THERMAL INACTIVATION OF POLYALACTURONASE IN A HELICALLY COILED HEAT EXCHANGER

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ABSTRACT

THERMAL INACTIVATION OF POLYGALACTURONASE IN A HELICALLY COILED HEAT EXCHANGER

By

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The softening of brined cherries was first noticed about 1947. A pectin degrading polygalacturonase has proven to be responsible for this damage. Since spent cherry brine is a strong pollutant, its disposal into rivers and estuaries or into municipal sewage systems may cause serious pollution problems depending upon the size of the system and the amount of the disposed brine. Recycling of the spent brine seems to be a satisfactory answer to the disposal question. Such a brine must be heated before reusing to prevent softening of the cherries due to polygalacturonase activity.

The basic objective of this study was to determine the amount of inactivation of polygalacturonase while it was receiving a certain and carefully controlled heat treatment in a helically coiled heat exchanger. The degree of inactivation was determined by the "cup-plate" procedure as described by Dingle et al. (1953) and Athanasopoulos (1976). Prediction of the expected inactivation under pre-established conditions was done by both the general and the analytical methods. Predicted and measured values were found to vary insignificantly, sometimes less than -1.0%.

Hot and cold water was used as the heating and the cooling medium, respectively. Side wall friction factors and heat transfer coefficients of a helical tube were found to be higher than those in a straight tube. The magnitude of the coil diameter has a significant effect on the pressure drop and heat transfer through the coils. The over-all heat transfer coefficient was predicted by an exponential relationship of temperature and time in the helix. The predicted values and those calculated by taking into account the resistance to heat flow from one fluid to the other were found to vary from -0.9% to +9.0%. Viscosity, density, thermal conductivity, and specific heat of the treated brine were assumed to be identical to those of water.

Based on heat transfer information and the heat stability of polygalacturonase measured by Athanasopoulos (1976) a temperature of 80°C throughout the holding coil for 9 sec resulted in 99.985% inactivation.

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To my wife, Katherine

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SYMBOLS

A	Area, m^2
A	Proportionality constant in Arrhenius equation, sec^{-1}
b	Temperature ratio in equation 43
c	Specific heat at constant pressure, KJ/Kg-°C
di	Inside diameter of tube, m
D_h	Diameter of the helix, m
D	Decimal reduction time, sec
Ea	Activation energy, cal/mole
f	Fanning friction factor
g	Acceleration of gravity, m/sec^2
h	Convective heat transfer coefficient, W/m^2-°C
k	Thermal conductivity, W/m-°C
k	Velocity constant of enzyme activity, sec^{-1}
K	Time parameter in equation 43, sec^{-1}
L	Length, m
$\dot{m}$	Mass flow rate, Kg/sec
P	Pressure, N/m^2
J	Temperature parameter in equation 43, °K
q	Heat transfer rate, KJ/hr
R	Universal gas constant, cal/g-mol-°K
t	Time, sec
T	Temperature, °C or °K

U	Over-all heat transfer coefficient, W/m^2-°C
V	Velocity, m/sec
∇	Design criterion
W	Mass of brine in contact with the area of the heat exchanger, Kg
x	Characteristic dimension in the dimensionless groups, m

Greek

α	Thermal diffusivity, m^2/sec
β	Coefficient of thermal expansion, 1/°C
ε	Effectiveness
μ	Dynamic viscosity, Kg/m-sec
ν	Kinematic viscosity, m^2/sec
ρ	Density, Kg/m^3

Dimensionless Groups

$D_n = Re(di/D_h)^{0.5}$	Dean number
$Gr = \frac{g\beta(Tw - T)\, x^3}{\nu^2}$	Grashof number
$Gz = \frac{\dot{m}\, c_p}{k\, L}$	Graetz number
$Pr = \frac{c_p\, \mu}{k}$	Prantdl number
$Re = \frac{\rho\, V\, x}{\mu}$	Reynolds number
$Nu = \frac{h\, x}{k}$	Nusselt number

INTRODUCTION

Spent cherry brine is difficult to handle in conventional waste treatment systems since it contains several thousand ppm SO_2, color and a considerable amount of solids. The brine has low pH also.

Recycling of the brine has been proposed as a solution to the pollution problem and several researchers investigated the feasibility of a reclamation process. Spent brine may be contaminated during brining with polygalacturonase. A possible reuse of this material may result in enzymatic softening of the fruit. Therefore, pasteurization of the recycled brine is necessary.

Athanasopoulos and Heldman (1976) examined possible commercialization of a reclamation system based upon the work of other investigators (Soderquist, 1971; Panasiuk et al, 1976). Their pasteurization unit consisted of a helically coiled heat exchanger which was connected in line with the brine reclamation system.

Pasteurization of polygalacturonase in a continuous process should be examined for a commercial operation to be rapid and controlled. Also such a unit will allow a predictable polygalacturonase inactivation.

The objectives of this study were:

1. To determine the amount of inactivation of polygalacturonase present in spent cherry brine after heat treatment in a helically coiled heat exchanger.
2. To review the literature in heat transfer and pressure drop through coils and to modify several equations according to the

situation examined.

3. To calculate the performance of a helically coiled heat exchanger.

LITERATURE REVIEW

A. Heat Transfer and Pressure Drop in Coils

Coils have two major advantages over straight tubes, namely:

(a) heat transfer coefficients in coils are higher than in straight tubes,

(c) coils allow a greater heat transfer area to be packed into a given space more economically and more conveniently than in straight tubes.

However, it is much easier to remove scale from straight tubes than in coils. Friction factors are also higher in coils.

The first theoretical analysis of the flow of noncompressible fluids through helices was done by Dean (1927), who predicted the flow in the helix to be as illustrated in Figure 1, which shows a cross section of the tube in the helix.

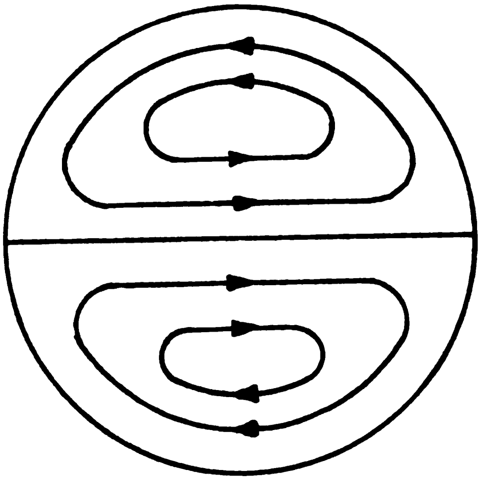


Figure 1. Cross section of a tube in a helix showing secondary flow.

The centrifugal force that exists in a curved pipe creates the secondary flow. This type of flow causes a larger pressure drop and heat transfer rate than that occurring in straight tubes. Dean introduced a dimensionless group which characterizes the dynamic similarity of fluid flow through a helix. This dimensionless group was named Dean's number (Dn) and it is equal to:

$$Dn = Re(di/D_h)^{0.5}$$

where Re = Reynolds number

di = inside diameter of the tube (cm)

D_h = diameter of the helix (cm)

A-1. Pressure drop.

Laminar Flow

The friction loss in a tube is given by the following equation:

$$\Delta H = 2fLV^2/g_c di \qquad (1)$$

where ΔH = Head loss (N/m^2)

L = Length of the tube (m)

di = Inside diameter of the tube (m)

g_c = Acceleration of gravity (m/sec^2)

f = Fanning friction factors, dimensionless

V = Velocity of the fluid (m/sec)

In straight tubes $f = 16/Re$. Therefore, frictional losses are highly dependent on Re. The parameters on the left side in equation (1) are all fixed for a certain process, except f. Thus, determination of f is of great importance in pressure loss calculations.

Based on Dean's theoretical analysis, White (1929) proposed the following equation:

$$f = C(16/Re) \qquad (2)$$

$$\text{for } 11.6 < Dn < 2000$$

$$\text{where } \frac{1}{C} = 1-[1-(11.6\frac{(D_h/di)^{0.5}}{Re})^{0.45}]^{1/0.45}$$

Since C is dependent on Re and D_h/di, high velocities and small helix diameters will contribute to high pressure losses.

For both isothermal and non-isothermal flow, Seban _et al_. (1963), taking into account the overall pressure drop, deduced the friction factor on the basis of fluid properties at the mean film temperature. They obtained results which were about 8 percent less than those obtained by equation (2).

Flow of air within a curved tube was examined experimentally and analytically by Mori _et al_. (1965). Using an approximation technique for a series solution they derived the following equations for the first and second approximation:

$$(f_c/f_s)_I = 0.1080Dn^{0.5} \qquad (3)$$

$$(f_c/f_s)_I = (f_c/f_s)_I / \frac{1-3.253}{Dn^{0.5}} \qquad (4)$$

where f_c = Fanning friction factor for coiled tube

f_s = Fanning friction factor for straight tube

Equation (4) gave results similar to those obtained with the following equation:

$$f_c/f_s = 21.5\ \mathrm{Dn} \ / \ (1.56 + \log \mathrm{Dn})^{5.73} \qquad (5)$$

Ito (1959) derived the above equation for

$$13.5 < \mathrm{Dn} < 2000$$

The use of the ratio f_c/f_s for pressure drop determination has the disadvantage that it cannot be used for Re greater than 2100 because under such conditions, the flow in a straight tube ceases to be in the laminar region while the flow in coiled tubes may be in the laminar region at much higher Re (Srinivasan et al. 1968).

The Critical Reynolds Number

The highest Re where the viscous forces still have an effect to the dynamic forces streamline, is called critical Reynolds number. While in straight tubes this value of Re is fixed, at approximately Re 2100, in coiled tubes the curvature ratio determines the magnitude of the critical Re.

Ito (1959) related the critical Re and the curvature ratio as follows:

$$(\mathrm{Re})_{crit} = 20000(di/D_h)^{0.32} \qquad (6)$$

According to Srinivasan et al. (1968) the critical Re can be calculated as follows:

$$(\mathrm{Re})_{crit} = 2100(1 + 12(di/D_h))^{0.5} \qquad (7)$$

Turbulent Flow

The work on pressure losses in smooth coiled tubes with isothermal turbulent flow was summarized by Ito (1959), who proposed two equations based on the results of several workers, namely

$$f_c = 0.0076(\mathrm{Re})^{-0.25} + 0.00725(di/D)^{0.5} \qquad (8)$$

for $0.034 < \mathrm{Re}(di/D_h)^2 < 300$

and

$$f_c/f_B = [\mathrm{Re}(di/D_h)^2]^{0.05} \qquad (9)$$

for $\mathrm{Re}(di/D_h)^2 > 6$

where f_B is the Blasious value for a straight tube

$$f_B = 0.0791/\mathrm{Re}$$

For Re in the range of 6000 to 65600, Seban et al. (1963) obtained friction factors for isothermal and non-isothermal conditions. Their experimental results and those of Ito were found to be in good agreement, although they took the friction factor for a straight tube as $f_s = 0.046/\mathrm{Re}^{0.2}$.

Rogers and Mayhew (1964) experimentally obtained pressure drop results for isothermal conditions through coils for Re in the range of 3000 to 50000. They used coils with a different curvature ratio and their results agreed within 1.5 percent of those of Ito's equations. For non-isothermal conditions they suggested the following equation:

$$f_c = f_{Ito}\,[(Pr)_b/(Pr)_w]^{-1/3} \tag{10}$$

Where f_{Ito} is given by equation (8) or (9), the subscripts b and w refer to the properties estimated at the bulk and wall temperature respectively. From the practical point of view, the best equation may be that published by White (1932):

$$f_c = 0.08\,Re^{-0.25} + 0.012(di/D_h)^{0.5} \tag{11}$$

for $1500 < Re < 100000$

The simplicity of equation (11) is evident and since it can be applied over a wide range of Re it can be used in various industrial helices where high velocities are encountered.

In their investigation of air flowing in curved pipes, Mori et al. (1967) developed a model which gave satisfactory results in comparison to equation (8) and to the experimental work of Seban et al. (1963) and Rogers et al. (1964). Their model can be simplified as

$$f_c = 0.076\,Re^{-0.25} + 0.00725(di/D_h)^{0.5} \tag{12}$$

A-2. Heat transfer in coils.

Generally the heat transfer in tubes depends upon two parameters, namely Reynolds and Prandtl number. In coiled tubes a third parameter is introduced, that is the curvature ratio, di/D_h, which associates the effect of the secondary flow in the mechanism of heat transfer.

Laminar Flow

For air, water, and Essolube 30 oil in laminar flow within a coil, Berg and Bonilla (1950) derived the following equation:

$$\frac{hi\ di}{c_p\ \mu_f} = [0.0000229 + 0.00063(di/D)]\ (Re)_f^{1.29} \tag{13}$$

where hi = inside heat transfer coefficient ($W/m^2.°C$)

μ_f = dynamic viscosity at the film temperature (Kg/m.sec)

c_p = specific heat of fluid (KJ/Kg.°C)

They found higher hi for oil and lower hi for air to the corresponding values of hi in straight tubes, although it is evident that coils have a higher hi.

The average hi is specified by Seban et al. (1963) as

$$\frac{hi\ di}{k} \left(\frac{v}{\alpha}\right)^{-1/3} = A\ [f/8(Re)_f^2]^{1/3} \tag{14}$$

for $12 < Re < 5600$

$100 < Pr < 675$

where v = Kinematic viscosity (m^2/sec)

α = Thermal diffusivity (m^2/sec)

f = Friction factor for coiled tube

The disadvantage of equation (14) is that it was applied for two cases of D_h/di (17 and 104), therefore, a relatively applicable value of A was not fixed. The effect of di/D_h in the mechanism of heat transfer was also neglected.

Kubair and Kuloor (1966) related Nusselt and Graetz numbers in heated water and glycerol as follows:

$$Nu = [1.98 + 1.8(di/D_h)]\ Gz^{0.7} \qquad (15)$$

for $10 < Gz < 1000$

$80 < Re < 6000$

$20 < Pr < 100$

A modification of the Dittus-Boelter equation was applied by Edney et al. (1973) to determine hi in a coil within an agitated vessel. Their results were in satisfactory agreement with previous works. The equation can be simplified as

$$Nu = 0.0225\ F_c\ (Re)^{0.8}(Pr)^{0.4} \qquad (16)$$

for $4.01X10^2 < Re < 9.2X10^5$

where F_c is a coil factor, ratio of hi in a coiled tube to that in a straight pipe.

Equation (16) may be valid within an agitated vessel but in the absence of agitation the validity is questionable.

Turbulent Flow

A correlation for turbulent flow and heat transfer data as reported by Seban et al. (1963) resulted in the equation:

$$(Nu)_f = 0.023(Re)_f^{0.85}(Pr)_f^{0.4}\ (di/D_h)^{0.1} \tag{17}$$

for $6000 < Re < 65600$

$15 < Nu < 240$

$2.9 < Pr < 5.7$

It is obvious that the equation is a modification of the Dittus-Boelter equation.

Later experimental work of Rogers et al. (1964) confirmed the validity of equation (17) with a constant value equal to 0.021 instead of 0.023.

While the effect of the curvature ratio (di/D) in pressure drop and heat transfer in coils is taken into account in every equation described previously, the effect of pitch, which is the shortest distance between the centers of two consecutive turns of the helix, has not been considered.

B. Shell-Side Heat Transfer and Pressure Drop

B-1. Pressure drop.

The flow in the shell-side is very similar to the flow normal to the tube banks of parallel tubes. McAdams (1954) suggests the following equation for flow over banks:

$$f = \frac{(-\Delta P)N}{2(m/A_f)^2} \tag{18}$$

For the same case Kreith (1961) proposed the following relationship:

$$f = [0.044 + \frac{0.08\ AP}{(RP-1)^n}]\ Re^{-0.15} \tag{19}$$

where ΔP = Pressure loss (N/m^2)

N = Number of tubes transverse to flow

m = Mass flow rate (Kg/sec)

A_f = Shell-side flow area (m^2)

AP = Axial pitch - $\frac{\text{axial separation + tube diameter}}{\text{tube diameter}}$

RP = Radio separation - $\frac{\text{radial pitch + tube diameter}}{\text{tube diameter}}$

f = Friction factor

$n = 0.43 + 1.13/AP$

Messa *et al*. (1970) found that equation (18) predicts the pressure data in the shell-side more accurately than equation (19).

B-2. Shell-side heat transfer coefficients.

