CHAPTER
SIX

THE OPTIMIZATION OF A SHELL AND COIL NCHE

The detailed model, as is shown in this chapter, can be used to optimize the design of a NCHE. Two sets of parameters were identified as having a potential affect upon simulation results, heat exchanger parameters, and system parameters. Heat exchanger parameters describe the geometry of the heat exchanger, namely:

1) number of coils,
2) length of heat exchanger (actual heat exchanger shell length),
3) tube diameter,
4) tube spacing, and

5) heat exchanger shell diameter

while system parameters are related to other components in the SDHW system, such as:

1) location (weather data),
2) glycol flow rate,
3) collector array size, and
4) water draw.

In this chapter, TRNSYS simulations were performed on various NCHE geometries while keeping system parameters constant. Each simulation produces a solar fraction value, which is used in conjunction with an economic analysis to determine the best heat exchanger design. A range of heat exchanger geometries was then subject to variations in system parameters in order to learn the effect of each of these parameters upon the determination of the best heat exchanger design.

It will be shown that the system parameters do have a substantial effect upon optimal heat exchanger design. Economic assumptions also affect the optimal design. As the system parameters and economic assumptions will differ, the following analysis, although it does produce an optimal heat exchanger design, is useful mostly as a guide for professionals who desire to design an optimal shell and coil NCHE for a different set of system and economic parameters, using the detailed model.

6.1 Regarding the Simulations in General

6.1.1 Default System Parameters

Unless otherwise noted, all TRNSYS simulations presented in this chapter were carried out for the following conditions and parameter settings:

Weather Data:

TRNSYS weather generator for Madison, WI

Thermo Dynamics' MICRO-FLO® Collector Specifications (SRCC 1993):

$FRUL_1$ = 14.49 KJ/hr-m^2-°C (2nd order fit)
$FRUL_2$ = 0.026 KJ/hr-m^2-°C
$FR\tau\alpha$ = 0.634
b_o=0.448 (2nd order fit)
b_1=-0.234
Collector test flow rate 11.86 kg/hr-m^2
Collector area 4.5 m^2

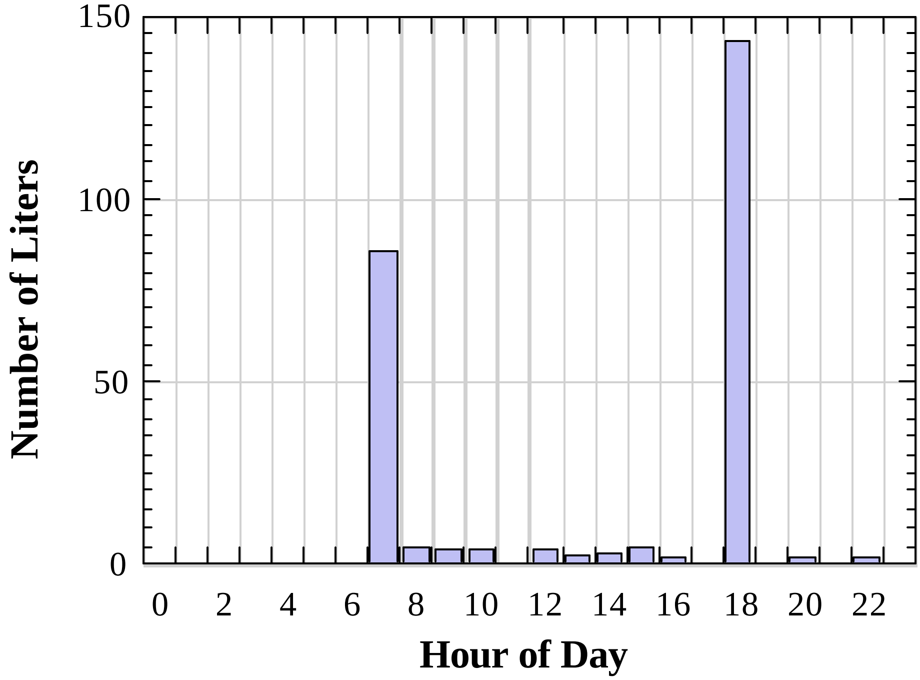

Figure 6.1.1 An average daily hot water draw profile taken from WATSIM (EPRI 1992).

