PRESSURE DROP CONSTRAINTS IN SLUDGE DOUBLE-PIPE HEAT EXCHANGER DESIGN

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ABSTRACT:

Recent years have witnessed a rapid development in the understanding of heat exchangers design. Those developments have justified the use of a global minimum allowable temperature approach under which the heat transfer area of the heat exchanger is minimized. Values of the allowable pressure drops of streams of exchangers are specified to calculate film heat-transfer coefficients of streams and heat-transfer area. By coupling this with the understanding of what dictates the energy consumption, it is possible to determine the trade-off between the heat exchanger capital cost and energy cost prior to design work. Pressure drop is an important issue in the design of a heat exchanger. Pumps and (or) compressors must be installed to overcome pressure losses when streams flow through heat exchangers. The total cost for a system of pumps and compressors consists of the purchase cost of equipments and the electricity cost to run these equipments. This cost could occupy a significant part of the overall cost for a heat exchanger. Therefore, the pressure drop aspect should be considered together with the costs of heat exchanger area.

This paper demonstrates how pressure drop is considered in the context of a sludge double-pipe heat exchanger design. A relationship between heat transfer coefficient and heat exchanger pressure drop was determined and its capital cost implications were assessed.

Keywords: **Heat Exchanger design, Pressure drop constraints, heat exchanger costs.**

محددات فرق الضغط في تصميم مبادلات الحمأة الحرارية نوع الأنبوب المزدوج

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الخلاصة:

شهدت السنوات الأخيرة تطور سريع في تصاميم المبادلات الحرارية. تميزت معظمها باستخدامها مفهوم أدنى فرق في درجات الحرارة للحصول على اقل مساحة انتقال حراري. وطبقا لتلك الطريقة يتم تحديد قيم فرق الضغط المقبول لحساب معامل انتقال الحرارة ومساحة الانتقال الحراري من خلال ربط هذه الطريقة مع أسلوب حساب استهلاك الطاقة في المبادلات الحرارية ويمكن إجراء المفاضلة بين كلفة المبادل الحراري وكلفة الطاقة اللازمة لتشغيله قبل مرحلة التصاميم التفصيلية. وحيث أن فرق الضغط في المبادلات الحرارية ينعكس على كلف ملموسة في المبادلات الحرارية تتمثل بكلف المضخات

والدافعات وكلف الطاقة اللازمة لتشغيل تلك المضخات والدافعات، فان دراسة تلك الكلف نسبة إلى كلفة المبادل الحراري تعتبر أمر ضروري ومهم.

يستعرض هذا البحث كيفية تضمين فرق الضغط في مبادلات الأنبوب المزدوج المستخدم لتسخين الحمأة في أحواض الهضم ضمن مرحلة التصميم من خلال إيجاد علاقة بين معامل انتقال الحرارة وفرق الضغط وتقييم انعكاس ذلك على الكلفة المثالية للمبادل الحراري.

INTRODUCTION:

The heat transfer rate of heat exchanger streams can be improved by generating turbulent flow (breaking the viscous and thermal boundary layers), but the pumping power may increase significantly and ultimately the pumping cost becomes high. Swirl flow devices for example, a twisted-tape [1-3], a wire-coil inserts, and tangential injection devices [4] impart a tangential velocity component to the fluid that increases the turbulence of the flow and consequently the heat-transfer coefficient. Double pipe heat exchanger is considered to be most reliable among other types of heat exchangers. It usually requires minimal maintenance. It is widely used for critical heating or cooling of slurries and high viscosity liquids. Sludge is one example of those high viscosity liquids. Sludge heat exchangers are used to heat sludge at inlet of anaerobic digesters. These exchangers are clog free and easy to wash from sludge encrusting. This is the result of having pipe completely straight and easy to dismantle. Besides to the previous advantages (suitable for severe fouling conditions), double pipe heat exchangers suffer a shortcoming of small heat transfer area (up to 50 m^2) and high pressure. The great attempt on utilizing different methods is to increase the heat transfer rate through compulsory force convection. In general, enhancing the heat transfer can be divided into two groups. One is the passive method, without stimulation by the external power such as a surface coating, rough surfaces, extended surfaces, swirl flow devices, the convoluted (twisted) tube, additives for liquid and gases. The other is the active method, which requires extra external power sources, for example, mechanical aids, surface-fluid vibration, injection and suction of the fluid, jet impingement, and use of electrostatic fields. Whitham [5] published his work on heat transfer enhancement by means of twisted-tape way back at the end of the nineteenth century. Koch [6] indicated that in turbulent flow, inserting of a twisted-tape increases the heat transfer, but the pressure drop also increases significantly. Kumar and Bharadwaj [7] obtained theoretically the heat transfer and pressure drop correlations using the Kreith and Sonju [8] solution for the velocity vector, which decays along the axis of the tube. Huang and Tsou [9] studied free swirl flow in a pipe. Aydin [10] investigated heat transfer and pressure drop in a concentric heat exchanger with turbulent decaying swirl flow. Liao and Xing [11] reported experimental data on the compound heat transfer enhancement technique and concluded that the enhancement of heat transfer in a tube with three dimensional internal extended surfaces by replacing continuous twisted-tape with almost segmented twisted-tape inserts results in a decrease in the friction factor but with a comparatively small decrease in the Stanton number. The Stanton number is defined as the ratio of heat transfer rate to the enthalpy difference and is a measure of the heat transfer coefficient. In the present study, pressure drop across sludge double pipe heat exchanger has been related to heat transfer coefficient, so its cost implications on capital cost are readily assessed. Therefore, it is convenient to consider the optimization of pressure drops instead of specifying fixed allowable pressure drops in the targeting stage. Both inner and outer pipes of the double pipe heat exchanger were assumed to follow the Dittus- Boelter equation.

MATHEMATICAL MODELING:

For allowable pressure drop to be considered in design optimization, a relationship between heat transfer coefficient and exchanger pressure drop was developed. The film heat transfer coefficient is given by the Dittus-Boelter equation:

$$Nu = 0.023\mathrm{Re}^{0.8}\,\mathrm{Pr}^{\frac{1}{3}} \quad \ldots\ldots (1)$$

Where:

Nu : Nusselt number.
Re : Reynolds number.
Pr : Prandtl number.

$$\frac{hd}{k} = 0.023\left(\frac{rud}{m}\right)^{0.8}\left(\frac{mCp}{k}\right)^{\frac{1}{3}} \quad \ldots\ldots (2)$$

Where:

h : Convective heat transfer coefficient.
d : Pipe diameter.
r : Fluid density.
m : Fluid viscosity.
Cp : Specific heat.
k : Conductive heat transfer coefficient.
u : Fluid flow velocity.

$$h = 0.023\left(\frac{rd}{m}\right)^{0.8}\left(\frac{mCp}{k}\right)^{\frac{1}{3}}\left(\frac{k}{d}\right)u^{0.8} \quad \ldots\ldots (3)$$

$$h = 0.023 r^{0.8}(Cp)^{\frac{1}{3}}\frac{1}{m^{0.47}d^{0.2}}k^{0.67}u^{0.8} \quad \ldots\ldots (4)$$

For average flow conditions of double pipe heat exchanger, thermal and flow properties can be assumed constants. Therefore:

$$h = c_1 u^{0.8} \quad \ldots\ldots (5)$$

Where:

$$c_1 = 0.023 r^{0.8}(Cp)^{\frac{1}{3}}\frac{1}{m^{0.47}d^{0.2}}k^{0.67}$$

Equation (5) clearly shows that heat transfer is directly proportional to fluid velocity raised to (0.8). C_1 is a constant and its value depends on thermal and flow properties.
The pressure drop through the pipe is given by the fanning equation:

$$\Delta p = 2f(\frac{l}{d})ru^2 \qquad (6)$$

Where:

Δp : Pressure drop.

f : Friction factor.

l : Pipe length.