The mechanism of flow and heat transfer on the outside of the coils is very complicated and no analytical solution to the problem has been found. Kern (1950) suggested that, in cases where the velocity of the fluid that flows over banks of tubes is very small, the following equation proposed by McAdams (1947) could be used:

$$Nu_f = 0.53\ [Gr_f\ Pr_f]^{0.25} \tag{20}$$

Since equation (20) was developed for pure natural convection, its use in situations, as flow over banks, where natural and forced convection coexist, may lead to considerable error.

Messa et al. (1970) successfully predicted the heat transfer coefficients in the shell-side of a helically coiled heat exchanger using the following equation:

$$Nu = 0.0085\ Re^{0.84}\ Pr^{0.57} \tag{21}$$

Their Re varied between 2000 and 10000. Equation (21) was not checked using smaller Re values.

Fand and Keswani (1973) experimentally studied convective heat transfer under conditions where natural and forced convection coexist. They used an electrically heated cylinder with water flowing in the vertically upward, vertically downward and horizontal directions. Their data were related to four zones depending on the ratio Gr_f/Re^2.

First Zone

$$Gr_f/Re^2 < 0.5 \tag{22}$$

The predominant heat transfer mechanism in this region is forced convection. Application of published forced convection equations may be applied with negligible error (± 5%).

Second Zone

$$0.5 < Gr_f/Re^2 < 2$$

In this case the heat transfer is still due to forced convection. However, natural convection affects the over-all heat transfer coefficient by as much as 10 percent. The heat transfer coefficient in this region is given by,

$$Nu = [0.225 + 0.699Re^{0.50} + C(Gr_f/Re^2)^m \, Gr_f^{0.25}]Pr^{0.29} \tag{23}$$

where m is a constant equal to 0.30, 0.15, and 1.0 for upward, downward, and horizontal flow respectively.

Third Zone

$$2 < Gr_f/Re^2 < 40$$

In the third zone natural and forced convection effects are of the same order of magnitude. In this region it is very difficult to reach steady state conditions because of the two extreme conditions (predominance of forced and predominance of natural convection). Heat transfer coefficients are, therefore, difficult to predict.

Fourth Zone

$$Gr_f/Re^2 > 40$$

This is the region where natural convection predominates. The Nu in this region is given as,

$$Nu_f = C\ [(KGr_f)^{0.5} + Re_f^2]^{0.5}\ Pr^{0.25} \tag{25}$$

where C, and K are constants having different values depending on the geometry and the tube arrangement.

C. Performance of Heat Exchanger

In a conventional direct-type heat exchanger the parameters relating to the heat transfer performance are as follows (Kays and London, 1955):

U = The over-all heat transfer coefficient (W/m^2.°C)

A = The surface area on which U is based (m^2)

Th_i = Inlet temperature of hot fluid (°C)

Th_o = Outlet temperature of hot fluid (°C)

Tc_o = Outlet temperature of the cold fluid (°C)

Ch = $(\dot{m}c_p)_h$ the hot fluid capacity rate (W/°C)

Cc = $(\dot{m}c_p)_c$ the cold fluid capacity rate (W/°C)

Flow arrangement - i.e. counterflow, parallel flow, crossflow, parallel-counterflow or combinations of these basic arrangements.

The over-all heat transfer coefficient combines the convective and conductive mechanisms responsible for heat transfer from the hot to the cold fluid into a relationship similar to Ohm's Law. This relationship can be expressed as:

$$\frac{1}{Ui} = \frac{1}{hi} + \frac{AiX}{Awkw} + \frac{Ai}{A_o h_o} + \frac{1A_i}{h_d A_o} \tag{25}$$

where Ui = is based on the inside area

Ai = Inside area (m^2)

Aw = Average wall area

X = Wall thickness (m)

A_o = External or outside area

kw = Thermal conductivity of the wall (W/m.°C)

1/hd = Resistance due to scale on the tubes

In situations where the wall is very thin and the resistance due to scale deposit is negligible, equation (25) is reduced to,

$$\frac{1}{U} = \frac{1}{hi} + \frac{X}{kw} + \frac{1}{ho} \tag{26}$$

Exchanger Heat Transfer Effectiveness

According to Holman (1976)

$$\text{effectiveness} = \varepsilon = \frac{\text{actual heat transfer}}{\text{maximum possible heat transfer}}$$

or

$$= \frac{C_h(T_{hi}-T_{ho})}{C_{min}(T_{hi}-T_{ci}} = \frac{C_c(T_{ci}-T_{co})}{C_{min}(T_{hi}-T_{ci})} \tag{27}$$

where C_{min} is the smaller of the C_h and C_c magnitudes. In general it is possible to express

$$\varepsilon = F(NTU,\ C_{min}/C_{max},\ \text{flow arrangement})$$

Where NTU (Number of Heat Transfer Units) is expressed as,

$$NTU = \frac{AU_{av}}{C_{min}} = \frac{1}{C_{min}} \int_o^A UdA \tag{28}$$

NTU and ε are related asymptotically for a given capacity ratio. Therefore, where NTU is small, the effectiveness is low and when NTU is large the effectiveness approaches the value of unity asymptotically, or the actual heat transfer approaches the value of the maximum possible heat transfer.

D. Recycling the Cherry Brine

Sweet cherries to be processed into maraschino, cocktail, or glacé fruit, are placed, directly after harvesting, into barrels that contain a brine solution of sulfur dioxide and various calcium salts. The brining process prevents quality deterioration of the fruit and further bleaches the cherries to a bright light yellow color. The bleached cherries also are stored in brine solution for later distribution in the market. After use, the brine may be discharged to a municipal sewage system or it may be stored in tanks, where the solids are removed by settling and the liquid is sprayed on land. The latter is adopted by a number of plants in Michigan.

Spent brine contains a high content of solids, SO_2, Ca^{++} and color pigment. In addition, it is characterized by low pH (Table A-1). Such a material is considered as a very strong pollutant. According to Soderquist (1971) a plant discharging spent brine at a rate of $38m^3$/day into the sewage system would be exerting a load (in terms of degradable organics) equivalent to 40,000 people or more.

Brine recycling and reuse would solve a pollution problem and further lower new brine preparation costs. In 1970 Beavers et al. introduced the idea of reclaiming the brine by passing it through activated carbon. Soderquist (1971) and Panasiuk et al. (1975) examined the feasibility of a brine reclamation system and proved that treatment of brine by passing it through activated carbon does not lower the content of SO_2, Ca^{++} and soluble solids nor does it affect the pH, while cherries stored in reclaimed brine were found to have superior quality to those in a freshly prepared brine.

Athanasopoulos and Heldman (1976) developed a large scale brine reclamation system based on the work of previous investigators. The system consisted of the following units:

a. sand filter for suspended solids removal,

b. six columns that contained activated carbon for anthocyanin and polyphenol removal, and

c. pasteurization unit for the inactivation of PG enzyme.

E. The Polygalacturonase Enzyme

E-1. Polygalacturonase in brined cherries.

Softening of brined cherries was first noticed about 1947 (Wiegand, 1954). In 1954 McCready reported that soft brined cherries seem to be attacked by a pectin degrading enzyme, since the stems fall off and part of the outer skin begins to slough. Fresh cherries infected with "cats claw" type of spoilage were found to contain an excessive amount of Polygalacturonase that causes softening of the fruit when brined (Steele *et al*. 1960).

The polygalacturonase is a pectic enzyme capable of hydrolyzing 1,4-glycositic linkages of pectinic and pectic acids (Kertesz, 1951). Possible sources of polygalacturonase in brined cherries may be from microbial growth, certain cherry diseases, or the fruit itself might synthesize the enzyme during ripening (Steele *et al*. 1960). The enzyme sometimes remains active in the brine more than a year, and exposure to this brine produces soft fruit (Brekke *et al*. 1966).

Several chemical methods have been proposed for polygalacturonase inactivation but their commercialization is in doubt since the safety of the chemicals involved may still be questioned. Inactivation by heat might be the only approach because no chemicals are needed (Soderquist, 1971; Walters *et al*. 1961).

E-2. Polygalacturonase kinetics, rate constants and stability.

The inactivation of an enzyme and the thermal destruction of an organism can be described by first order kinetics,

$$dC/dt = -kC \tag{29}$$

where dC/dt = activity change with time

C = activity at any time

k = velocity constant of the reaction (sec^{-1})

By integration equation (29) gives,

$$\ln C/C_o = -kt \quad \text{or } \log C/C_o = -kt/2.303 \tag{30}$$

It is obvious that C in this relationship can never reach zero. By introducing the factor D (=2.303/k), which stands for decimal reduction time at a certain temperature, equation (30) gives,

$$C/C_o = 10^{-t/D} \tag{31}$$

Plotting temperature versus D on semi-log paper a straight line is obtained, which is called the "thermal-death time" curve of a microorganism or an enzyme. The reciprocal of the slope of the line is termed as the Z value. The straight line can be expressed as,

$$D_{T_1} = D_{T_2} \text{ X } 10^{\frac{T_2 - T_1}{Z}} \tag{32}$$

If the Z value is large the temperature has a smaller effect on the TDT than if Z is small. Z and D values are fixed for a certain organism or enzyme under specified conditions. Athanasopoulos (1976) experimentally and theoretically determined the Z value of polygalacturonase to be 8.4°C and the D value to be 44.76 sec at 70°C.

The velocity constant k can be expressed by the Arrhenius equation as follows:

$$k = A \exp(-Ea/RT) \tag{33}$$

where Ea = activation energy (cal/mole)

R = universal gas constant (cal/g.mole.°K)

T = absolute temperature (°K)

For a given environment, the velocity constant, k, for spores of a particular species will be a function of temperature only. Equation (33) can also be expressed as,

$$\log k = \log A - Ea/2.303RT \tag{34}$$

which is the "Arrhenius Plot" of the reaction. The slope of the line is -Ea/2.303R and the intercept log A. The activation energy of polygalacturonase is 64668.4 cal/mole (Athanasopoulos, 1976).

The inactivation rate constants are highly pH dependent. The pH and decimal reduction time relationship is not linear because of the enzyme being more resistant at pH 2.8 to 3.5 (Athanasopoulos, 1976). Sugar concentration also affects the stability of the enzyme. The decimal reduction time does not change significantly when the soluble solids of the brine are in the range of 9 to 12%. However, when soluble solids exceed 12% they affect the inactivation rate constants since the strength of the brine varies from one variety to another, influencing the composition (SO_2 content, pH, solids). Windsor brine has a higher concentration of SO_2, Ca^{++}, and pigments, and exhibits lower rate constants than Napoleon brine.

F. Polygalacturonase Inactivation Measurements and Calculation

F-1. Enzyme activity measurements.

The activity of an enzyme can be measured by determining the substrate losses or by measuring certain changes in the physical properties such as viscosity. Runklyadeva and Korchagina (1975) introduced the interferometric method for commercial pectic enzyme preparation. The method is rapid (less than two hours) and accurate (± 2%). Recently, the solid media technique has been introduced in the quantitative determination of enzyme activity. This procedure, developed for microorganism cultivation, has been used for detection of enzymes by microorganisms (Hankin and Anagnostakis, 1975).

The agar "cup-plate" diffusion procedure has been applied to the quantitative determination of polygalacturonase (Dingle et al. 1953). This test has been used by Steele and Yang (1960) and Beavers et al.(1970) for polygalacturonase activity determination in cherry brine.

F-2. Process calculation.

The general method, first described by Bigelow et al. (1920), was an extension of partial sterilization. Ball et al. (1957) defined partial sterilization as follows: if the TDT is t_1 min at T degrees, then heating which lasts for t_2 min at T degrees, results in sterilization which is t_2/t_1 complete. Ball proved that sterilities are additive. In 1928 Ball introduced a hypothetical TDT curve passing through 1 min at 250°F. Since lethality of a given process is the time in min of an equivalent process at 250°F, then F in Ball's hypothetical TDT curve is equal to 1, where F is the time required to destroy an organism at 250°F. According to this information equation (32) can be rearranged as follows:

$$\log t/F = \frac{250-T}{Z} \qquad (35)$$

where t, is the time in min to destroy the organism at T degrees. Since F is 1 min, equation (35) is reduced to:

$$\log t = \frac{250-T}{Z} \qquad (36)$$

or

$$t = 10^{\frac{250-T}{Z}} \qquad (37)$$

and

$$1/t = 10^{\frac{T-250}{Z}} \qquad (38)$$

where 1/t is defined as the lethal rate (L) at temperature T. Therefore, for a given process where Z is fixed and the temperature time relationship is known, the total lethality (degree of inactivation) can be calculated since lethalities are additive.

Stumbo (1940, 1949, 1953) showed that in the case of conductive heating, a process with F=4 does not have twice the sterilizing capacity of another process with F=2min.

Deindoerfer and Humphrey (1959) introduced the following analytical method:

By integration of equation (29)

$$\ln Co/C = \int_0^t kdt = \nabla \qquad (39)$$

where ∇, the design criterion, is a measure of the size of the job to be accomplished or the degree of inactivation. If k varies in accordance with the Arrhenius equation, equation (39) gives

$$\nabla = A \int_0^t \exp(-Ea/RT)\, dt \qquad (40)$$

at constant temperature

$$\nabla = At \exp(-Ea/RT) = -kt \qquad (41)$$

In continuous sterilization processes, where the medium flows through the heating, cooling, and holding section, ∇ for heating and cooling is calculated from equation (40) and ∇ for holding from equation (41). Therefore:

$$\nabla_{total} = \nabla_{heat.} + \nabla_{cool.} + \nabla_{hold.} \qquad (42)$$

For a certain process, if the temperature-time profiles are known, then equation (40) results in integrals that can be solved analytically.

The disadvantages of the method are: (a) it assumes that the entire population consists of spores of the design species, and (b) it requires heat transfer information for the analytical solution of the integrals.

Another method devised by Stumbo (1949), Hicks (1951), and Gillespy (1962), bases the process time on the probability survival in the whole container.

The following assumption is another approach to the problem: if it is assumed that after a process the slowest point is safe, then the rest of the fluid which must have a longer heating time must be safe.

Richards (1965) has adopted a graphical solution which provides a ready means for the determination of the normal sterilization cycle. The time-temperature profile for a batch sterilization is plotted over the entire cycle. The cumulative values are then determined by stepwise integration over the complete cycle.

METHODS AND PROCEDURES

A. Instruments and Materials

Instruments

Figure 2 shows a diagramatic description of the apparatus used in the experiment. The pasteurization unit consisted of three sections, namely, heating, holding, and cooling. The helix in the heater was 5.8 m (19 ft) long, in the cooler 3.97 m (13 ft), and in the holding section 4.88 m (16 ft). The tubes were made of stainless steel. The specifications of the unit are given in Table 1.

An open pressure cooker was used to maintain the heating medium at a constant and controlled temperature. The heating medium was recirculated using a centrifugal pump and the flow rate was measured with a rotameter located in the line downstream from the pump. To lessen scale build up on the outside of the brine coil in the heater the heating medium was passed through a copper coil immersed in a hot water bath. This enclosed indirect heating system reduced the amount of scale build up essentially to zero.