Tank:

Tank Volume:	454 L
5 node stratified tank	
Daily DHW Draw:	260 L
Demand Profile:	As shown in Figure 6.1.1

Piping:

As shown in Figure 6.1.2 and Table 6.1.1

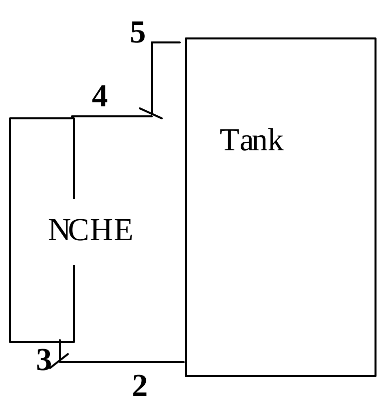

Figure 6.1.2 Diagram of pipe locations.

Table 6.1.1 Piping Parameters Used in Simulations

Pipe Location	Length [m]	Vertical Rise [m]	Diameter [m]	K
2	0.5	0	0.01905	1.5
3	0.5	0.0635	0.01905	1.5
4	0.5	0	0.01905	1.5
5	0.5*	0.965*	0.01905	1.5

* For simulations that employ variable heat exchanger shell lengths, the dimensions of pipe 5 are compensated to ensure that $\oint_{loop} dz = 0$, as described in Section 6.2.2.

6.1.2 April and Yearly Simulations

Simulation results for April closely approximate yearly results. Figure 6.1.3 presents the solar fractions of a SDHW system by month. April's solar fraction (April is month 4 in Figure 6.1.3) of 0.575 approaches the yearly solar fraction 0.559. Due to time constraints and the large number of simulations performed, TRNSYS simulations were performed for the month of April rather than for a complete year.

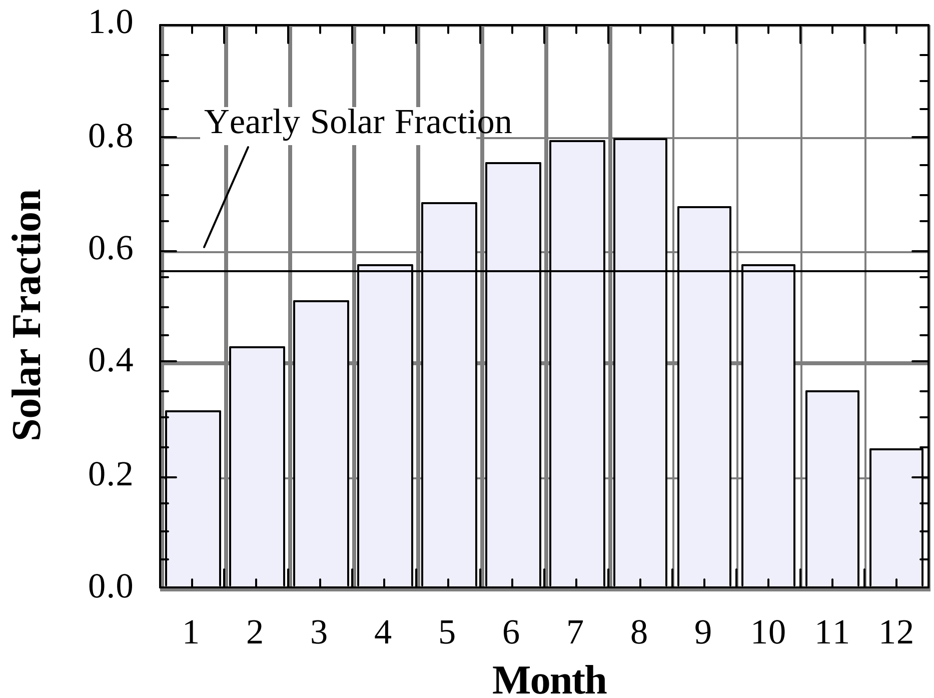

Figure 6.1.3 Solar fraction by months. April's solar fraction (located at 4) closely corresponds to the yearly solar fraction.