The friction factor is given by the Blasius equation:

$$f = 0.046\,\mathrm{Re}^{-0.2} \qquad (7)$$

Substituting equation (7) into equation (6):

$$\Delta p = 0.092(\frac{rud}{m})^{-0.2}(\frac{l}{d})ru^2 \qquad (8)$$

$$\Delta p = 0.092\,r^{0.8}d^{-1.2}m^{0.2}lu^{1.8} \qquad (9)$$

Again for average flow conditions of double pipe heat exchanger, thermal and flow properties can be assumed constants. Therefore:

$$\Delta p = c_2 lu^{1.8} \qquad (10)$$

Where:

$c_2 = 0.092\,r^{0.8}d^{-1.2}m^{0.2}$

Equation (10) clearly shows that pressure drop across double pipe heat exchanger are directly proportional to the velocity of fluid raised to (1.8). This relation reflects the higher sensitivity of pressure drop to fluid velocity than the heat transfer coefficient. The velocity of fluid can be expressed in terms of its volumetric flow rate:

$$u = \frac{4V}{pd^2} \qquad (11)$$

Where:

V : Volumetric flow rate.

The heat transfer area is given by:

$$A = pdl \qquad (12)$$

Where:

A : Heat transfer area.

Multiplying equation (11) by equation (12) yields:

$$Au = pdl\frac{4V}{pd^2} = \frac{4Vl}{d} \qquad (13)$$

$$l = \frac{d}{4V} Au \qquad \text{……} (14)$$

Substituting for l in equation (10) from equation (14):

$$\Delta p = c_2 \frac{d}{4V} Au^{2.8} \qquad \text{……} (15)$$

$$\Delta p = c_2 c_3 Au^{2.8} \qquad \text{……} (16)$$

Where:

$$c_3 = \frac{d}{4V}$$

Equations (16) and (5) show both pressure drop and heat transfer coefficient are functions of fluid flow velocity. From equation (5):

$$u^{0.8} = \frac{h}{c_1} \qquad \text{……} (17)$$

$$u = \frac{h^{1.25}}{c_1^{\ 1.25}} \qquad \text{……} (18)$$

Substituting for u from equation (18) into equation (16):

$$\Delta p = c_2 c_3 A(\frac{h^{1.25}}{c_1^{1.25}})^{2.8} = \frac{c_2 c_3}{c_1^{3.5}} Ah^{3.5} \qquad \text{……} (19)$$

Equation (19) shows that pressure drop across double pipe heat exchanger is related to heat transfer coefficient, so the augmentation of heat transfer coefficient through turbulence effect is directly reflected on pressure drop increase and consequently on energy consumption cost. As heat transfer coefficient increases, heat transfer area decreases and heat exchanger capital cost will be reduced accordingly. Both heat transfer area and energy consumption costs were estimated using Aspen correlations. Both costs were directly expressed in terms of heat transfer coefficient.

$$A = \frac{Q}{h \Delta T} \qquad \text{……} (20)$$

Where:

Q : Heat flow rate.

ΔT : Temperature difference.

$$\Delta p = c_4 \frac{Q}{\Delta T} h^{2.5} \qquad \text{……} (21)$$

Where:

$$c_4 = \frac{c_2 c_3}{c_1^{3.5}}$$

Equations (20) and (21) demonstrate the dependence of both heat transfer area and pressure drop on heat transfer coefficient. It is clearly seen that heat transfer area is inversely proportional to heat transfer coefficient while pressure drop is directly proportional to it. Therefore, the augmentation of heat transfer coefficient leads to considerable reduction in heat transfer area (heat exchanger capital cost), at the same time, this augmentation will cause an increase in pressure drop across the exchanger (energy consumption cost). These costs are expressed using Aspen Correlations [12] as:

$$CC = k_1 A^2 = k_1 \left(\frac{Q}{h\Delta T}\right)^2 \quad \text{......} \quad (22)$$

Where:

CC : Heat exchanger capital cost.

k_1 : Constant depends on material of construction.

$$EC = k_2 \Delta p = k_2 c_4 \frac{Q}{\Delta T} h^{2.5} = k_3 \frac{Q}{\Delta T} h^{2.5} \quad \text{......} \quad (23)$$

Where:

EC : Energy consumption cost of heat exchanger.

k_3 : Constant depends on fluid properties and geometry.