The flow rate of the brine was of great importance. Steady-state conditions within the pasteurization unit could not have been achieved with a fluctuating flow rate. A varying flow rate would result in a fluctuating and unpredictable degree of enzyme inactivation. The problem was solved by using two pumps and a supply tank positioned four

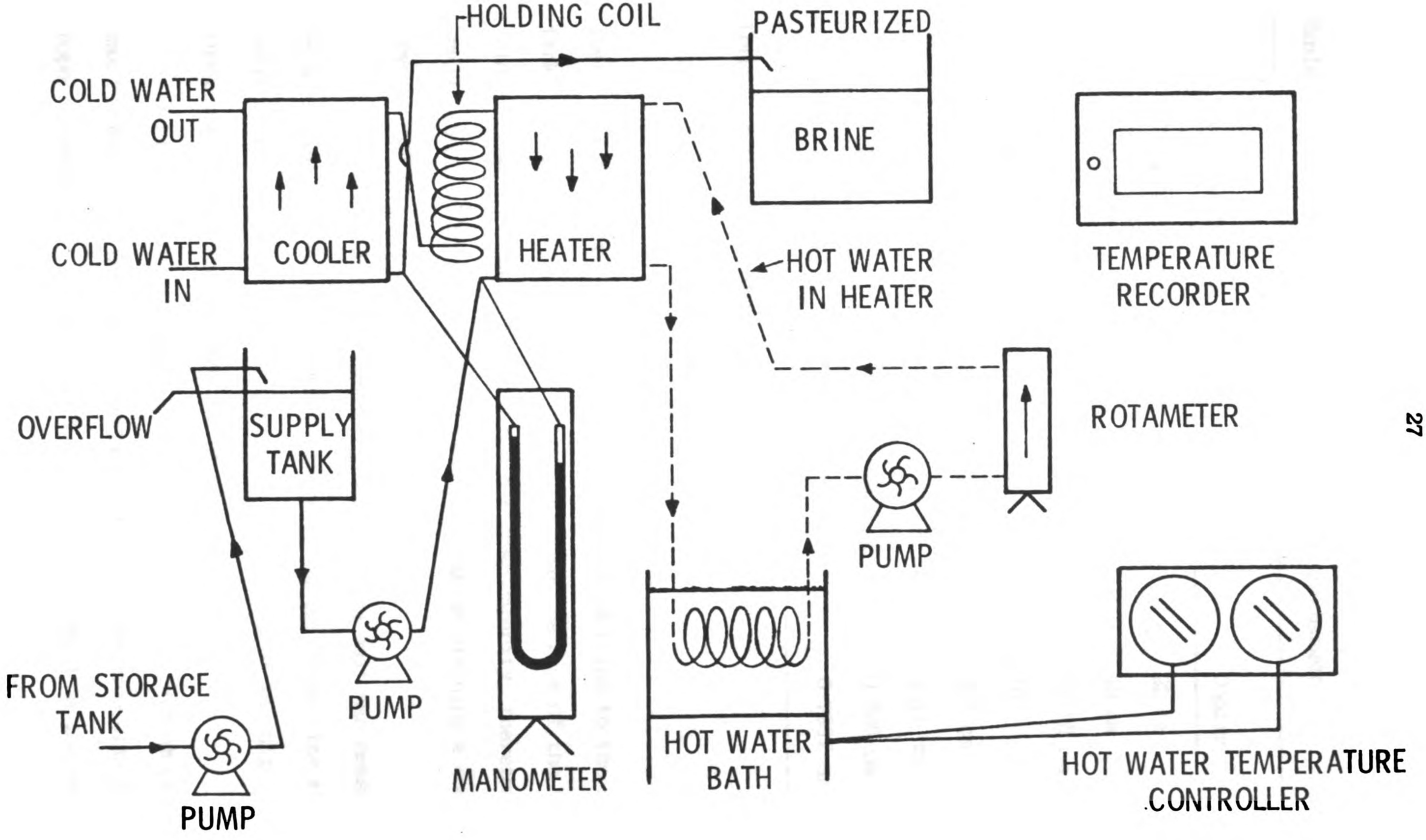


Figure 2. Schematic diagram of heating and cooling system for pasteurization of cherry brine.

Table 1. Specifications of Experimental Heat Exchanger.

Specification	Heater	Cooler
Shell diameter	22 cm	22 cm
Shell height	43 cm	43 cm
Helix diameter	19.05 cm	19.05 cm
Number of turns	10	10
Tube length	580 cm	397 cm
Pitch	3.18 cm	3.81 cm
Tube I.D	0.635 cm	0.635 cm
Tube O.D	0.7937 cm	0.7937 cm

Length of tube in holding section: 488 cm

feet above the pumps. The first pump transferred brine to the supply tank and the second pump pumped the brine from the base of the tank into the heater. The first pump had a higher capacity, therefore, a small amount of brine overflowed from the tank maintaining a constant level at 30 cm.

The pressure drop through the pasteurization unit was measured by attaching one end of a mercury manometer in the brine line at the entrance of the heater and the other at the outlet of the unit. Therefore, the pressure drop was measured in mm of mercury.

The temperature at several points throughout the system (Figure 3) was recorded on a Honeywell potentiometer. All thermocouples were copper-constantan wire of equal length and calibrated in ice water.

Petri dishes with dimensions 15 X 85 mm were used for the "cup-plate" procedure. Small plastic collars (9 mm diameter) were placed in the center of the dish before the addition of the substrate in order to provide cups into which the enzyme-brine solution was added using a syringe.

Materials

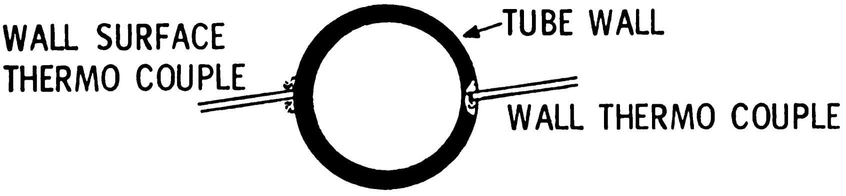
Hot and cold water were used for the heating and cooling medium respectively.

The constant temperature along the holding coil was considered to be the process temperature, which in turn, determined the hot water temperature. Because the amount of inactivation desired was determined by the process temperature, the degree of inactivation was directly related to the hot water bath temperature. The heating medium was flowing downward in the heater and the brine upward in the coil as indicated in Figure 3; therefore, the liquids were in a counterflow arrangement.

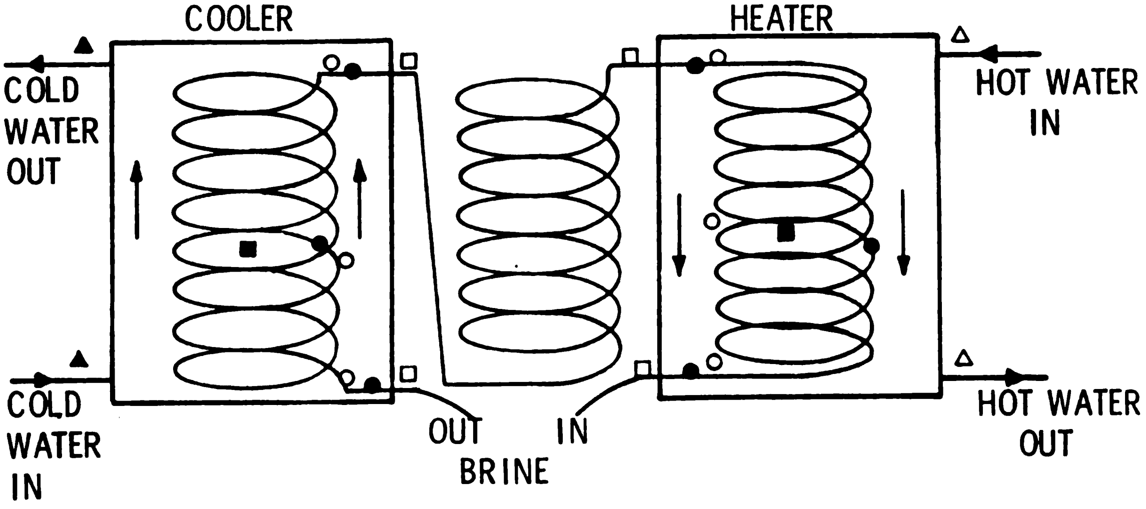
Tap water was selected for the cooling medium, which flowed upward in the cooler. The brine flowed downward in the cooler to produce a counterflow arrangement.

The flow rate of the hot and cold water was adjusted to 3.78 Kg/min and 5.6 Kg/min respectively, while that of the brine was 1 Kg/min.

Brine of Napoleon variety was chosen for the experimental trials. The brine was supplied by a plant located in Traverse City, Michigan. Extraneous matter including color, stems, leaves and seeds were removed using a reclamation system developed by Athanasopoulos and Heldman.



Attachment of the bare junction in the wall
and on the wall surface of the tube.



△ Inlet and outlet hot water stream temperature

▲ Inlet and outlet cold water stream temperature

○ Wall surface temperature measurement

● Wall temperature measurement

□ Brine stream temperature measurement

■ Bulk temperature measurement

Figure 3. Thermocouple stations for wall, wall surface and fluid stream temperature measurement.

Reclaimed brine was found to have a pH of 3.1 and 10.5% soluble solids, thereby eliminating further adjustment or pretreatment.

Purified commercial polygalacturonase (Pectinase[1]) from Aspergillus niger was used in this study at a concentration of 1 mg of the enzyme per 1 ml of brine.

The substrate of the enzyme for the "cup-plate" procedure was made from the following:

Distilled water	1000 ml
Polygalacturonic acid	2.5 g
Ammonium oxalate	2.5 g
Bacto agar	20 g
Potassium acid phthalate	10 g
Thymol	0.25 g

All of the substances were mixed in a 2 liter Erlenmeyer flask and the mixture was sterilized at 120°C for 5 min.

B. Temperature Measurements

The properties of the fluids used in the experiment were determined at the mean film temperature conditions (T_f) using the equation

$$T_f = \frac{T_b + T_w}{2}$$

where T_b = mean bulk temperature (°C)

T_w = mean wall temperature (°C)

To obtain the wall temperature a U shaped groove was cut on the inner part of the tube wall of the helix. The bare junction of the thermocouple

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[1] Pectinase is the trade name for polygalacturonase supplied by SIGMA Chem. Co.

was placed in the groove in direct contact with the metal wall of the tube. The groove was then filled with a sticky plastic material in order to have adequate insulation. Three thermocouples were installed along the helix, with one at each end and one in the middle. The average of the three temperatures was considered to represent the mean wall temperature of the coil on the inner side of the heat exchanger.

For the shell-side measurements, the following equation was used:

$$T_f = \frac{T_s + T_b}{2}$$

where T_s = wall surface temperature (°C)

Wall surface temperatures were measured using the same procedure as described for the wall except the thermocouples were attached on the surface of the wall instead of the U shaped groove.

The bulk temperature for the streams in each section was measured by connecting a rubber tube on the entrance and the exit of each stream and placing the bare thermocouple junction in the center of each tube. The mean bulk temperature was considered to be the average of the inlet and the outlet stream temperatures. For the shell-side measurements, three individual bulk temperatures were recorded: at the entrance, the outlet, and the center of the shell. In Figure 3 the several positions of the attached thermocouples are indicated, while in Appendix C the recorded temperatures at the various positions for four different trials are presented.

C. Pressure Drop and Heat Transfer Determination

The ability of the apparatus to maintain a desired heat treatment condition was checked with water in place of cherry brine. More than 100 trial runs were made to achieve the desired accuracy. Before making heat treatment trials on brine, the pH and soluble solids content were determined. An ABBE refractometer and a pH meter were used. With soluble solids of 10.5% and pH of 3.1 no adjustment was necessary. The instrumentation was also checked on the brine in the same manner as was done previously on water. Temperature readings were taken after steady-state conditions had been attained in a particular run which usually required 10 min.

C-1. Pressure drop.

The tube side or coil side head loss was examined under the following conditions:

Processed liquid: water
Flow rate: 1-2 Kg/min
Fluid temperature: 11,5-70 °C
Wall temperature: 32.5-60 °C
Reynolds number: 5051-10092
Critical Re: 6700

The properties of the water were evaluated at mean film temperature conditions. The critical Re was determined by equation (7). Laminar friction factors were obtained with equation (5), while equation (9) was used for turbulent flow. For straight tubes $f = 16/Re$ and $f = 0.046/Re^{0.2}$ for laminar and turbulent flow respectively. After the friction factors were evaluated, the pressure drop was calculated by applying equation (1).

C-2. Heat transfer calculations.

Inside heat transfer coefficients were evaluated under the following experimental conditions:

Processed liquid: Brine
Brine flow rate: 1 Kg/min
Reynolds number: 5350-7300
Prandtl number: 2.93-4.6

As in pressure drop measurements the properties of the brine were evaluated at mean film temperature conditions, and they were assumed to be identical to those of water. Because the present work and that of Seban et al. (1963) and Rogers et al. (1964) were carried out under similar experimental conditions, equation (17) with a constant value of 0.21 was used, for the inside heat transfer coefficient determination.

The experimental conditions for the outside heat transfer coefficient estimation were as follows:

Hot water flow rate: 3.78 Kg/min
Cold water flow rate: 5.6 Kg/min
Grashof number: $5X10^{6}-1.2X10^{7}$
Reynolds number: 50-70
$Gr_f/Re^2 > 40$

The ratio $Gr_f/Re^2 > 40$ indicates that both natural and forced convection coexist, the natural being the predominant. Therefore, equation (24) should be used, where $K = 1.7$ and $C = 0.235$. In the cooler, natural convection did not seem to exist since the water was flowing upward while the density was increasing downward. It was assumed that, because the velocity of the water was very small (1 m/min), the equations used for the heater would apply.

Reynolds number calculations were based on the maximum velocity. The maximum velocity was considered to exist in the smallest cross sectional area. As the water flows upward (cooler) or downward (heater) two areas are involved: one between the shell wall and the coil (0.0162 m^2) and the other which corresponds to the diameter of the coil (0.0248 m^2). Therefore, the Reynolds number was based on an area of 0.0162 m^2.

Assuming no scaling deposit exists on the tubes, equation (26) was used for over-all heat transfer coefficient calculations where k_w for stainless steel equals 16.26 W/m.°C.

The over-all heat transfer coefficient was also predicted by noting that the wall temperature was found to vary exponentially. If it is assumed that the stream temperature will vary the same way, then according to Deindoerfer <u>et al</u>. (1959),

$$T = J\,(1 + be^{-Kt}) \tag{43}$$

where T = stream temperature at any time t (°K)

$$b = \frac{T_o - T_H}{T_H}$$

T_o = stream temperature (°K)

$J = T_H$

$$K = \frac{U\,A}{W\,c_p}$$

U = over-all heat transfer coefficient

W = mass of flowing medium in contact with the surface area A (Kg)

A = surface area across which heat transfer occurs

c_p = specific heat (KJ/Kg.°C)

After time t, the medium will exit the heater at a certain temperature T_{exit}. Equation (43) gives

$$-Kt = \ln \left(\frac{T_H - T_{exit}}{T_H - T_o}\right) \tag{44a}$$

T_H, T_{exit}, and T_o can be measured, while t could be calculated based on the flow rate and the diameter of the tube. Therefore, K may be predicted relatively easily. A, W, and c_p are also known or predictable. Then

$$U = \frac{K\,W\,c_p}{A}$$

For the cooling section, $b = T_o - T_c/T_c$, where T_c is the temperature of the cooling medium (°K). After time t', the medium will exit the cooler at temperature T' exit. Equation (43) then is reduced to

$$-Kt' = \ln \left(\frac{T'_{exit} - T_c}{T_o - T_c}\right) \tag{44b}$$

Where K is based on the U, A, W, and c_p of the cooler. The over-all heat transfer coefficient may be calculated as given previously.

An energy balance in each section of the unit can also give a good approximation of the value of the over-all heat transfer coefficient. Based on the logarithmic mean temperature difference (LMTD) the following expression can be used:

$$q = \dot{m}\, c_p \;\; \Delta T = U\; A\; F \;\; \Delta T_m \tag{44c}$$

where q = rate of heat transfer between the fluids (W)

$\dot{m}$ = mass flow rate (Kg/min)

ΔT = change in bulk temperature between entrance and exit of the tube

ΔT_m = LMTD (°C)

F = correction factor, dimensionless

The correction factor for the experimental heat exchanger was assumed to be similar to that of a multiple-pass counterflow heat exchanger. From charts (Holman, 1976) F was found equal to 0.95 and 0.99 for the heater and cooler respectively.

D. Performance Calculation

The over-all heat transfer coefficient estimation, for the heater and cooler was described in the previous section. Inlet and outlet temperatures of the fluids were recorded on the potentiometer. The fluid capacity rate was calculated by multiplying the flow rate by the specific heat of each fluid. The surface area through which heat transfer occurred was calculated based on the logarithmic mean area of the inner and outer areas of the coils.

The effectiveness, ε, of each section was determined by using equation (27), while equation (28) was applied for NTU calculation.

E. Degree of Inactivation Measurement and Calculation

Purified polygalacturonase was added to the untreated brine at the rate of 1 mg of enzyme per 1 ml of brine. The pasteurized brine was collected and the degree of inactivation was measured by determining the concentration of the remaining enzyme. The "cup-plate" procedure as used by Athanasopoulos (1976) was applied. To the cup of each Petri dish 0.15 ml of enzyme-brine solution was added and incubated for 20 hr at 35°C. The diameter of the clear zone was related to a standard curve and the amount of inactivation recorded. The standard curve was prepared by diluting 100, 80, 60, 40, and 20 mg of enzyme to 100 ml of Napoleon brine. This resulted in 100, 80, 60, 40, and 20% of enzyme activity. 0.15 ml of each dilution was added to Petri dishes containing the solidified agar. The dishes were placed in the incubator together with those containing the pasteurized brine. The diameter of the clear zone of each dilution was plotted against enzyme concentration on semi-log paper. The straight line obtained was regarded to be the standard curve (Figure 4).