6.2 The Variation of Heat Exchanger Geometric Parameters

In this section, simulation results are presented which detail the performance of several heat exchanger geometries for the given set of system parameters outlined in Section 6.1.1. Plots are presented which compare the solar fraction as function of heat exchanger length, number of coils, tube diameter, and tube spacing. The purpose of the simulations run in this chapter is not to find the heat exchanger with the largest solar fraction, but to take note of what ranges of geometric parameters have the most significant effect upon system performance. The ranges are indicated by regions with the steepest solar fraction curves. Note that the addition of each consecutive coil and/or increment of heat exchanger length brings with it, not only an increase in system performance, but an increase in heat exchanger initial cost as well. Choosing the optimum heat exchanger design balances overall system performance with heat exchanger cost.

6.2.1 Variation of the Coil Spacing Intervals, Number of Coils, and Shell Diameter in the Heat Exchanger

In order to investigate whether adding additional coils or varying the coil spacing would significantly effect system performance, simulations were run for a variety of NCHE geometries. The number of coils in the heat exchanger was varied in the following manner. The 9.53 mm core and the 25.4 mm inner diameter (ID) coil #1 were common to all heat exchangers investigated. For a given S_T value, successive coils' inner diameters were found using the equation:

$$ID_{coil,\,i+1} = ID_{coil,\,i} + 2\,S_T \qquad (6.2.1)$$

Figure 6.2.1 displays some of the geometric parameters that were varied in the simulations.

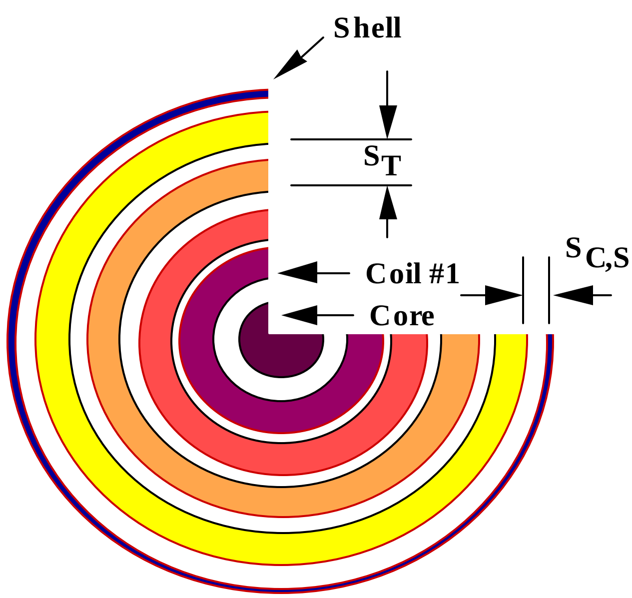

Figure 6.2.1 Diagram of some NCHE geometric parameters. $S_{C,S}$ is the distance between the outermost coil's outer diameter and the inner diameter of the heat exchanger shell.

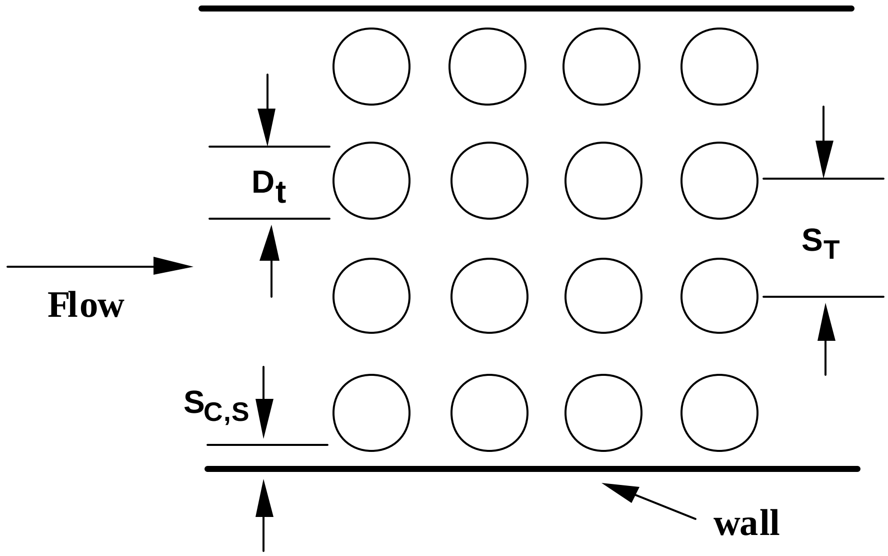

Figure 6.2.2 Diagram of some geometric parameters for tube bundles in cross flow.

