With the decrease in porosity of a particular mesh size, (i.e. increase of wire diameter for particular mesh) pressure drop first increases very slowly and then drastically.

c) Small size and low pressure drop in H.E. result in high refrigeration capacity and economical performance of the system. For small size and low hydraulic diameter, high porosity and low hydraulic diameter is recommended by Lins and Elkans [1].



Wire diameter of mesh, dw in mm

Fig 4: Graph for mesh size – 30

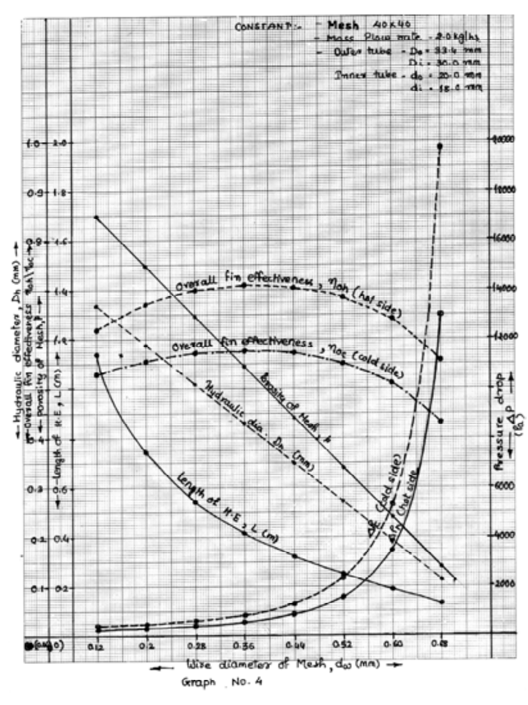



Fig 5: Graph for mesh size – 40

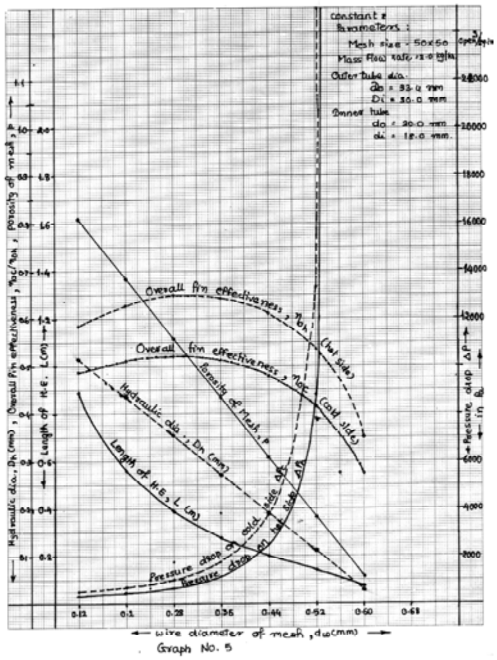



Fig 6: Graph for mesh size – 50

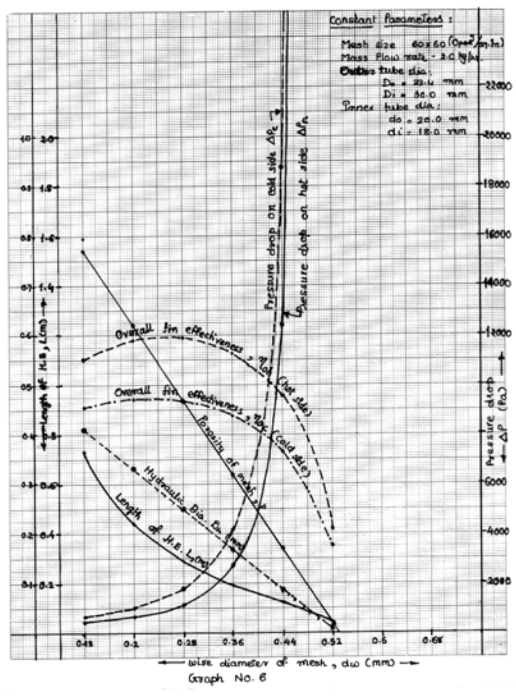



Fig 7: Graph for mesh size – 60

This statement can also be explained through following example.

For particular wire diameter, dw = 0.12 mm constant, when mesh size increases from 30 to 40, the decrease in porosity is 4.2 %and decrease in hydraulic diameter is 27.5%. Due to this large decrease in hydraulic diameter, length of H.E. achieved is 24.7% (37 cm approximately) and pressure drop increase is very small. This shows that keeping wire diameter constant, if small variation in mesh size is done, porosity will not be much affected as compared to hydraulic diameter.

This result is observed due to the effect of hydraulic diameter only and this parameter involves wire diameter as well as porosity.