The degree of inactivation was calculated by both, the General and the Analytical methods, as illustrated by the following example.

Example Problem

Napoleon cherry brine containing 1 mg polygalacturonase enzyme per 1 ml of brine was pasteurized in a helically coiled heat exchanger. Determine the amount of inactivation if the following information is available:

	Heater	Cooler
Area of coiled tube (m^2):	0.129	0.088
Length of coil (m):	5.8	3.96
Brine flow rate (Kg/min):	1	
U ($W/m^2 .\ ^\circ C$):	1085	1011
Amount of brine in contact with coil (Kg):	0.180	0.124
Time of heat exposure (sec):	10.83	7.45
Specific heat of brine:	4186.9 J/Kg.°C	
Constant in Arrhenius equation, A (sec^{-1}):	7.5×10^{39}	
Velocity constant of polygalacturonase at 76.66°C:	0.3057 sec^{-1}	
Activation energy of polygalacturonase:	64668 cal/mole	
A value of polygalacturonase	8.4°C	
Temperature-time profile in coils:	exponential	

A. Analytical Method.

According to equation (40), for both heating and cooling,

$$\nabla = A \int_0^t \exp(-Ea/RT)dt$$

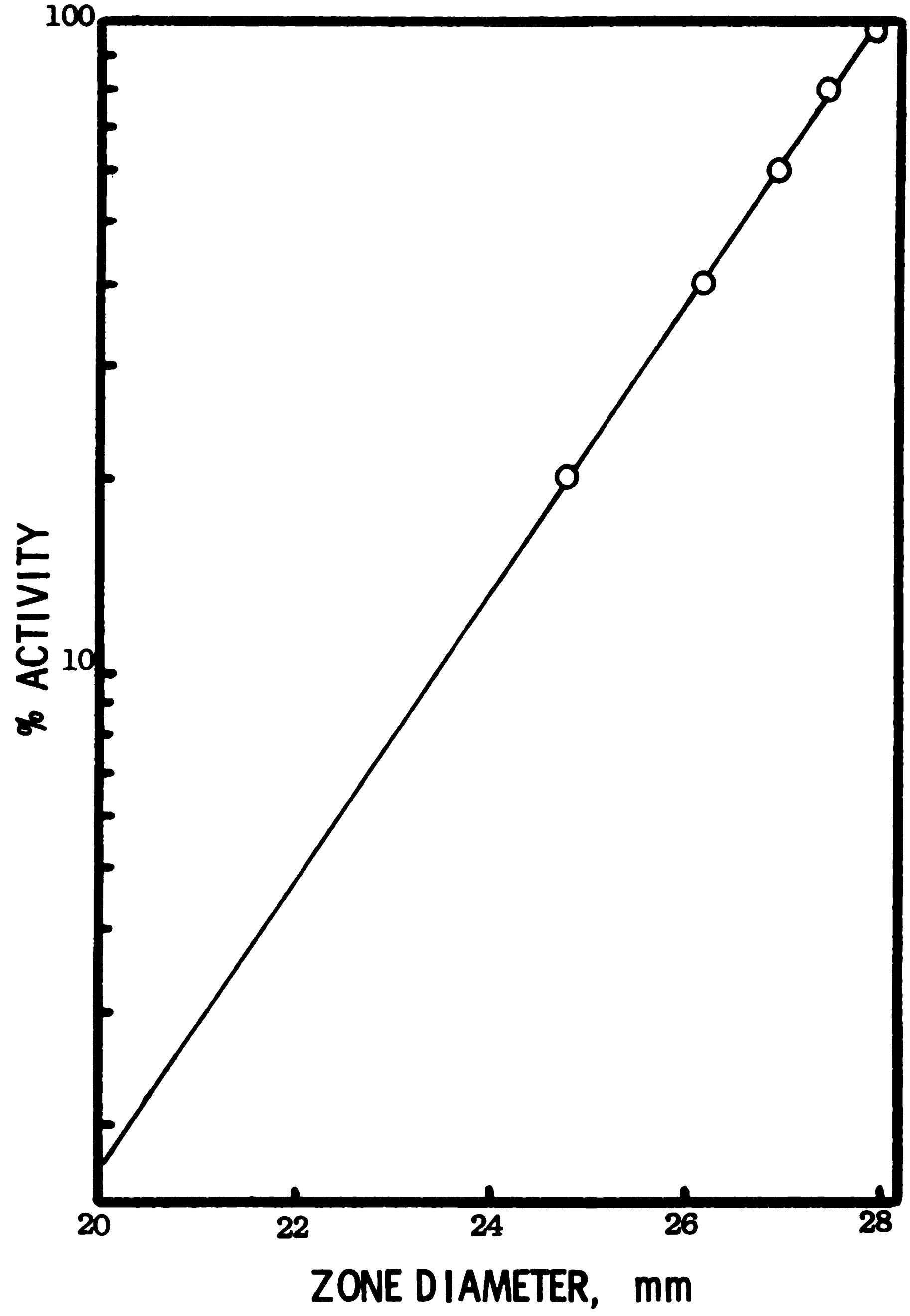

Figure 4. Standard curve for polygalacturonase activity measurement.

Since the temperature-time profiles are exponential, combination of equation (40) and (43) results in integrals which are first order exponential integrals that can be numerically evaluated for the various values of their lower limits or arguments. Thus for exponential temperature-time profiles in the heating and cooling sections we have:

$$\nabla = \frac{A}{K} \; E_1 \left(\frac{a}{1+b}\right) - E_1 \left(\frac{a}{1+b^{-kt}}\right) \tag{45}$$

$$- \frac{Ae^{-a}}{K} \; E_1\left(\frac{a}{1+b^{-a}}\right) - E_1\left(\frac{a}{1+be^{-kt}} - a\right)$$

By definition $E_1(z) = \int_z^{\infty} \frac{e^{-x}}{x} \, dx$ (46)

z = lower limit or argument

E_1= first order exponential integral function

For positive arguments up to 15, the first order exponential integral function can be taken from tables (Deindoerfer <u>et al</u>. 1959). Above this argument, the following approximation yields a satisfactory value:

$$E\ (z) = \frac{e^{-z}}{z} \tag{47}$$

To solve the problem equation (45) and (47) should be used.

Heating Section

J = heat source temperature = $\frac{93.9 + 77.5}{2}$ = 84.7°C or 358.9°K

$$a = \frac{64668}{1.987 \text{ X } 358.9} = 90.68$$

$$K = \frac{U\,A}{W\,c_p} = \frac{1085\ \text{W/m}^2.{}^\circ\text{C} \times 0.1294\ \text{m}^2 \times 1\text{J/W.sec}}{0.180\ \text{Kg} \times 4186.9\ \text{J/Kg.}{}^\circ\text{C}} = 0.186\ \text{sec}^{-1}$$

$T_o = 295.4°K$,

$$b = \frac{T_o - T_H}{T_H} = -0.1769 \qquad e^{-Kt} = 0.147$$

$$\frac{a}{1+b} = 110.16\ , \qquad \frac{a}{1+be^{-Kt}} = 93.10$$

$$\frac{a}{1+b} - a = 18.48\ , \qquad \frac{a}{1+be^{-Kt}} - a = 2.42$$

$$E_1(110.10) = \frac{e^{-110\cdot10}}{110.10} = 1.3 \times 10^{-50}$$

$E_1(93.10) = 3.96 \times 10^{-43}$, $E_1(18.48) = 5 \times 10^{-10}$

$E_1(2.42) = 3.67 \times 10^{-2}$

$$\frac{A}{K} = 4.03 \times 10^{40} \qquad \frac{Ae^{-a}}{K} = 16.73$$

Equation (45) becomes,

$$4.03 \times 10^{40}\ [1.30 \times 10^{-50} - 3.96 \times 10^{-43}]$$

$$- 16.73\ [5 \times 10^{-10} - 3.67 \times 10^{-2}] = -0.01595 + 0.61399$$

$$\nabla_{\text{heat.}} = 0.59804$$

Cooling

$J = T_c$ = cooling medium temperature $= \frac{13 + 22}{2} = 17.5°C = 290.7°K$

$$a = \frac{64668}{1.987 \text{ X } 290.7} = 111.95$$

$$K = \frac{1011 \text{ X } 0.0885 \text{ X } 1}{0.124 \text{ X } 4196.9} = 0.1723 \text{ sec}^{-1}$$

$T_o = 349.86°K$, $b = 0.2035$, $e^{-Kt} = 0.277$

$\frac{a}{1+b} = 93.02$, $\frac{a}{1+be^{-Kt}} = 105.97$

$\frac{a}{1+b} - a = -18.93$, $\frac{a}{1+be^{-Kt}} - a = 5.98$

$A/K = 4.35 \text{ X } 10^{40}$, $\frac{Ae^{-a}}{K} = 1.17 \text{ X } 10^{-8}$

$E_1(93.02) = 4.29 \text{ X } 10^{-43}$, $E_1(105.97) = 8.96 \text{ X } 10^{-49}$

$E_1(-18.93) = -8791114.3$, $E_1(-5.98) = -66.12$

Equation (45) becomes,

$$4.35 \text{ X } 10^{40} \ [4.298 \text{ X } 10^{-43} - 8.96 \text{ X } 10^{-49}]$$

$$-1.17 \text{ X } 10^{-8} \ [-8791114.3 + 66.12] = 0.01869 + 0.10285$$

$$\nabla_{cool.} = 0.12154$$

$\nabla_{hold.} = kt = 0.3057 \text{ sec}^{-1} \text{X } 9.02 \text{ sec} = 2.7574$

(the brine is exposed in the holding coil for 9.02 sec)

$$\nabla_{total} = \nabla_{heat.} + \nabla_{cool.} + \nabla_{hold.}$$

$$= 0.59804 + 0.12154 + 2.7574 = 3.47698$$

$$\nabla_{total} = \ln N_o/N\ , \quad (\text{assume } N_o = 100)$$

then, $100/N = 32.36$ and $N = 3.09\%$

or

$100 - 3.09 = 96.91\%$ inactivation

B. General Method

A basic requirement in process calculation by the general method is measurement of the temperature of the sterilized medium during heat treatment. In the present study the temperature of the treated brine was measured at several points. The temperature between these points was predicted by using equation (43). Therefore, the following method may be considered as a modification of the general method.

For the heating section equation (43) becomes,

$$T = 358.9\ (1-0.769\ e^{-0.186t})$$

t varies from 0 to 10.83 sec

For the cooling section equation (43) gives,

$$T = 290.7\ (1 + 0.2035\ e^{-0.1723t})$$

t varies from 0 to 7.47 sec

By giving t several values, the corresponding temperatures can be obtained. A plot of temperature versus time results in the temperature -time profile (Figure 14).

The first column of the following table represents the midpoint time for time intervals 1 sec. The second column is the temperature that corresponds at each time, and is taken from Figure 14. From equation (38)

$$L = 1/t = 10^{\frac{T - 250}{Z}} \qquad (F = 1 \text{ min})$$

In the problem the process temperature is 170 F(76.66°C) $\neq$ 250°F, therefore, $F_{170} \neq 1$ min. Thus the above equation will become

$$L = F/t = 10^{\frac{T - 76.66}{Z}}$$

Since the time intervals = 1 sec, the third column, which represents the time interval lethality

$$= L \times \Delta t = 1 \times 10^{\frac{T - 76.66}{Z}} \text{ sec}$$

Table 2. Calculation of cumulative lethality.

Midpoint Time sec	Midpoint Temp. °C	$10^{\frac{T - 76.66}{Z}}$
0.5	27.5	1 X 10^{-6}
1.5	37.8	2.3 X 10^{-5}
2.5	46.0	0.00022
3.5	52.5	0.00132
4.5	58.0	0.0060
5.5	62.5	0.020619
6.5	66.4	0.06005
7.5	69.5	0.14048
8.5	72.5	0.31971
9.5	75.0	0.63442
10.5	76.5	0.95708
11.5	76.66	1.0
12.5	76.66	1.0
13.5	76.66	1.0
14.5	76.66	1.0
15.5	76.66	1.0
16.5	76.66	1.0
17.5	76.66	1.0
18.5	76.66	1.0
19.5	76.66	1.0
20.5	70.0	0.16117
21.5	61.6	0.01611
22.5	54.5	0.00230
23.5	48.7	0.00046
24.5	43.6	0.00011
25.5	36.2	0.00001
26.5	35.5	-
	Total =	11.3200 sec

Cumulative lethality = 11.3200

The cumulative lethality represents the equivalent process time at temperature 76.66°C. Since $k = 0.3057\ \mathrm{sec}^{-1}$, equation (30) becomes,

$$\ln C/Co = - kt = -0.3057\ \mathrm{sec}^{-1} \times 11{,}3200\ \mathrm{sec}$$

If Co = 100, then C = 3.14%

Thus, the process resulted in

100 - 3.14 = 96.86% inactivation.

RESULTS AND DISCUSSION

A. Pressure Drop

The frictional losses in a pipe are highly dependent on the fluid films that exist on the metal walls. Since the fluid flow pattern (laminar or turbulent) effects these boundary layers, the degree of turbulence determines the amount of pressure drop through a pipe. Figure 5 illustrates the effect of Reynolds number on the friction factors in laminar and turbulent flow, for straight and coiled pipes. The friction factors are affected to a greater extent by the Re in the laminar flow conditions than in turbulent flow. This holds for both straight and coiled pipes. The fact that, in the laminar flow the friction factor is inversely proportional to Re (equation 2) while in the turbulent region it is inversely proportional to $Re^{0.2}$ ($f_s = 0.046/Re^{0.2}$), which explains the previously mentioned effect on the friction factors. Therefore, with an increasing Re, the friction factors fall more gradually in the turbulent flow than in the laminar flow region.

The centrifugal force which exists in a curved pipe produces a pressure gradient in a cross section. This pressure gradient yields a pair of secondary flows which cause a larger amount of pressure drop in coiled pipes than in straight tubes. The curvature ratio (di/D_h) may be considered as an expression of the magnitude of the centrifugal force. The coil diameter is the factor that determines the value of this force

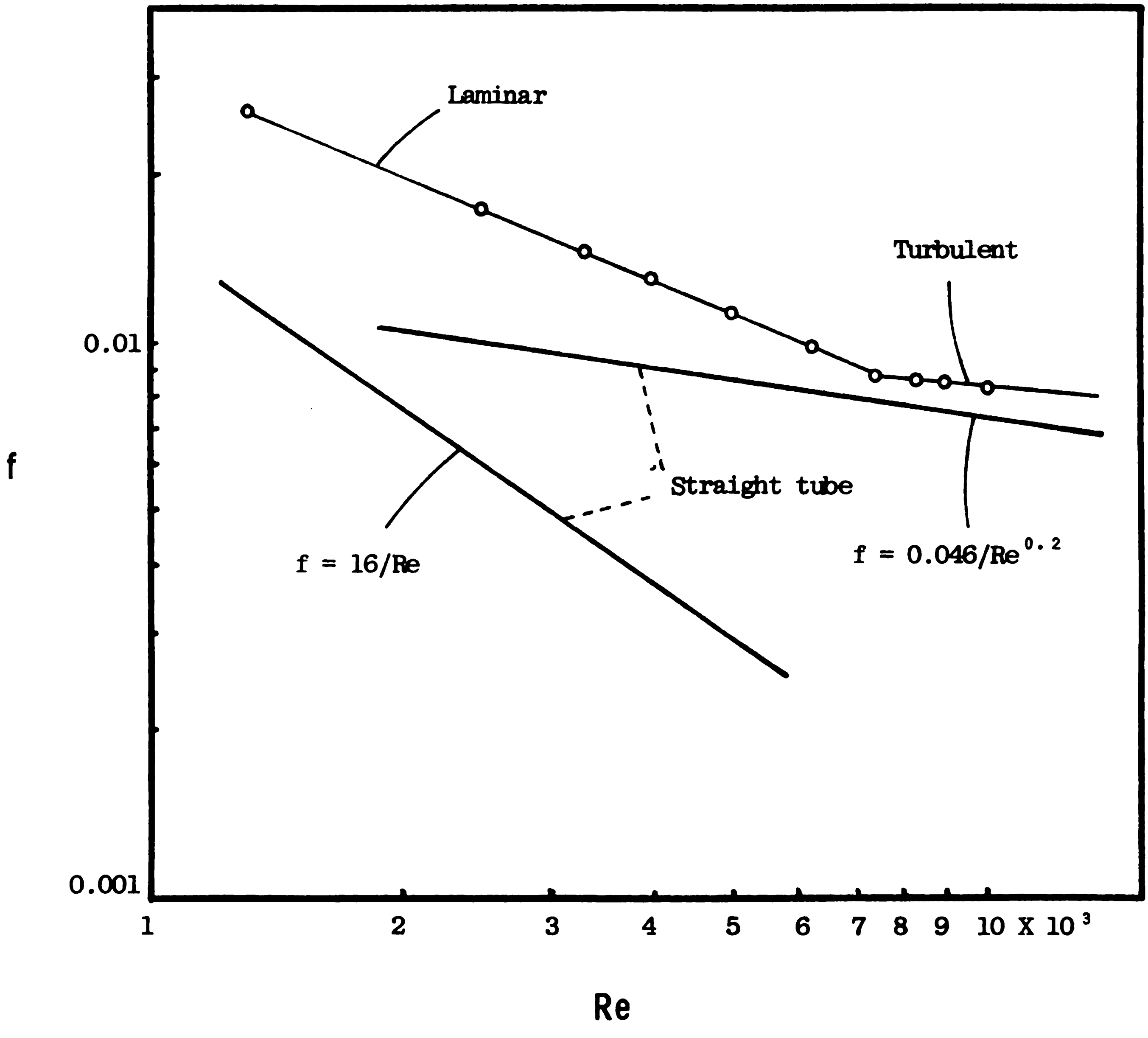

Figure 5. Effect of Reynolds number on friction factor in straight and coiled tubes for laminar and turbulent flow.

for a certain inside tube diameter. Small coil diameters contribute to a larger centrifugal force and larger pressure gradient. Figure 6 shows the variation in the friction factors among coils with different diameters. Under a constant Re, smaller coil diameters result in friction factors of higher values.