As shown in Figure 6.2.2, the crossflow correlations assume a distance between the tube bundle and the outside wall, such that:

$$S_{C,S} = \frac{1}{2}\left(S_T - D_t\right) \qquad (6.2.2)$$

where $S_{C,S}$ is the distance between the tube and the heat exchanger wall. In all simulations performed in this chapter, the $S_{C,S}$ was fixed as a function of S_T, as shown in Equation 6.2.2. The heat exchanger length and the tube diameter were kept constant at 0.4064 m and 6.35 mm respectively, unless otherwise noted.

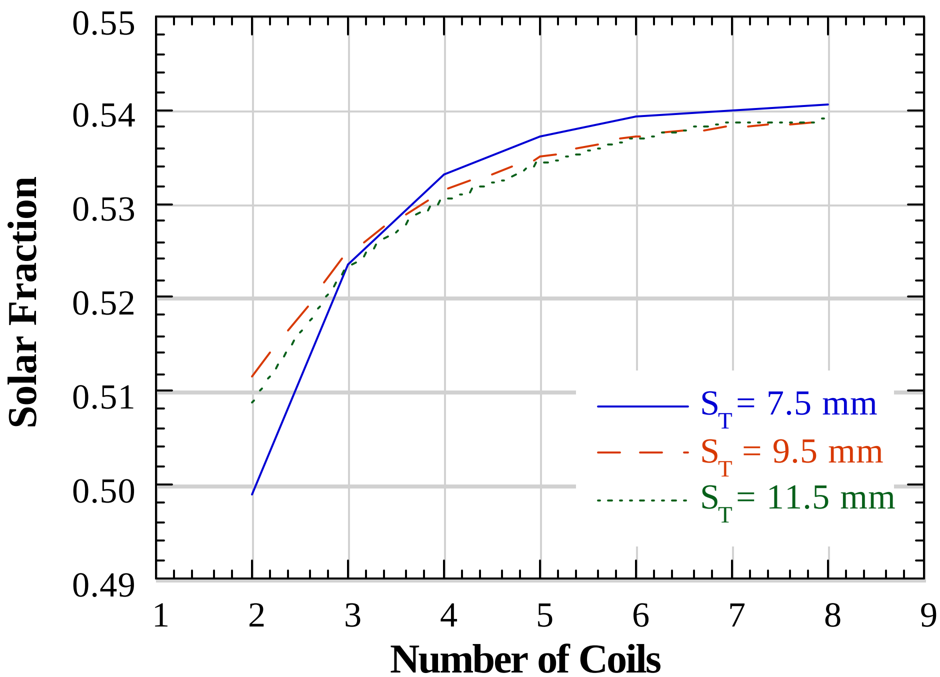

Figure 6.2.3 System performance as a function of number of coils and tube spacing.

Figure 6.2.3 presents a comparison of the solar fraction for 3 different coil spacings. For heat exchangers with 2 and 3 coils, an S_T value of 9.5 mm leads to a larger solar fraction, while for heat exchangers with 4 or more coils, a smaller spacing of 7.5 mm provides the best system performance. The behavior of the curve corresponding to 7.5 mm spacing can be easily explained. For small tube spacings, for heat exchangers with 2 or 3 coils, there is reduced inter-coil water-side flow space. As a result, the shear pressure drop in the heat exchanger increases, which inhibits water flow, and thereby reduces heat transfer in the heat exchanger. As the number of coils increases, so too does the available water-side flow space.

Although the curve corresponding to S_T=7.5 mm provides the best performance for larger numbers of coils, the curve corresponding to S_T=9.5 mm is preferred for two reasons: fouling could become more of a problem for smaller coil spacings, and as will be seen, adding extra coils results in extra expense. In any case, the effect of S_T upon simulation results is small.

6.2.2 Variation of Heat Exchanger Length

In order to determine the effect of the heat exchanger length, TRNSYS simulations were conducted for heat exchangers with various lengths. In order to accommodate for the change in heat exchanger length, the length and vertical rise of the pipe directly above the heat exchanger had to be compensated accordingly in order to maintain a balanced piping geometry. Figure 6.2.4 details two systems with differently sized heat exchangers, and the consequent changes in pipe length and vertical rise that must accommodate the new heat exchanger dimension. When the heat exchanger length is expanded, the pipe corresponding to position 5 in Figure 6.2.4 must contract in length and vertical rise. The other pipe dimensions do not change.