- From the above discussion, it is concluded that for small size and low pressure drop of heat exchanger, one should select the mesh and wire size such that it can give high porosity and low hydraulic diameter.
- As the maximum allowable pressure drop limit is taken as 10% (i.e. 10000 Pa) on cold side, it is observed from the graph that for each of mesh size, there is a certain value of porosity and wire diameter after which the value of pressure drop becomes higher than 10%. At the value of pressure drop ΔPc = 10000 Pa, obtained length will be minimum that can be achieved for that related mesh size.
- A table – 1 given below presents the values of wire diameter, porosity and length of H.E. achieved for different mesh sizes for 10% pressure drop.

Table 1:

| Sr. No. | Mesh size | Length of H.E., L (m) | Wire dia. dw (mm) | Porosity, p | Pressure drop on cold side ΔPc (Pa) |
|---|---|---|---|---|---|
| 1 | 40X40 | 0.15 | 0.65 | 0.175 | 10000 |
| 2 | 50X50 | 0.14 | 0.516 | 0.185 | ” |
| 3 | 60X60 | 0.135 | 0.423 | 0.205 | ” |

From above table and graphs, it is seen that for a particular value of pressure drop, minimum length of heat exchanger reduces for higher mesh sizes. Also the range of wire diameter selection with the increasing mesh size becomes narrower in which heat exchanger can give satisfactory performance.

Also it is seen that as we go for higher finer mesh and lower wire diameter, we can get smaller length of H.E. with high porosity.

Here, among all mesh sizes taken, 60 is the best mesh size for constant mass flow rate and tube diameters to give desired performance.

### 4.1 Overall Fin Effectiveness ηoh (Hot Side) and ηoc (Cold Side)

From the graphs, also the effect of variation in wire diameter, ‘dw’ on overall fin effectiveness is observed. The curves of fin effectiveness on hot side $\eta_{oh}$ and on cold side $\eta_{oc}$ both are plotted in the graph of respective mesh size. For a particular mesh size, as the wire diameter increases, overall fin effectiveness both $\eta_{oh}$ and $\eta_{oc}$ initially increase and after reaching some maximum value they decrease. For each mesh size, there is a maximum value of fin effectiveness after which it decreases as the wire size increases further.

Table-2, given below shows the values of optimum fin effectiveness with wire diameter at which it occurs for different mesh sizes.

It is seen from the table that as the mesh size increases, maximum fin effectiveness $\eta_{oc}$ decreases and moves towards lower wire diameter.

Table 2:

| Sr. No. | Mesh size | Optimum fin effectiveness $\eta_{oh}$ | Optimum fin effectiveness $\eta_{oc}$ | Porosity p | Wire diameter dw, in mm for Optimum $\eta_{oh}$ or $\eta_{oc}$ |
|---|---|---|---|---|---|
| 1 | 30X30 | 0.7785 | 0.6578 | 0.520 | 0.5 |
| 2 | 40X40 | 0.7081 | 0.5764 | 0.545 | 0.36 |
| 3 | 50X50 | 0.6478 | 0.5192 | 0.560 | 0.28 |
| 4 | 60X60 | 0.5956 | 0.4750 | 0.545 | 0.24 |

It also appears from the table – 2 that for each mesh size taken, for maximum effectiveness, porosity remains approximately same & nearly 0.55. This shows that selection of mesh size & wire diameter should be such that the porosity remains greater than 0.5 for economic performance.

## 5. CONCLUSIONS

From above discussion and analysis, some conclusions are made for wire mesh heat exchanger and presented here as follows:

- Keeping the mass flow rate and tube diameters constant, when wire diameter of mesh is changed for different mesh sizes, it is observed that,

1. For economic and satisfactory performance of wire mesh heat exchanger, porosity should be as high as possible and hydraulic diameter should be as low as possible.
2. For higher porosity and lower hydraulic diameter, one should select finer mesh size and smaller wire diameter.
3. For constant mesh size as the wire diameter increases length of an exchanger decreases continuously but pressure drop increases very slowly at small wire diameter and rapidly for higher wire diameter.
4. With the increase in mesh size, pressure drop curves become steeper quickly as the wire diameter increases.
5. As the mesh size increases, the wire diameter which can give a minimum length at 10% pressure drop decreases. This shows that as we go for high mesh size, the range of wire diameter selection becomes narrower.
6. As the mesh size increases, the value of porosity when becomes lower than 0.3 approximately, pressure drop starts to rise drastically.

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## 7. NOMENCLATURE

| Symbol | Meaning | unit |
|---|---|---|
| Dh | hydraulic diameter | mm |
| **€** | Effectiveness of Heat Exchangers | Dimensionless |
| NTU | Number of Heat Transfer Unit | Dimensionless |
| $\Delta P_c$ | Pressure Drop on Cold Side of Heat Exchanger in Pa | **Pa** |
| $\Delta P_h$ | Pressure Drop on Hot Side | Pa |
| T | Temperature | K |
| $C_p$ | Specific Heat at Constant Pressure | kJ/kg.k |
| $d_w$ | Wire Diameter | mm |
| $\dot{m}$ | Mass flow Rate | kg/hr. |
| p | porosity | Dimensionless |
| $\eta_o$ | Overall Fin Effectiveness Of Wire Mesh Matrix | Dimensionless |