Probably the most important factor that influences the pressure drop in a tube is the flow rate. The relationship between frictional pressure losses and flow rate is shown in Figure 7. The pressure losses increase gradually with small flow rates (low fluid velocity), whereas the pressure drop increases rapidly at high flow rates (high fluid velocities). Such a relationship may be expected since, according to equation (1), the pressure drop is directly proportional to the square of the fluid velocity. Table 3 and Figure 7 give a comparison between the calculated and the measured pressure drop. The calculated values were found to deviate from -1.3 to +9.2% from the measured ones. Actually, equation (9) was developed for isothermal conditions. The use of mean film temperature may give a good approximation.

The shell-side pressure drop was measured. Seven trials gave values from 689 N/m^2 (0.1 psi) to 1034 N/m^2 (0.15 psi). The small velocities that were encountered in the shell-side in both sections contributed to the lower pressure drop. Therefore, it may be concluded that under the conditions carried out in the experiment the shell-side pressure drop is of no significance.

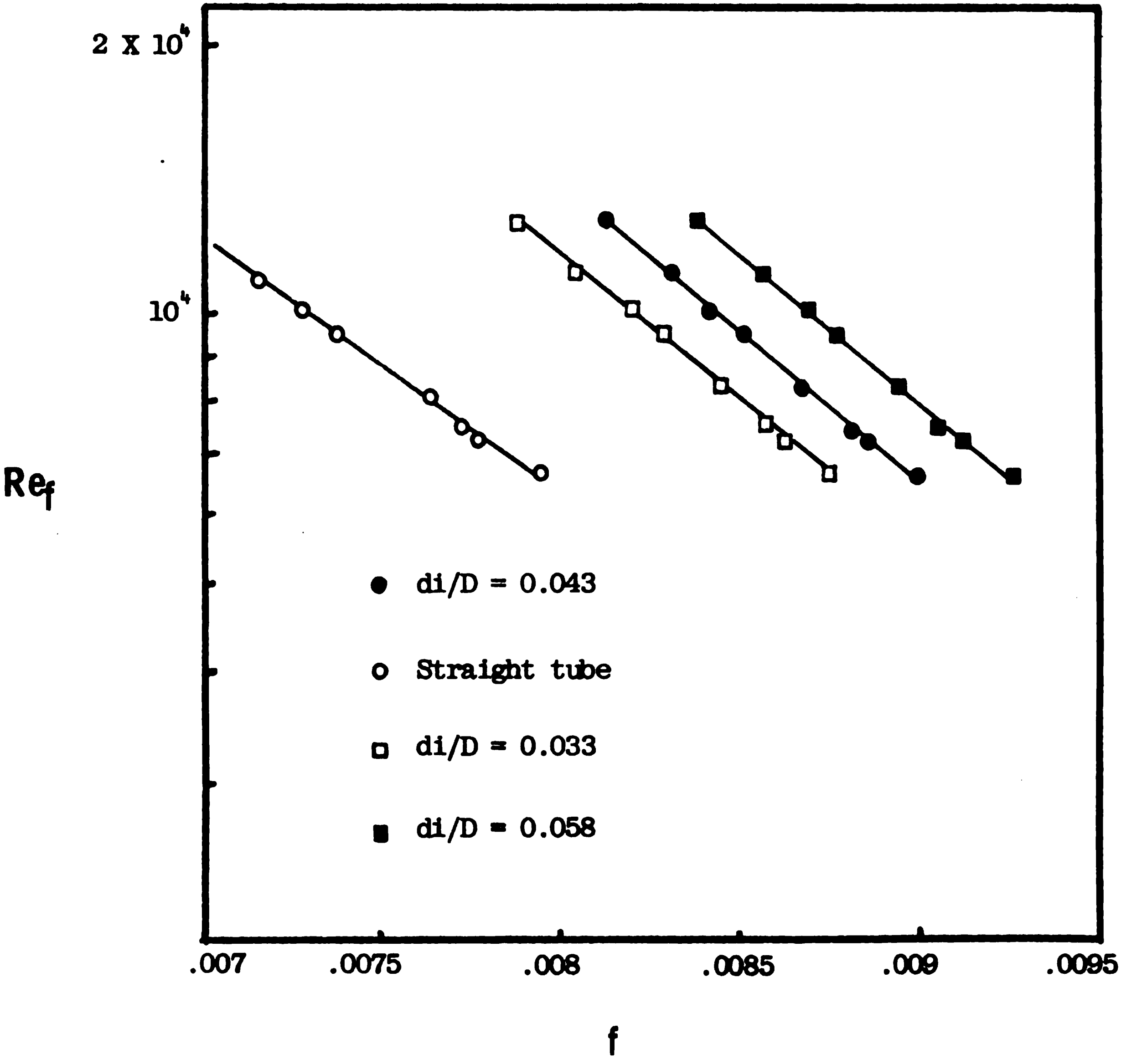

Figure 6. Effect of curvature ratio on frictional factors in coils.

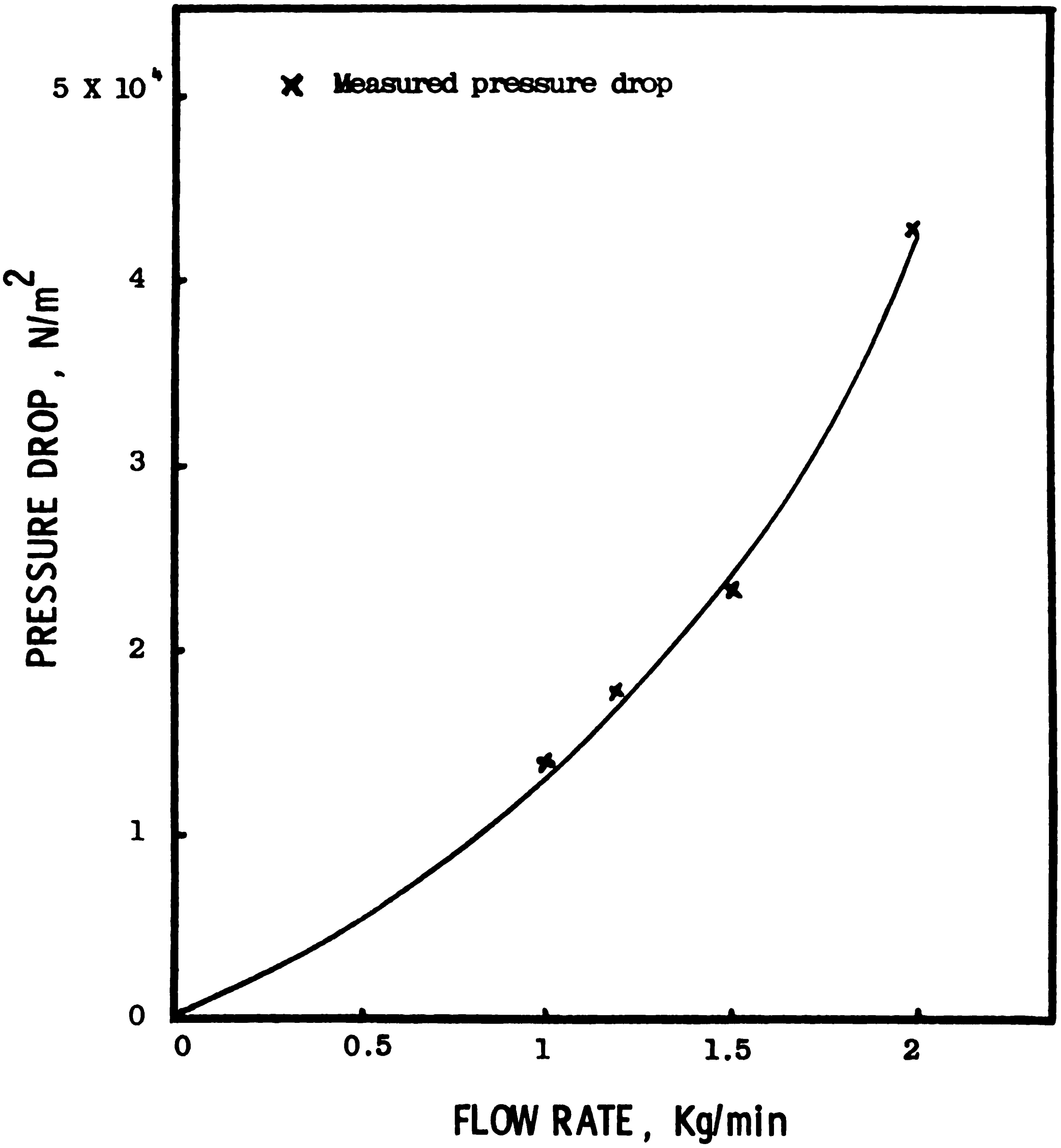

Figure 7. Effect of flow rate on calculated total pressure drop in the coils of the pasteurization unit.

Table 3. Calculated and measured total internal pressure drop in the pasteurization unit, at several flow rates.

Flow rate (Kg/min)	Calculated $\Delta P(N/m^2)$	Measured $\Delta P(N/m^2)$	Difference %
1.0	12769	13858	-7.85
1.2	16915	17161	-1.43
1.5	24483	22414	+9.23
2.0	42196	42754	-1.30

B. Heat Transfer

The influence of the curvature ratio on the inside heat transfer coefficient is similar to that of the pressure drop described previously. The straight lines in Figure 8 are derived from equation (17) using different di/D_h values. The centrifugal force promotes mixing in the coiled pipe and this results in higher heat transfer coefficients in coils than in straight tubes. Figure 8 also shows how high Re contributes to high heat transfer coefficients as a result of increasing turbulence with increasing Re. Coils with the same tube diameter but different coil diameter will give different heat transfer coefficients under the same conditions. Those with a smaller diameter give the higher values. When the influence of the coil diameter on the centrifugal force and the effect of coil diameter on the heat transfer coefficients are considered, the values given in Figure 8 are valid and reasonable. Also illustrated in Figure 8 the higher Nu of coils when compared with those in straight tubes.

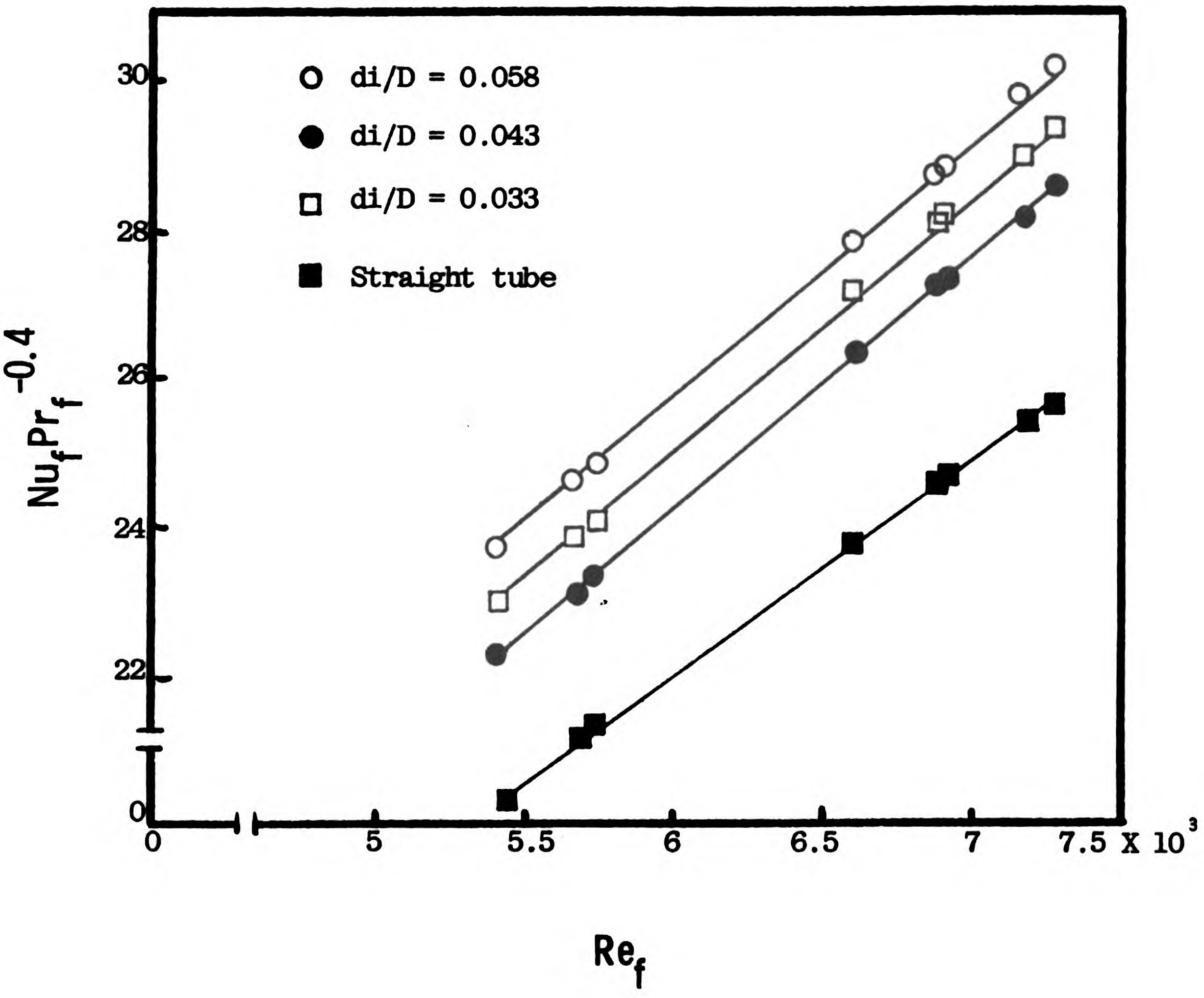

Figure 8. Inside heat transfer coefficients in coils for different Re and different curvature ratio.

Table 4. Comparison of calculated and predicted shell-side heat transfer coefficients ($W/m^2 . °C$), under different temperature conditions.