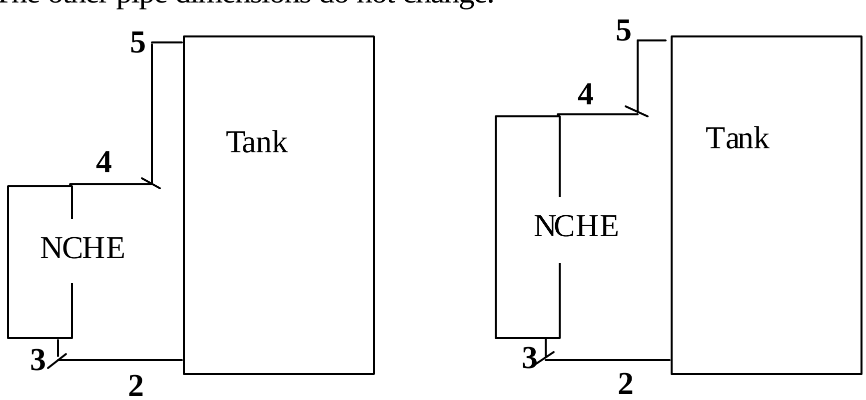

Figure 6.2.4 Diagram detailing the consequent change in pipe 5’s length and rise with the increase in NCHE height.

The following heat exchanger geometric parameters are used and kept constant in this study:

N_{coils} =	4	
$ID_{coil,1}$	=	25.4 mm
$ID_{coil,2}$	=	44.5 mm
$ID_{coil,3}$	=	63.5 mm
$ID_{coil,4}$	=	82.6 mm

Dtube	=	6.35 mm
Pitch	=	8.89 mm
D_{shell}	=	101.6 mm

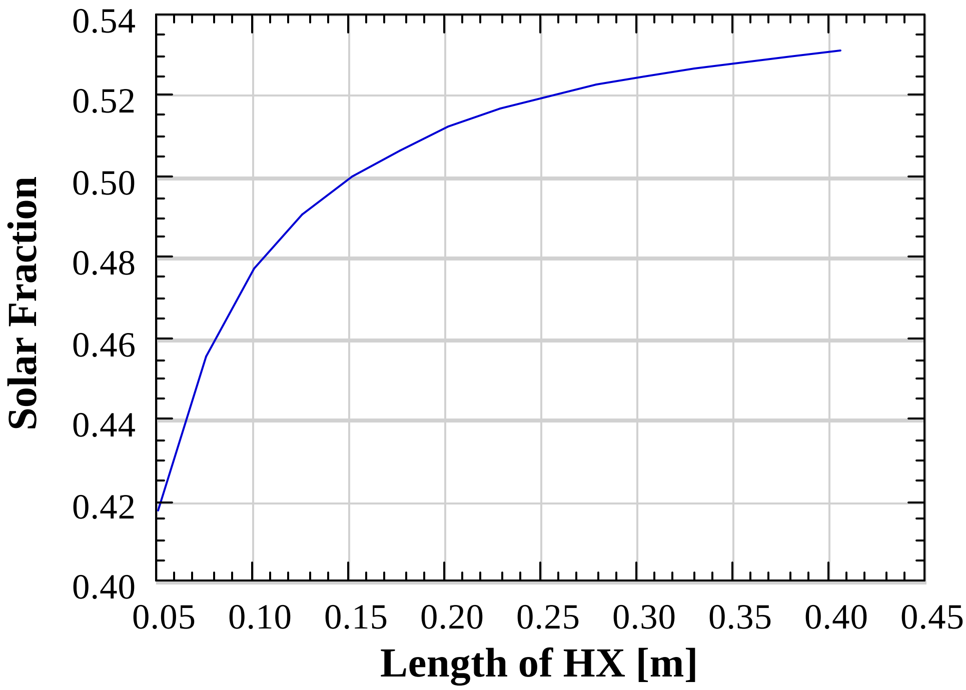

Figure 6.2.5 Solar Fraction as a function of heat exchanger length.

As is shown in Figure 6.2.5, increasing heat exchanger length leads to enhanced system performance. When the cost of each additional increment of heat exchanger length is considered, as will be in Section 6.3, what becomes most important in Figure 6.2.5, is that every additional increment of length provides dimishing returns in system performance.