The main characteristics of a double pipe heat exchanger that used to heat a domestic wastewater sludge in an anaerobic digester is shown in **Table (1).**

The schematic representation of a sludge double pipe heat exchanger is shown in **Figure (1).** A countercurrent scheme is adopted for this investigation study as shown in the figure.

RESULTS AND DISCUSSIONS:

Table (2) presents the dependency of heat transfer coefficient (h), heat transfer area (A) and pressure drop across sludge pipe (ΔP) on the sludge stream velocity. It clearly shows that the increase of heat transfer coefficient is approximately linear function of the sludge stream velocity. This increase reflects oppositely on heat transfer area and directly on pressure drop. The heat transfer area decreases linearly while the pressure drop increases in exponent behavior.

Table (3) shows the capital cost (CC) of the double pipe heat exchanger as a function of the heat transfer area. The cost shown in the table represents the purchase cost of the heat exchanger only.

Table (4) indicates the yearly pumping energy consumption cost (EC) of the sludge stream as a function of pressure drop. The table shows the steep increase in energy consumption cost as the pressure drop increases.

Figures (2-7) schematically demonstrate the behavior of the main parameters of heat exchanger (heat transfer coefficient, heat transfer area, pressure drop, capital cost, and pump energy cost) in relation to sludge stream velocity. **Figure (1)** reflect the linearity of the relationship between the heat transfer coefficient and the sludge stream velocity.

The little increase in heat transfer coefficient is reflected on heat transfer area as shown in **Figure (2)** and on pressure drop in **Figure (3). Figures (4-5)** explain how capital cost drops and energy cost escalates as velocity increases. The total cost of the heat exchanger is shown in **Figure (6).** It can be seen that the optimum cost happens to be low velocities rather than high velocities.

CONCLUSIONS:

Both heat transfer area and pressure drop across double pipe heat exchanger are related to sludge stream velocity to specify the economic stream pressure drop ahead of design. It was concluded that the reduction in heat transfer area due to velocity augmentation not necessarily determines the minimum cost of double pipe heat exchanger. It was seen that stream pressure drop dominates the cost of the heat exchanger beyond threshold stream velocity. This leads to the use of the extended heat transfer area rather than the approach of inducing turbulence to augment the heat transfer coefficient when exceed the threshold stream velocity.

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Table (1) Main characteristics of a sludge double pipe heat exchanger

Sludge inlet temperature	31 C^0
Sludge outlet temperature	39 C^0
Sludge flow rate	94 m^3/h
Heat transfer rate	1129 KW
Heating fluid inlet temperature	65 C^0
Heating fluid outlet temperature	50 C^0
Heating fluid flow rate	50 m^3/h

Table (2) relationship between sludge stream velocity and heat transfer coefficient, heat transfer area and pressure drop.

u (m/s)	h (w/m^2 C)	A (m^2)	ΔP (N/m^2)
0.5	1084.7	103	1398.3
1.0	1888.7	59.3	5608.4
1.5	2612.4	42.8	12615.2
2.0	3288.4	34	22393.8

Table (3) Capital cost of heat exchanger as a function of heat transfer area.

A (m^2)	CC ($)
103	232337
59.3	77011
42.8	40117
34	25316

Table (4) Sludge stream pumping cost as a function of pressure drop.