TEMPERATURE OF FLUIDS °C				HEAT TRANSFER COEFF.		VARIATION %
T_{bi}	T_{bo}	T_{wi}	T_{wo}	Predicted	Calculated	---
23.3	70.0	87.8	72.2	1505	1629	+8.2
69.7	33.3	11.7	25.6	1374	1499	+9.2
22.2	72.0	86.1	74.4	1686	1635	-3.0
77.7	32.2	13.0	19.5	1491	1544	+3.5
22.2	76.7	93.9	77.5	1720	1675	-2.5
76.1	34.0	13.0	22.0	1536	1567	+1.8
22.2	81.0	98.9	79.4	1902	1680	-11.6
80.5	34.4	11.1	19.2	1620	1556	-3.9

T_{bi} = Brine inlet temperature

T_{wi} = Water inlet temperature

T_{bo} = Brine outlet temperature

T_{wo} = Water outlet temperature

The Grashof and the Reynolds number of the shell-side varied from 5×10^6 to 1.2×10^7 and 50 to 70 respectively. This indicates that natural convection was the predominant force in the heat transfer mechanism in the shell-side. Outside heat transfer coefficients were calculated by equation (24). They were also predicted through equations (44a) and (44b) for heating and cooling respectively. In Table 4 the difference between the predicted and calculated values is illustrated. They were found to vary from -3.0 to +11.6%. Actually, in the cooler, natural convection did not seem to exist because the water was flowing upward while the density was increasing downward. The assumption that natural convection equations can be applied to the shell-side of the cooler due to the low velocities of the water (1m/min), seems to be correct as illustrated in Table 4. The outside heat transfer coefficients were found to be smaller than those in the inside. The lower inlet to outlet velocities in the shell-side than those inside the coils may have contributed to the lower heat transfer coefficients.

The most important heat transfer parameter in designing a heat exchanger is the over-all heat transfer coefficient. Equations (25), (44c), and (44a) or (44b) for heating and cooling respectively, were used to calculate the over-all heat transfer coefficient. In Table 5 the results of the three approaches are presented. The three methods were found to be in satisfactory agreement. Figure 9 and 10 illustrate the relationship between over-all heat transfer coefficients and the logarithmic temperature difference ratio. The straight line in Figure 9 is the solution of equation (44a) and in Figure 10 the solution of equation (44b). Both figures indicate that for constant heating or cooling medium temperatures the difference between inlet and outlet

Table 5. The over-all heat transfer coefficient (W/m^2. °C), calculated by three different methods under several temperature conditions.

TEMPERATURE OF FLUIDS °C				OVER-ALL HEAT TRANSFER COEFFICIENT		
T_{bi}	T_{bo}	T_{wi}	T_{wo}	Equation (44a) or (44b)	Equation (25)	Energy Balance
23.3	70.0	87.8	72.2	1005	1056	965
69.7	33.3	11.7	25.6	923	977	902
22.2	72.0	86.1	74.4	1084	1062	993
77.7	32.2	13.0	19.5	977	994	943
22.2	76.7	93.9	77.5	1107	1085	1022
76.1	34.0	13.0	22.0	1002	1011	936
22.2	81.0	98.9	79.4	1187	1096	1078
80.5	34.4	11.1	19.2	1039	1016	1009

T_{bi} = Brine inlet temperature

T_{bo} = Brine outlet temperature

T_{wi} = Water inlet temperature

T_{wo} = Water outlet temperature

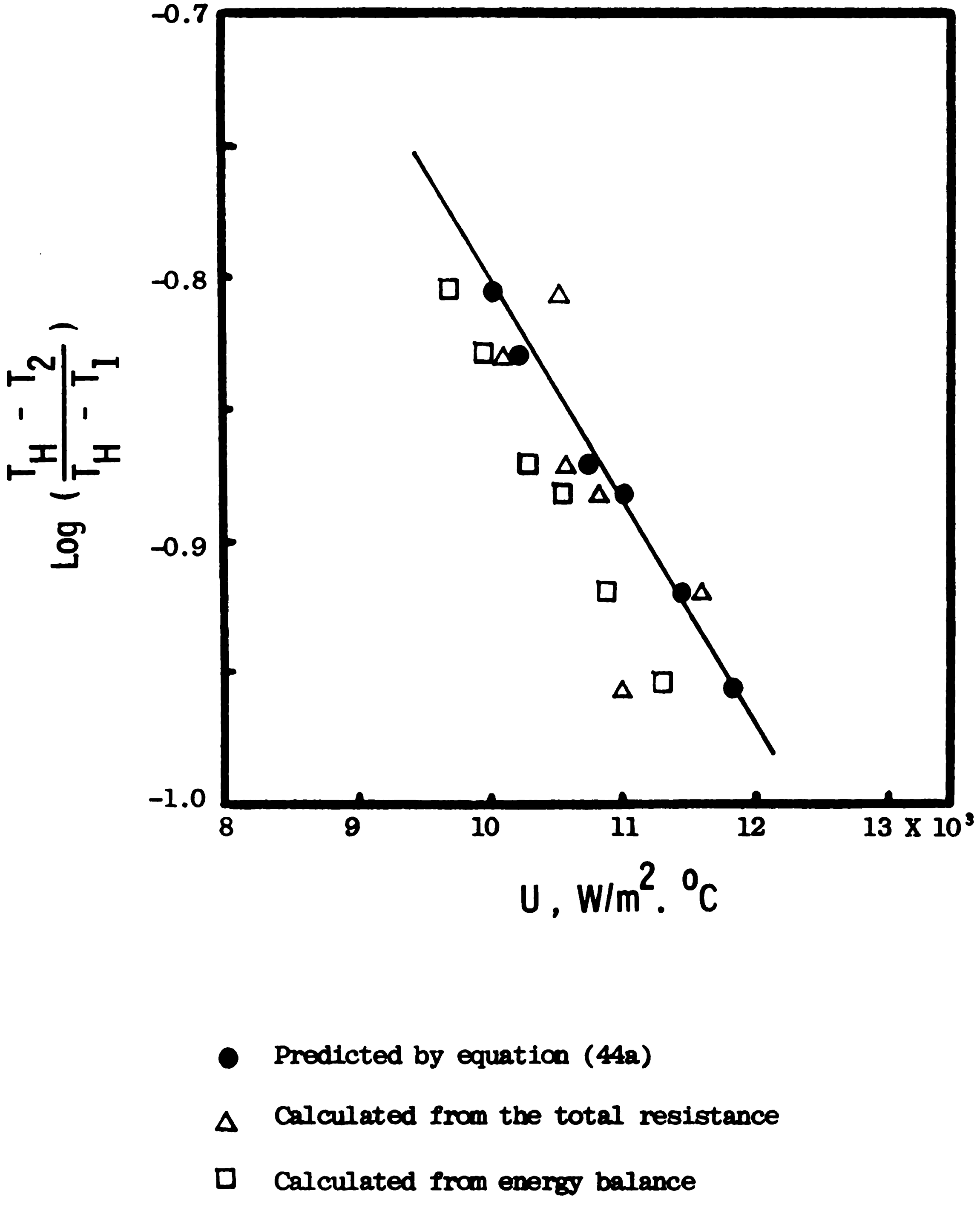

Figure 9. Effect of temperature difference ratio of the two fluids in the heater, on the over-all heat transfer coefficient.

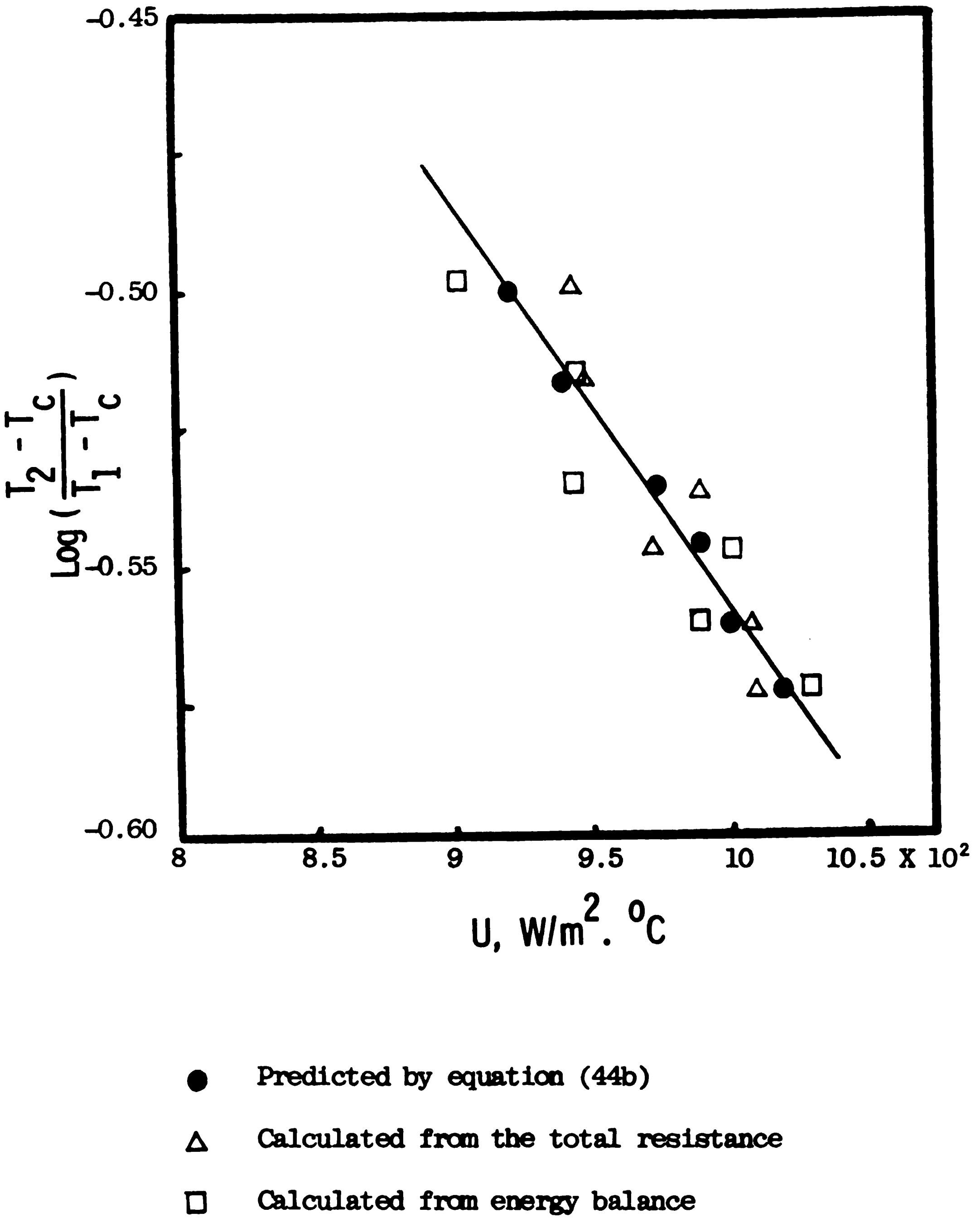

Figure 10. Effect of temperature difference ratio of the fluids in the cooler on the over-all heat transfer coefficient.

temperature of the fluid in the coils determines the magnitude of the over-all heat transfer coefficient. Thus, large temperature differences contribute to high U values, while small temperature differences result in small U values.

C. Performance of the Experimental Heat Exchanger

Figures 11 and 12 illustrate the temperature distribution of the fluids in the heating and cooling section. The brine inlet and outlet temperature was measured, while the temperature change with time was predicted by equation (43). The distribution is very similar to that of a double pipe counterflow heat exchanger. Therefore, the LMTD of the experimental heat exchanger may be calculated in a manner similar to that of a double pipe heat exchanger. Since the LMTD can be calculated the over-all heat transfer coefficient can be calculated by using equation (44c).

The difference between inlet and outlet temperatures of the hot and cold water is effected by the flow rate of each stream. The larger the flow rate the smaller the temperature difference. The drop in the hot water curve is larger than the rise of the curve of the cold water as shown in Figures 11 and 12. Different flow rates (3.8 Kg/min for the hot water and 5.6 Kg/min for the cold water) account for the variation.

Table 6 indicates that the effectiveness of the experimental heat exchanger varied from 73 to 78% for the heating section and 63 to 68% for the cooling section. The lower effectiveness of the cooling section may be due to the lower over-all heat transfer coefficients of the cooler. Generally the temperature for both fluids was lower than that in the heater.

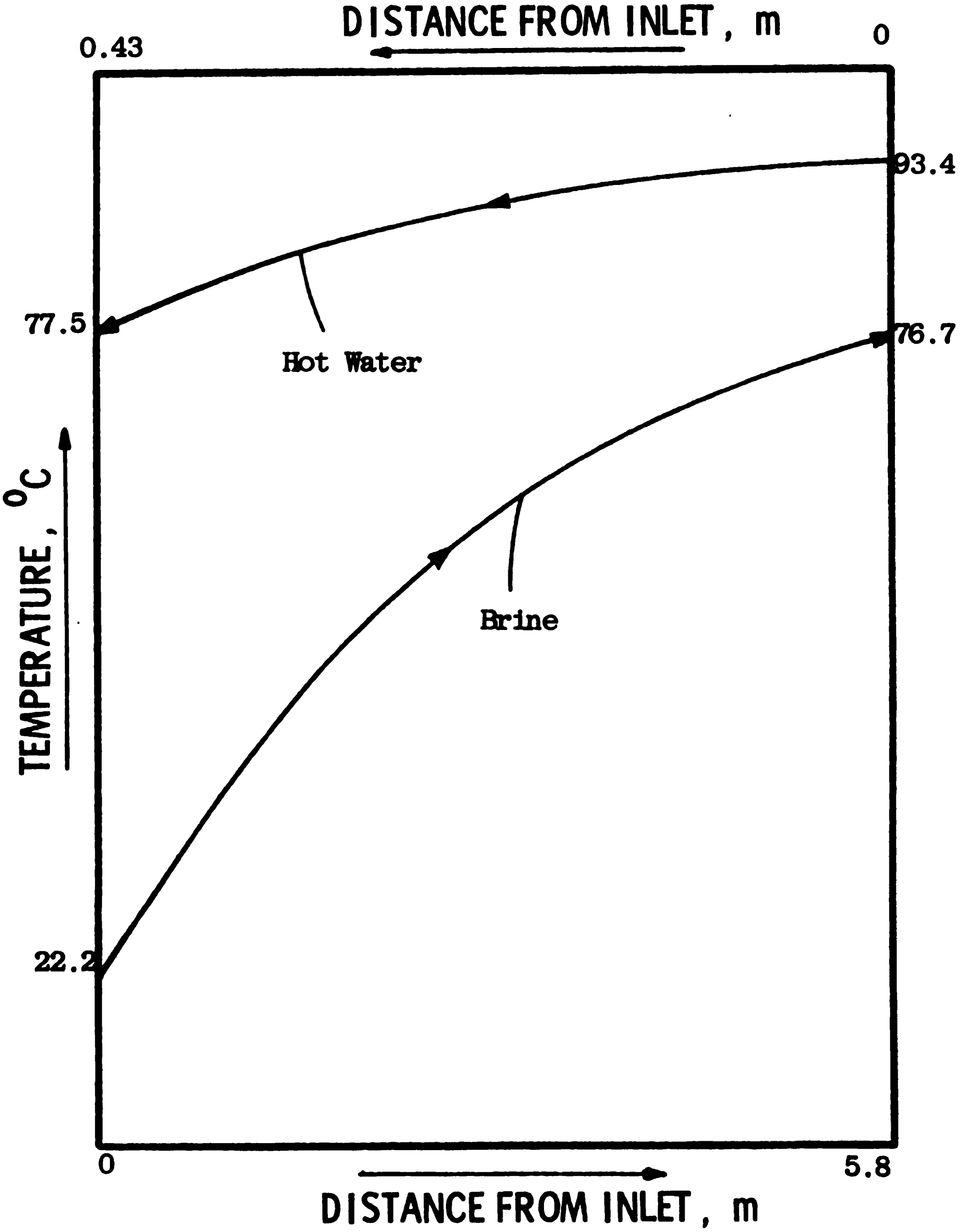

Figure 11. Temperature profiles in the heater of the experimental heat exchanger.