6.2.3 The Effect of Tube Diameter on System Performance

In order to learn the effect of tube diameter on system performance, the tube diameter was varied along with the number of coils, shell diameter, tube spacing and pitch. Based upon a tube diameter of 6.35 mm, a tube spacing of 9.5 mm, and a pitch of 8.9 mm; S_T and pitch parameters were proportionally chosen for each tube diameter using the proportion:

$$\frac{D_{tube}}{6.35\ \ \mathrm{mm}} = \frac{S_T}{9.5\ \ \mathrm{mm}} = \frac{pitch}{8.9\ \ \mathrm{mm}} \tag{6.2.3}$$

Only standard tube sizes were simulated. As is shown in Figure 6.2.6, smaller tube diameters lead to enhanced system performance. This trend is most pronounced for heat exchanger geometries with few coils. However, use of 3.18 mm tubes is problematic due to large pumping power requirements. Small diameter tubes, are associated with high Reynolds numbers for a given glycol flow rate, which in turn is associated with high shear pressure losses, and consequently an extraordinarily large required pumping power. This trend is exacerbated by geometries employing few coils. As a result, the 6.35 mm tubing was chosen for further design considerations.

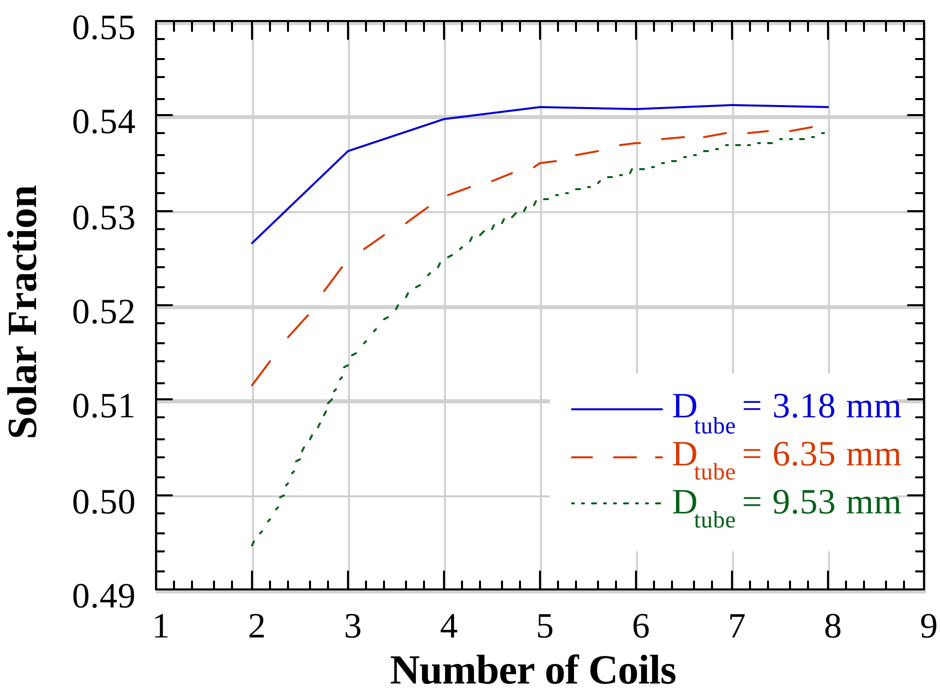

Figure 6.2.6 Solar fraction as a function of tube diameter and number of coils.

6.2.4 Summary of Findings

It was found that tube spacing has little impact upon simulation results, and tube diameter has some, but of all the geometric heat exchanger parameters varied to this point, those parameters relating to heat transfer area (i.e. the heat exchanger length and number of coils in the heat exchanger) have the greatest impact upon system performance. It could be summarized that increasing the heat transfer area, A_S, enhances system performance to a threshold value. Thereafter, additional heat transfer area does little to increase system performance. In Figure 6.2.7, the data from Figures 6.2.3, 6.2.5, and 6.2.6 have been rearranged to illustrate the relationship between heat transfer area and solar fraction. Data for heat exchangers of differing coil spacings, tube diameters and heat exchanger lengths are collected together to relate the dependence of solar fraction upon heat transfer area.