ΔP (N/m^2)	EC ($)
1398.3	75368.4
5608.4	302292.7
12615.2	679948.5
22393.8	1207025.8

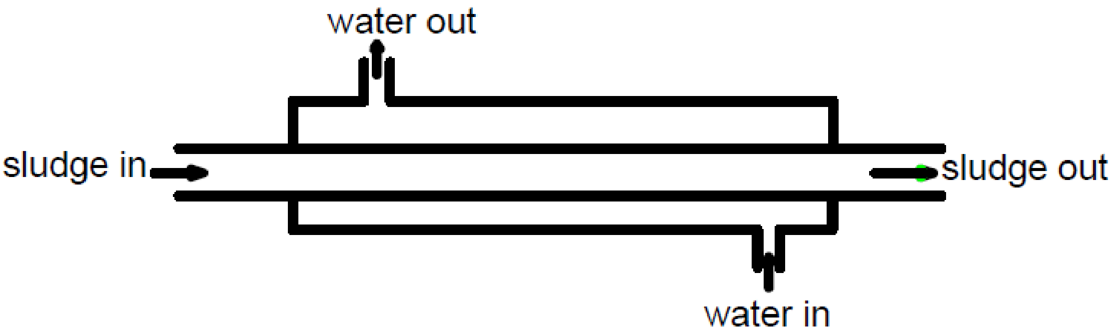

Figure (1), Schematic representation of a sludge double pipe heat exchanger

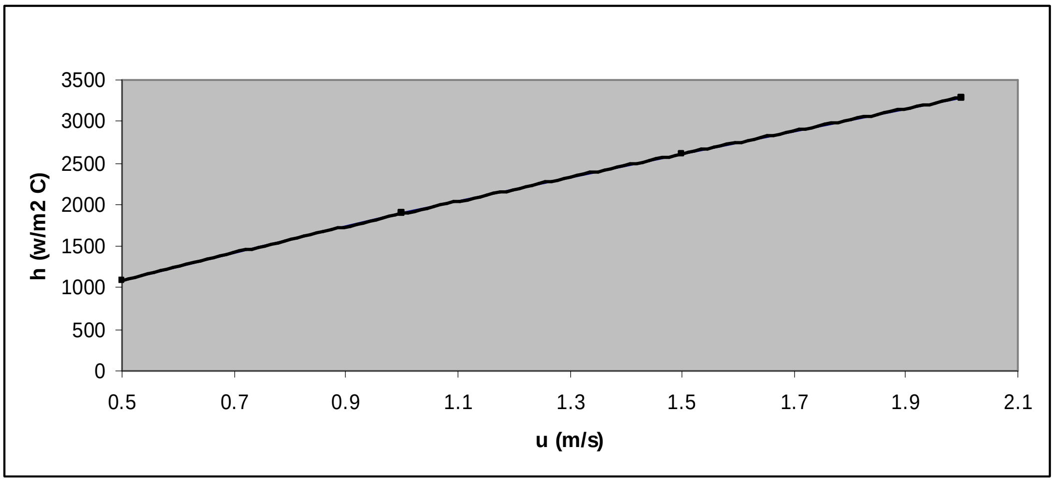

Figure (2) Relationship between heat transfer coefficient and sludge stream velocity.

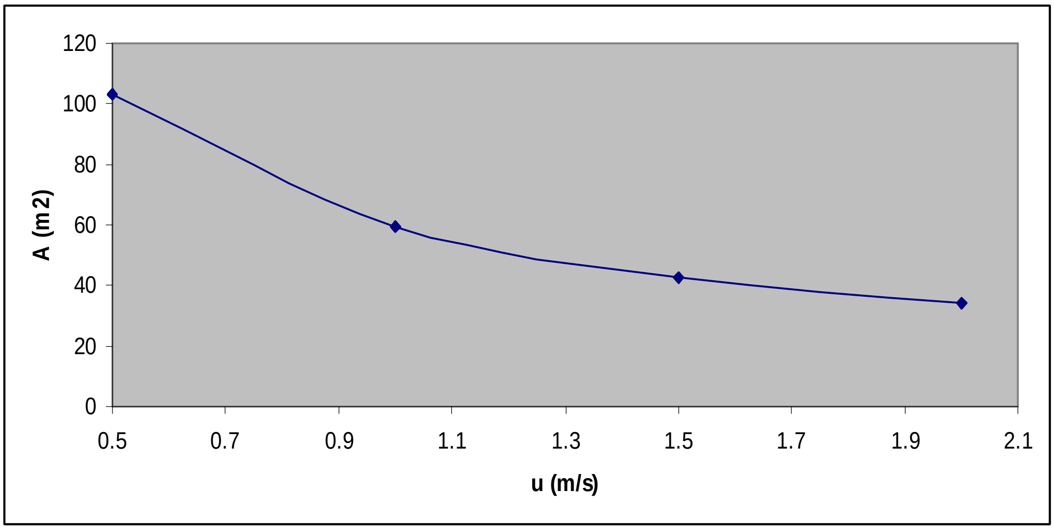

Figure (3) Relationship between heat transfer area and sludge stream velocity.