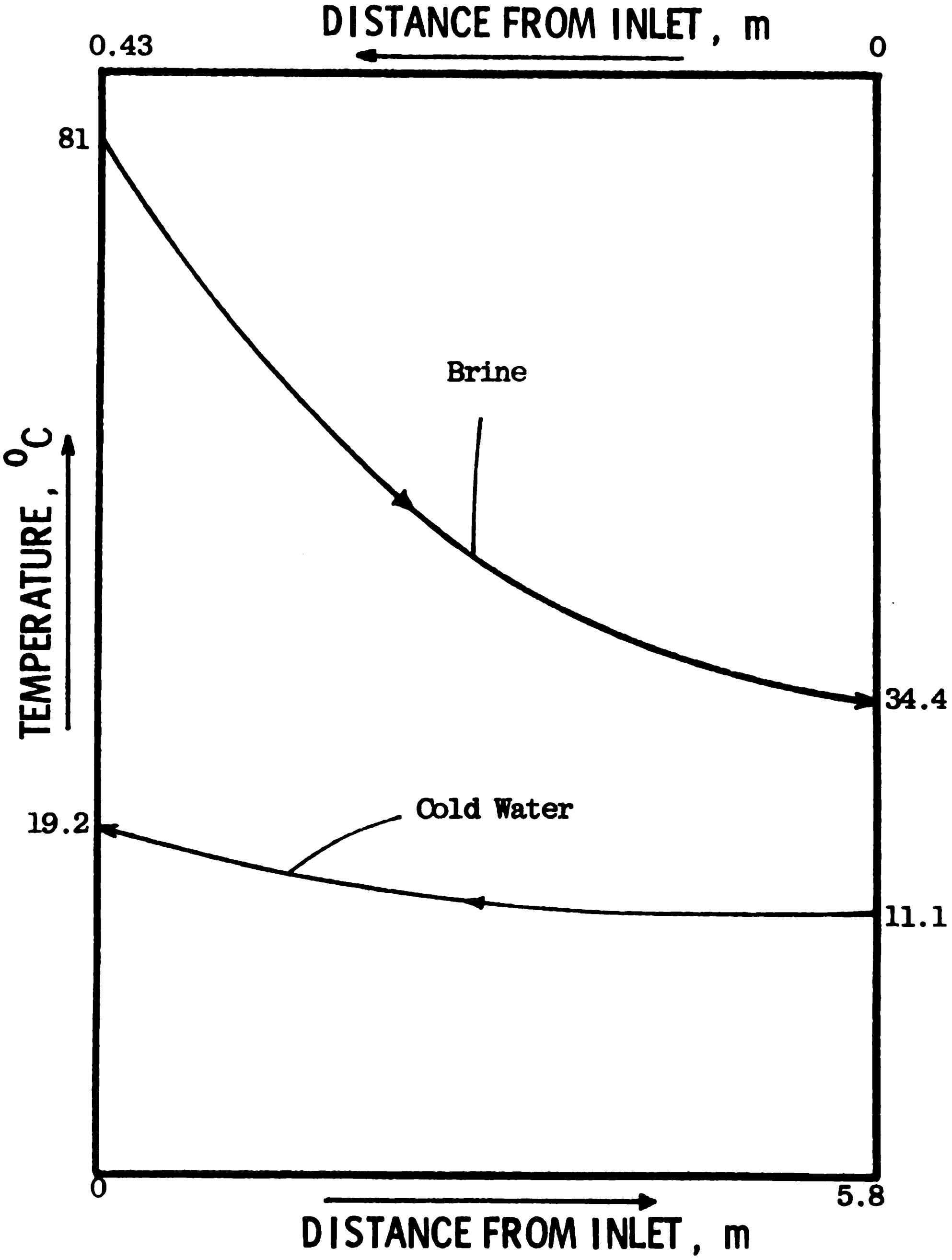

Figure 12. Temperature profiles in the cooler of the experimental heat exchanger.

Table 6. Effectiveness and NTU of the experimental heat exchanger for pasteurizing cherry brine at a flow rate of 1 Kg/min at several process temperatures.

Process Temperature °C	Effectiveness, % Heater	Effectiveness, % Cooler	NTU Heater	NTU Cooler
70.0	73	63	1.96	1.26
72.0	78	67	1.97	1.26
76.6	77	67	2.01	1.28
81.0	77	68	2.03	1.29

The NTU of the heat exchanger is a function of the over-all heat transfer coefficient and the heat transfer area (equation 28). Higher NTU values also occurred in the heater as shown in Table 6.

D. Polygalacturonase Heat Treatment

The D values of an organism increases very rapidly with small increases in temperature. An increase in temperature also results in a sharp decrease in the velocity constant, k, of a reaction. Therefore, the percentage of inactivation is highly temperature dependent. By increasing the temperature from 70 to 72°C the amount of inactivation increases from 44 to 63%, as shown in Table 7. The effect of temperature on the degree of inactivation is illustrated in Figure 13. At lower temperatures the increase is relatively sharp and asymptotically approaches the value of 100% inactivation. The temperature-time profiles in Figure 14, as predicted by equation (43), were used for calculating the inactivation at several process temperatures or constant temperatures along the holding coil. Because of the exponential nature of the heating and cooling portions, the enzyme was subjected to higher temperatures for a longer period in the heater than in the cooler for the same range of temperature. Thus, the heating section contributed more to the total inactivation than the cooling section, as in Table 8 is shown. The three methods which were used for the inactivation determination, namely plating, analytical, and general, gave results in close agreement with each other, as shown in Table 7. If it is assumed that plating is the more accurate method, the analytical and the general were found to vary from the plating values from -1.9 to +2.34% (Table 7). Therefore, use of either the general or

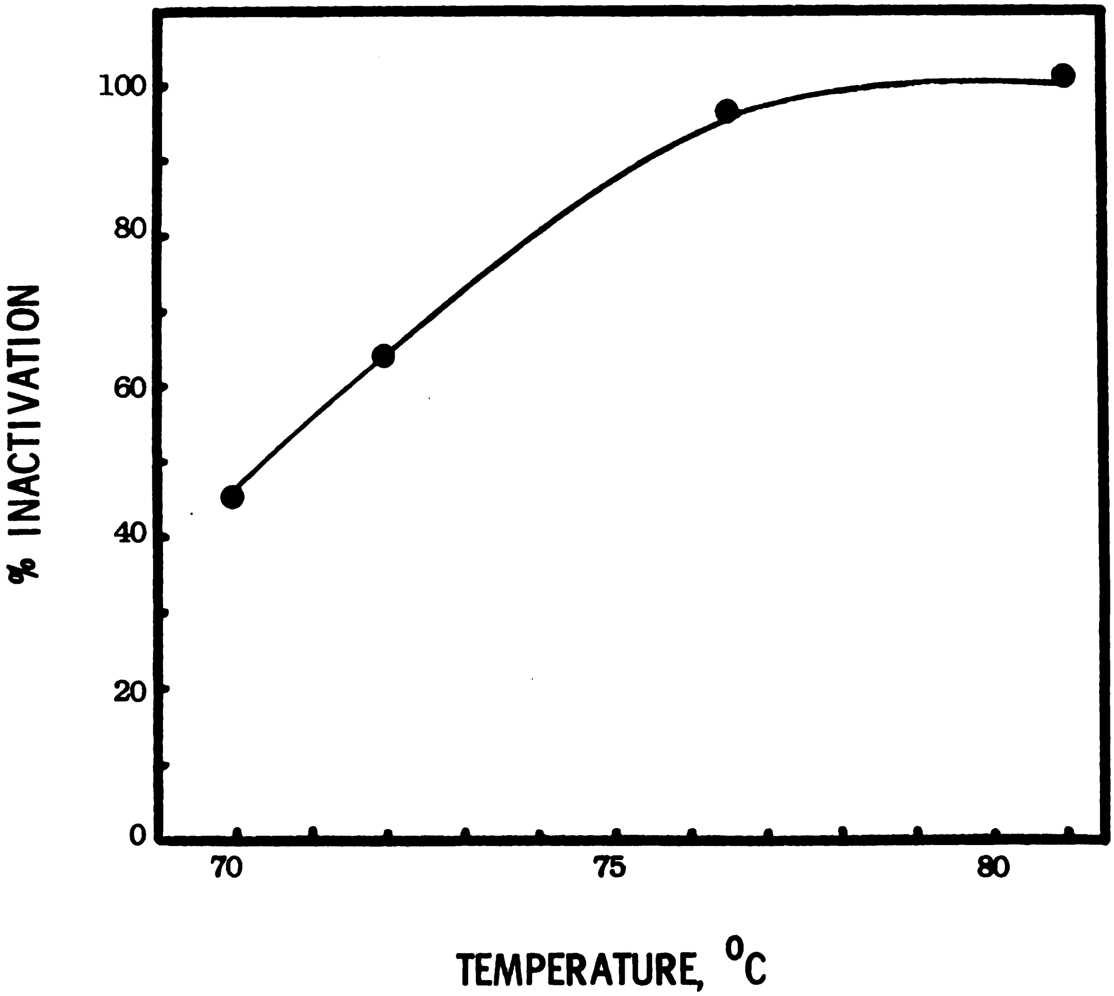

Figure 13. Effect of temperature on the degree of enzyme inactivation.

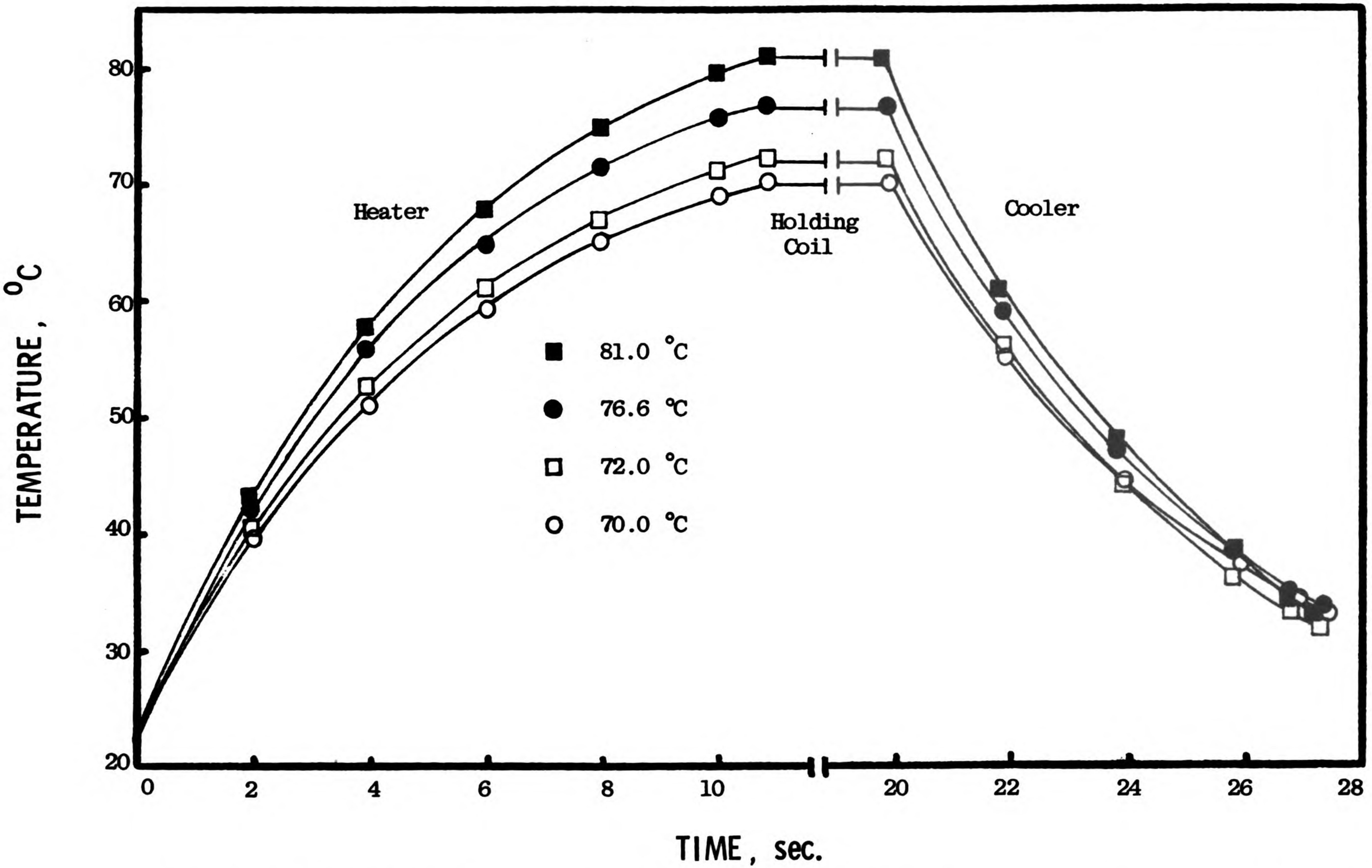

Figure 14. Predicted time temperature profiles at four constant temperatures along the holding coil.

Table 7. Percent inactivation determined by the three methods, and percent deviation of the General and Analytical methods from Plating.

TEMP.	% INACTIVATION			% DEVIATION	
°C	Plating	Analytical	General	Analytical	General
70.00	44.00	45.03	43.16	+2.34	-1.9
72.00	63.00	64.23	62.00	+1.92	-1.58
76.66	95.40	96.91	96.86	+1.58	+1.53
81.00	99.985	99.998	99.997	+0.013	+0.012

Table 8. Polygalacturonase inactivation occurring in each section of the experimental heat exchanger, calculated by the analytical method, using a constant temperature along the holding coil.

TEMP. °C	TOTAL Percent	HEATER Percent	COOLER Percent	HOLDING COIL Percent
70.0	45.00(100)	9.84(22)	1.44(3.2)	33.7(74.8)[1]
72.0	64.23(100)	14.14(22)	1.83(2.8)	48.22(75.2)
76.7	96.16(100)	16.53(17.2)	3.36(3.5)	76.24(79.3)
81.0	99.998(100)	18.01(18)	2.16(2.2)	79.81(79.8)

[1] The numbers in parentheses represent the percentage of the total percent.

the analytical method for determining the polygalacturonase inactivation in the pasteurization unit may give a satisfactory approximation. The analytical may be more advantageous if adequate heat transfer data is available, since it is faster than the general method. It should also be noted, that the variation becomes smaller as the degree of inactivation increases. A temperature of 81°C along the holding coil resulted in a 4.7D process, which may be considered adequate to prevent softening of the cherries. Such an inactivation is difficult to determine by plating if the diameter of the collar is higher than 1 cm, because the diameter of the clear zone that the enzyme creates, is less than 1.2 cm. Therefore, collars with a very small diameter must be used.

In general inactivation in the holding coil is not difficult to predict, because of the isothermal nature of the process. Application of equation (41) under isothermal conditions gives the degree of inactivation. The case is more complicated where temperature varies with time. At temperatures of 76.66 and 81°C the degree of inactivation was as follows:

Holding coil:	76.4 and 79.16%
Heating and cooling section:	19.89 and 20.16%

In the experimental heat exchanger treating cherry brine at a rate of 1 Kg/min, an approximate degree of inactivation at a temperature range between 76.66 - 81°C, may be determined by,

(a) Calculating the degree of inactivation in the holding section.

(b) Adding 20% to the above value.

The total inactivation, calculated in (b) will be ± 0.16% from the actual inactivation.

In order to treat an enzyme, other than polygalacturonase, the rate constants must be known. Suparath (1976) found that pectinase from Penicillium janthinellum, which causes softening in brined pickles, is the most heat resistant enzyme in cucumber brine. From Suparath's data the k and Z values of pectinase from P. janthinellum were calculated. They were compared with those of polygalacturonase, which were determined by Athanasopoulos (1976). The Z value of the pickle pectinase was found to be 4.592°C. Polygalacturonase has a Z value equal to 8.4°C. Figure 15 illustrates the thermal resistance curves of the two enzymes. Because pectinase from P. janthinellum has smaller Z value than polygalacturonase, it is more sensitive to temperature changes, as illustrated in Table 9. At low temperatures the pickle enzyme is more heat stable than polygalacturonase. At temperatures higher than 80°C, polygalacturonase is considered more heat resistant. The Arrhenius plots of the two enzymes are presented in Figure 16. In Figure 16 the sharper increase or decrease of the k values of the pickle pectinase can be seen.

Suparath's data were inadequate because the D values for the pickle pectinase were determined for two temperatures only. In the comparison between polygalacturonase and the pickle enzyme, it was assumed that the two D values determined by Suparath are points on the straight line shown in Figure 15. If the assumption is correct, then the experimental heat exchanger used for polygalacturonase inactivation may be used for pickle brine pasteurization since at temperatures above 80°C polygalacturonase has higher D and lower k values (Table 9). The composition of the pickle brine should also be considered. Usually the soluble solids of the pickle brine is around 12% which is very close to the cherry brine as indicated in Table A-1.