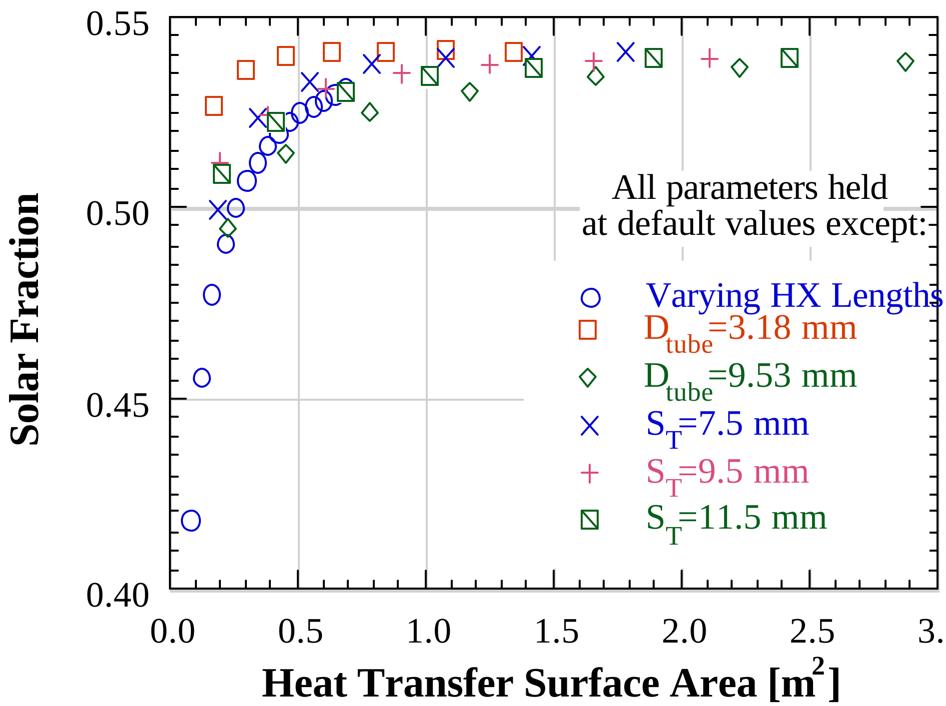

Figure 6.2.7 Solar fraction as a function of heat transfer area. Data taken from simulations that varied tube diameter and number of coils.

A tube spacing, S_T, of 9.5 mm and a tube diameter of 6.35 diameter was chosen for the optimization simulations that follow.

6.3 The Economic Analysis of a Modified Heat Exchanger Design

In designing an optimum shell and coil heat exchanger, economic considerations need to be considered. The potential increased cost of purchasing a larger or more intricate heat exchanger must be balanced with the additional yearly savings possible from an improved design. Frequently domestic hot water systems can be improved to promote system efficiency, but at considerable cost in hardware, that such improvements are not economically justified. Optimization might even include reducing the solar fraction of the NCHE, if in doing so, the design of the NCHE is simplified and substantially less expensive. The optimal design, then, herein presented, will not optimize system performance, but will optimize consumer savings over a period of years. The best heat exchanger will be that which over the long run contributes to the most economically efficient SDHW system.

6.3.1 Life Cycle Savings Method

The economic analysis to be used in weighing the heat exchanger cost against the projected yearly fuel savings will employ the life cycle savings (LCS) method in conjunction with the P_1, P_2 method (Duffie and Beckman 1991). The life cycle cost is the sum of all costs associated with an energy delivery system over its lifetime or over a period of economic analysis. This would include the down payment and mortgage payments on the system, parasitic and maintenance costs, etc. The life cycle savings is defined as the difference between the LCC of a conventional DHW system and the LCC of a solar plus auxiliary energy DHW system.