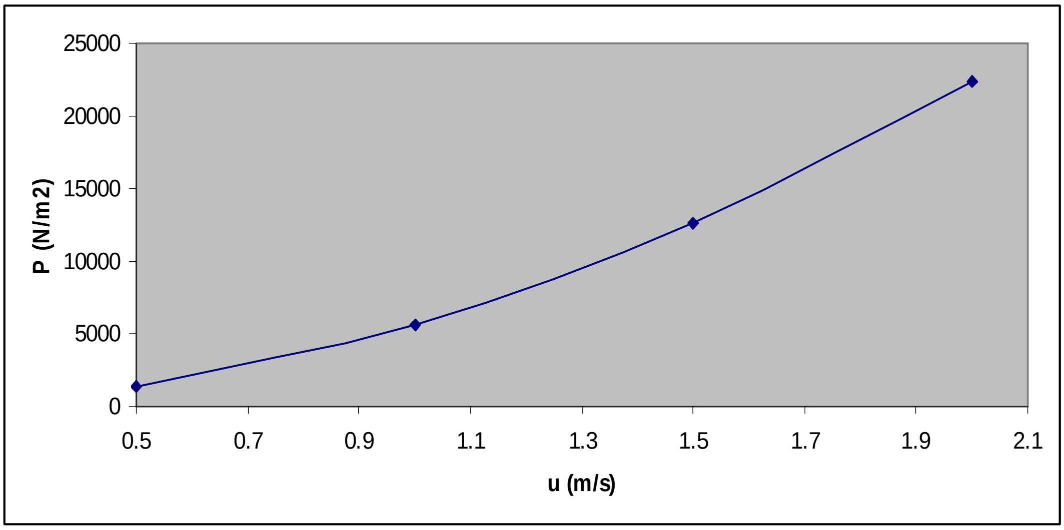

Figure (4) Relationship between pressure drop and sludge stream velocity.

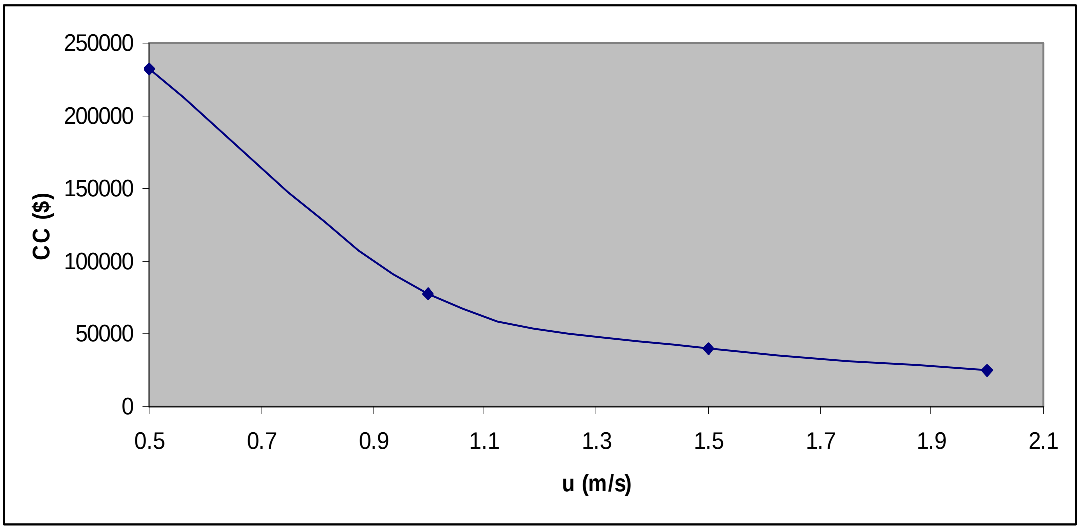

Figure (5) Capital cost of heat exchanger decline in relation to stream velocity.