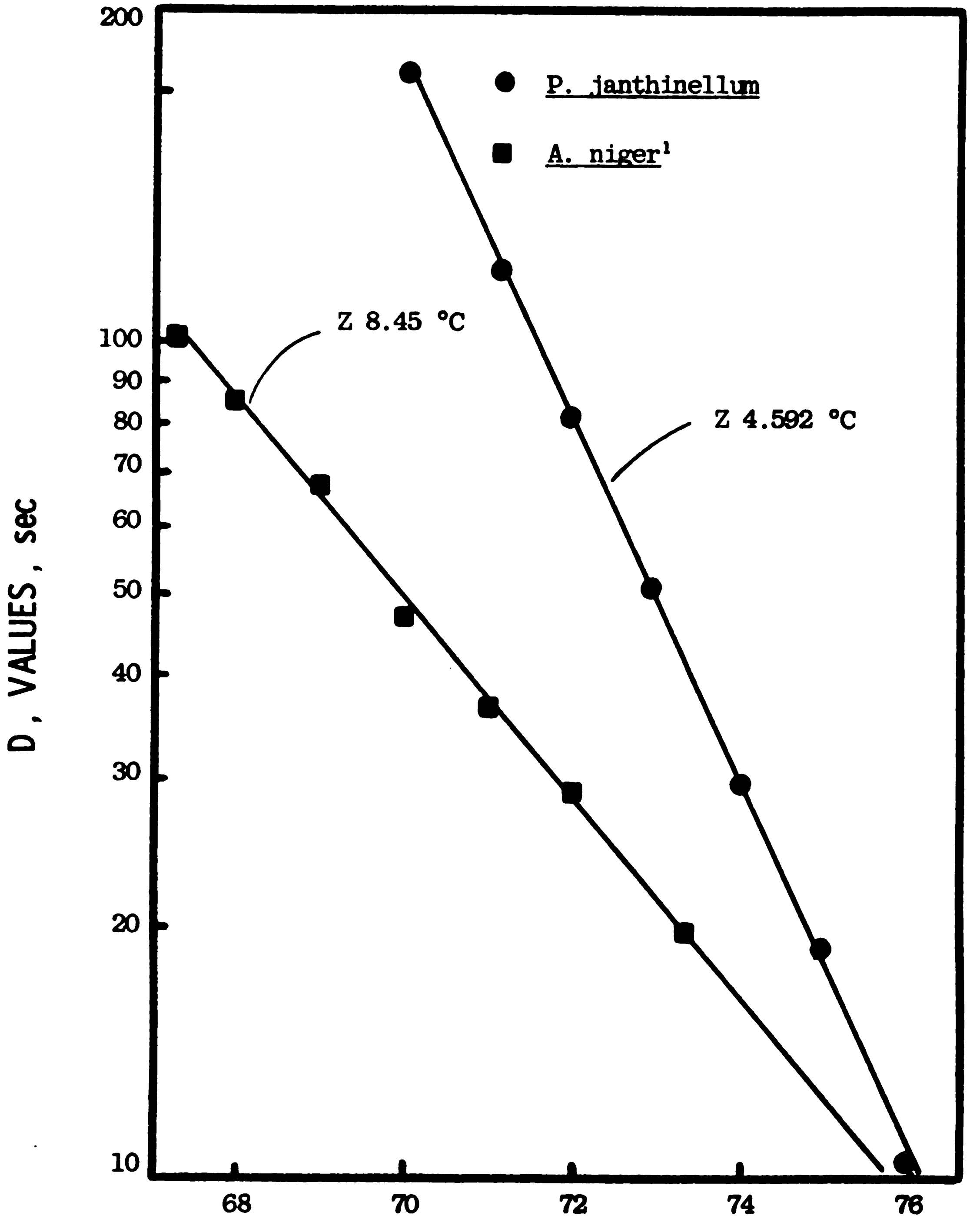

Figure 15. Thermal resistance curves for pectinase from *Aspergillus niger* (PG) at pH 3.0 and for pectinase from *Penicillium janthinellum* at pH 3.3

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[1] From Athanasopoulos (1976)

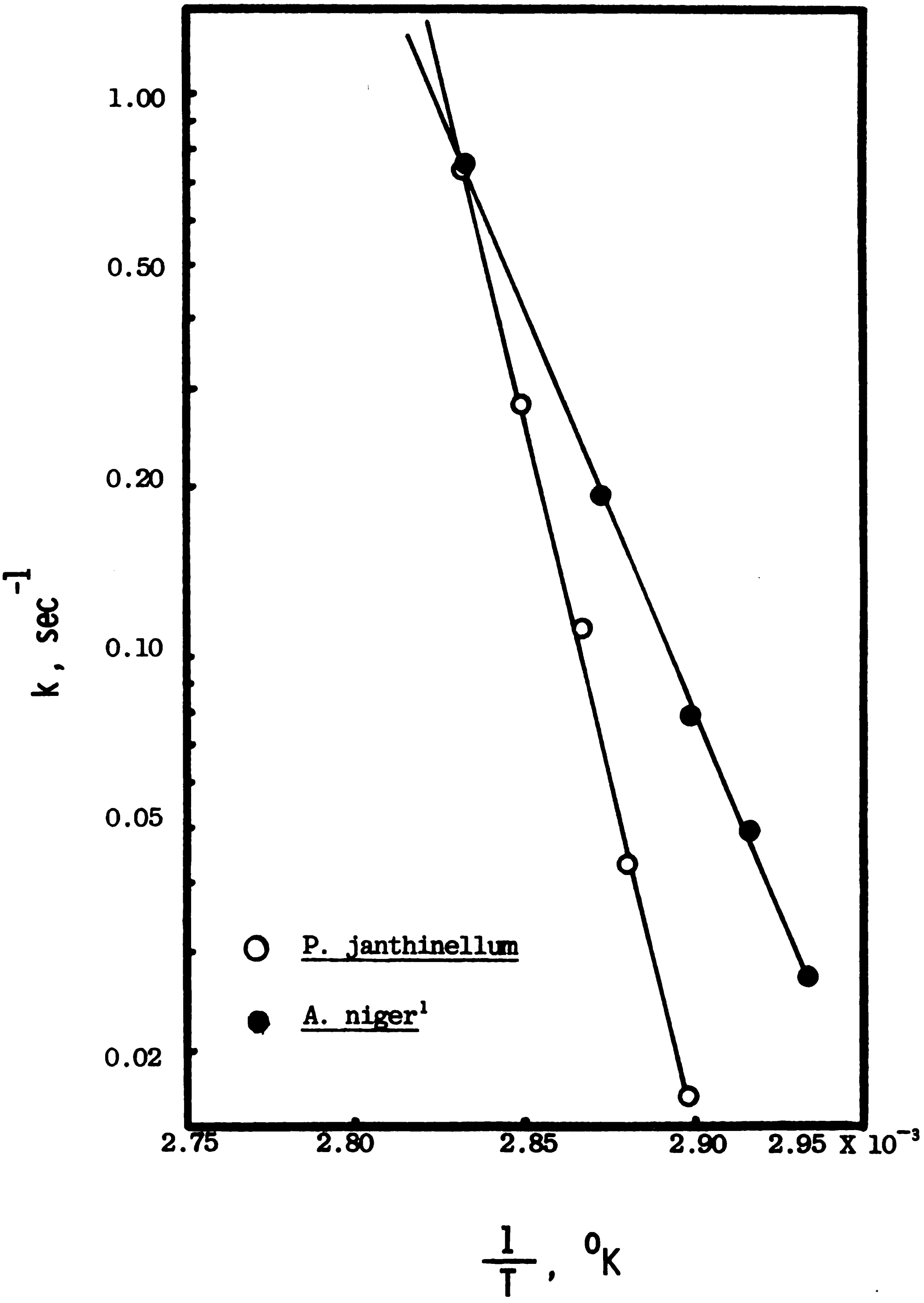

Figure 16. Arrhenius plot for polygalacturonase at pH 3.0 and pectinase from P. janthinellum at 3.7.

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[1] From Athanasopoulos (1976)

Table 9. Rate constants for polygalacturonase at pH = 3.0[1] and pectinase from *P. janthinellum* at pH = 3.7[2].

TEMP. °C	*P. janthinellum* D, VALUE, sec	*P. janthinellum* k x 100, sec^{-1}	Polygalacturonase D, VALUE, sec	Polygalacturonase k x 100 sec^{-1}
70	350.90	0.656	46.76	4.925
72	136.30	1.689	29.26	7.870
74	52.94	4.356	15.62	14.743
76	20.56	11.201	9.02	25.532
78	8.00	28.787	5.20	44.288
80	3.10	74.290	3.01	76.511

[1] From Athanasopoulos (1976)

[2] From Suparath (1976)

E. Operational Cost of the Pasteurization Process

To evaluate the economics of the pasteurization of cherry brine, several factors must be considered including:

Labor
Fuel
Pumping
and cold water needs.

Because the heat exchanger was designed to be in line with the reclamation system, labor cost for the heat treatment itself is difficult to evaluate. Thus, in the following calculations labor will be excluded.

The operational cost will be based on the following assumptions:

Amount of brine for pasteurization: 1000 gallons
Degree of inactivation desired: 99.985%
Flow rate of brine: 1 Kg/min
Hot water flow rate: 2.7 Kg/min
Cold water flow rate: 5.6 Kg/min
Temperature drop in the heater: 40°F or 22.2°C
Cost of fuel: \$2.26 per 1,000,000 BTU[1]
Cost of the water: \$2.50 per 1000 ft^3[2]

The system includes three pumps with 20 ft or 6 m total length of tubing.

Under these conditions, heating of the water requires

14016 BTU/hr

which will cost

14016 BTU X 63 hr X \$2.26/1,000,000 BTU = \$1.99

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[1] From Rippen (1977)

[2] From Rippen (1977)

During the process, 734.12 ft of cold water is needed in the cooler. This will cost,

$$734.12\ \text{ft}^3 \times \$2.5/1000\ \text{ft}^3$$

$$= \$1.83$$

The pumping requirements were calculated to be

$$\$0.10$$

Therefore, the cost for pasteurizing 1,000 gallons is

$$1.99 + 1.83 + 0.10 = \$3.92$$

The above operational cost could be lowered by having a regenerative unit connected after the holding coil where the pasteurized brine would preheat untreated brine. The preheated brine after the regenerative unit would be pumped directly into the heating section. Thus, regeneration would save both fuel and cold water since the untreated brine would exit the regenerative unit at a temperature higher than that of the room, while the pasteurized brine would exit the regenerative unit at a temperature lower than that of the holding coil. For example, if the designed percent of regeneration would be 50%, the amount of heat added to the brine in the heater and that removed from the brine in the cooler would be reduced by 50%. This will further reduce the amount of fuel and cold water required by 50%. Thus, the operational cost may be reduced approximately 50%.

A serious corrosion problem could occur with regeneration because a large surface area is in contact with the brine. Cherry brine is considered to be a highly corrosive material. The metals that should be used in the manufacture of a pasteurization unit must be of high quality in order to resist corrosion of the metal surfaces in contact with brine. Such a requirement will affect the initial and installation cost of the pasteurization unit.

Cucumber brine is probably as corrosive as cherry brine. Using the following information the cost of pasteurizing 1000 gallons is estimated as follows:

Labor cost @ $4.00/hr	2.64[1]
Propane	1.29
Pumping	0.79
Chemicals	2.41
Total operational cost:	$7.05
Pasteurizer 5 yr depreciation $8915/yr:	$10.51

The above indicates that the initial cost of the pasteurization unit is higher than the operational cost ($10.51 vs $7.05). Therefore, a less expensive heat exchanger may contribute in cost reduction of the pasteurization process.

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[1] From McFeeters *et al*. (1977)

CONCLUSIONS

1. The degree of inactivation of polygalacturonase present in spent cherry brine can be measured by the agar "cup plate" technique after pasteurization of the spent cherry brine in a helically coiled heat exchanger.
2. Application of the general and analytical methods for predicting the degree of polygalacturonase inactivation can give satisfactory results.
3. If adequate heat transfer information is available, the analytical method is more advantageous since the computations are more rapid than the general method.
4. Predicted inactivation and that measured by the agar "cup plate" procedure were found to vary from -1.9 to +2.34%. The variation becomes smaller as the degree of inactivation increases.
5. When the constant temperature along the holding coil is 81°C and the brine is held 9 sec at this temperature, polygalacturonase is inactivated by 99.985%.
6. Shell-side convective heat transfer coefficients in a helical heat exchanger can be calculated by considering coexistence of natural and forced convection.
7. The LMTD of a helical heat exchanger and further the over-all heat transfer coefficient can be estimated in a manner similar to that of a multiple-pass counterflow heat exchanger.
8. The thermal effectiveness of the heater was 77% and that of the cooler 68%. The pasteurization process under these conditions resulted

in 99.985% polygalacturonase inactivation.

9. A regeneration unit that would give 50% regeneration in the heating and cooling sections, may contribute to approximately 50% reduction of the operational cost of the pasteurization unit.

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APPENDICES

Table A-1. Brine composition for certain cherry varieties in Michigan.[1]

Cherry variety	pH	SO_2 (%)	Ca (%)	Soluble Solids (%)
Schmid	3.1	0.345	0.312	10.5
Windsor	3.4	0.355	0.374	11.1
Bing	3.5	0.414	0.414	10.1
Napoleon	3.1	0.330	0.286	10.5

[1] From Panasiuk _et al_. (1975).

APPENDIX B

Units Conversions

	SI	ENGLISH
Length	1 m	3.2808 ft
Area	1 m^2	10.7639 ft^2
Volume	1 m^3	35.3134 ft^3
Velocity	1 m/sec	3.2808 ft/sec
Density	1 Kg/m^2	0.06243 lb_m/ft^3
Force	1 N	0.2248 lb_f
Mass	1 Kg	2.2046 lb_m
Pressure	1 N/m^2	1.45 X 10^{-4}psi
Energy, heat	1 KJ	0.9478 Btu
Heat flow	1 W	3.4121 Btu/hr
Heat flux per unit area	1 W/m^2	0.317 Btu/hr-ft^2
Energy per unit mass	1 KJ/Kg	0.4299 Btu/lb_m
Specific heat	1 KJ/Kg-°C	0.2388 Btu/lb_m-°F
Thermal conductivity	1 W/m-°C	0.5778 Btu/hr-ft-°F
Convective heat transfer coefficient	1 W/m^2-°C	0.1761 Btu/hr-ft^2-°F
Dynamic viscosity (v)	1 Kg/m-sec	0.672 lb_m/ft-sec
Kinematic viscosity (v)	1 m^2/sec	10.764 ft^2/sec
Volumetric flow	1 m^3/sec	35.3134 ft^3/sec
Power	1 KW	1.34 hp
Thermal diffusivity	1 m^2/sec	10.764 ft^2/sec
Temperature	1 °C	5/9(°F+32)
	1 °K[1]	5/9(°R)

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[1] °K = °C + 273.15

°R = °F + 459.67

APPENDIX C

Recorded temperatures at various positions within the pasteurization unit for four different trials (°C).

POSITION OF THE THERMOCOUPLE	TRIAL 1	 2	 3	 4
	Temperature along the holding coil			
	70	72	76.7	81
Brine inlet (Heater)	23.3	22.2	22.2	22.2
Brine outlet (Heater)	70.3	72.2	77.2	81.7
Brine inlet (Cooler)	69.7	71.7	76.1	80.5
Brine outlet (Cooler)	33.3	32.2	34.0	34.4
Hot water inlet	87.8	86.1	93.9	98.9
Hot water outlet	72.2	74.4	77.5	76.6
Hot water stream	81.5	79.4	84.4	87.5
Cold water inlet	11.7	13.0	13.0	11.1
Cold water outlet	25.6	19.5	22.0	19.2
Cold water stream	18.0	16.5	18.0	15.0
Upper wall (Heater)	73.3	74.4	78.8	84.4
Middle wall (Heater)	66.6	68.3	72.8	77.0
Lower wall (Heater)	37.2	38.9	40.0	40.5
Upper wall (Cooler)	53.3	54.4	57.2	58.6
Middle wall (Cooler)	32.2	34.0	36.5	37.2
Lower wall (Cooler)	24.1	23.3	25.0	25.8
Upper wall surface (Heater)	72.2	75.0	80.5	85.8
Lower wall surface (Heater)	37.0	37.7	40.5	42.5
Upper wall surface (Cooler)	48.8	42.0	45.0	47.0
Lower wall surface (Cooler)	25.5	19.5	22.0	20.0