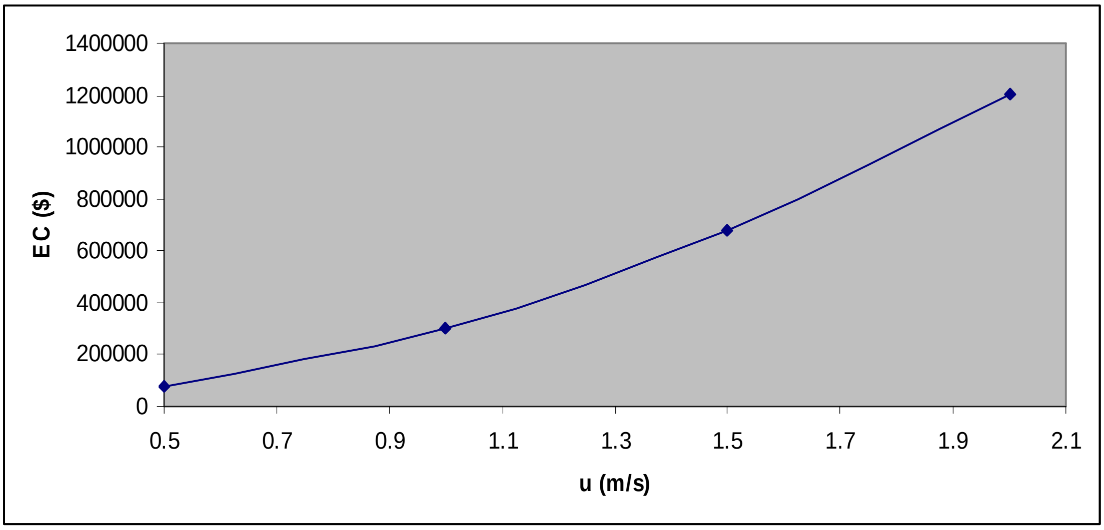

Figure (6) Pump energy cost as a function of stream velocity.

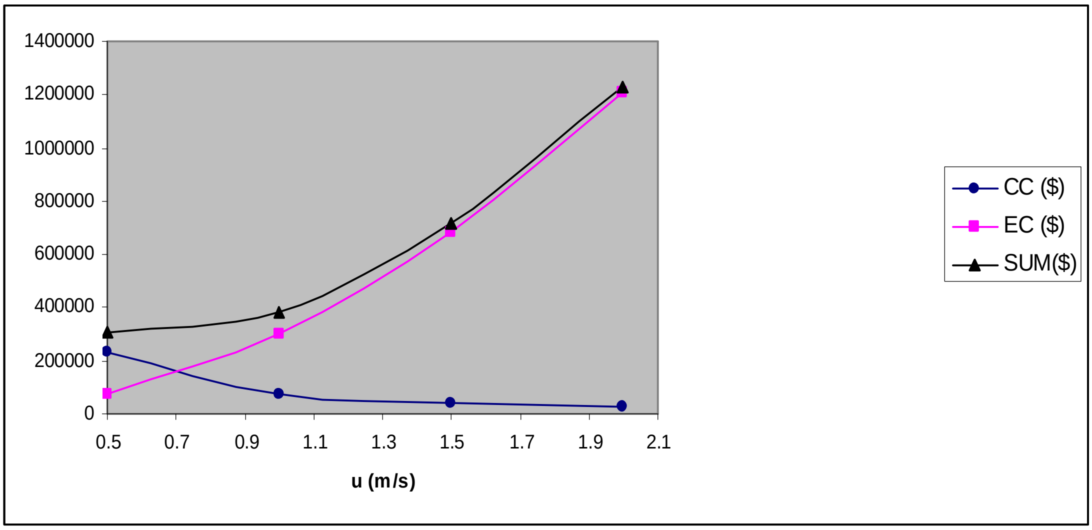

Figure (7) Total cost of double pipe heat exchanger in relation to sludge stream velocity.