The P_1, P_2 method is a simple way to calculate the LCS of any changes in the heat exchanger design. For this analysis the LCS is given by:

$$LCS = P_1 C_{F1} L F - P_2 \left(C_E + C_A A_C \right) \qquad (6.3.1)$$

where: P_1 = ratio of life cycle fuel cost to 1st year fuel costs

P_2 = ratio of life cycle expenditures that occur due to additional capital investment in the energy delivery system to the initial cost

C_{F1} = initial fuel cost

L = the yearly energy load that must be met by the energy delivery system

F = the solar fraction of the SDHW system

C_E = total cost of the solar equipment which is independent of collector area

A_C = solar collector area

C_A = cost of the solar equipment which is a function of collector area

The procedure for calculating P_1 and P_2 can be found in Duffie and Beckman (1991). The optimization in this work is based upon a 10 year economic analysis. Following are the economic assumptions used in calculating P_1 and P_2 . Maintenance and parasitic costs are considered negligible, there are no assumed property taxes on the equipment, and the resale value after 10 years is zero. The down payment is 1/6 of the equipment cost, while the rest is paid in a 5 year mortgage with a 9.5% interest rate. The inflation rate of fuel is assumed to be 6.5%. The discount rate is taken as 10.5%. The owner's effective tax bracket is assumed to be 0.42. Using the parameters listed above, P_1 is found to be 7.709 and P_2 is 0.894.

The objective of the optimization is to increase the *LCS* of the a SDHW system employing the existing Thermo Dynamics shell and coil heat exchanger. As a result, a *DLCS* is calculated, such that a positive *DLCS* represents an additional life cycle savings beyond the savings accrued by using the Thermo Dynamics heat exchanger. Accordingly a negative *DLCS* represents a system that over the period of economic analysis, would save the consumer less than the Thermo Dynamics system. Given the Thermo Dynamics heat exchanger in a SDHW system, the *LCS* can be found as:

$$LCS = P_1 C_{F1} L F_{TD} - P_2 \left(C_{TD} + C_E + C_A A_C \right) \tag{6.3.2}$$

where F_{TD} and C_{TD} are the solar fraction and heat exchanger cost for the Thermo Dynamics heat exchanger. For heat exchanger designs considered in this chapter, which form a part of a SDHW system, the *LCS* can be found using:

$$LCS = P_1 C_{F1} L F_{Design} - P_2 \left(C_{Design} + C_E + C_A A_C \right) \tag{6.3.3}$$

where F_{Design} and C_{Design} are the solar fraction and heat exchanger cost for the heat exchanger designs explored in this chapter. The change in LCS, the $DLCS$, is found by taking the difference of equations 7.1.1 and 7.1.2, such that:

$$D\,LCS = P_1\,C_{F1}\,L\left(F_{Design} - F_{TD}\right) - P_2\left(C_{Design} - C_{TD}\right) \qquad (6.3.4)$$

The optimal heat exchanger design is the design with the best combination of high system performance and low estimated cost such that it yields the highest $DLCS$.

6.4 The Optimization of the Shell and Coil NCHE

6.4.1 Estimating Heat Exchanger Cost

In order to employ the ΔLCS method, it is necessary to make some estimates on the price of a modified NCHE design relative to Thermo Dynamics' existing NCHE design. Although Thermo Dynamics does not sell a shell and coil NCHE separately, but only as part of a complete SDHW system, an estimate of the price of the 0.4064 m, 4 coil, shell and coil NCHE was given as approximately \$400.00. The following assumptions are made in order to estimate the prices of the heat exchanger designs investigated in this work:

1) Every heat exchanger will have a fixed cost associated with it for headers, inlet and outlet piping, and heat exchanger shell. This cost, C_{fixed}, is assumed to be \$100.00.

2) It is assumed that it would take a laborer 15 minutes to bend copper tubing into a coil sized for a 0.4064 m heat exchanger. Longer or shorter coils would take proportionally more or less time. Assuming a worker's wage and all costs associated with the employee (i.e. health insurance, unemployment insurance, etc.) cost the manufacturer \$40.00 hourly, then the labor involved to bend one coil of 0.4064 m would cost \$10.00. The labor cost can therefore be represented by:

$$C_{labor} = \frac{\$10.00}{\text{coil}} N_{coils} \frac{L_{HX}}{0.4064\ \text{m}} \tag{6.4.1}$$

where L_{HX} is the heat exchanger shell length.

3) The manufacturer can purchase 6.35 mm OD copper tubing for \$1.97/m. The tubing cost per heat exchanger can be represented by:

$$C_{tubing} = \frac{\$1.97}{\text{m}} L_{tubing} \tag{6.4.2}$$

where L_{tubing}, the total tube length, was found using the methods described in Section 4.1.

4) The manufacturer charges twice the manufacturing cost for the heat exchanger. Profit and other overhead are included in this markup.

Hence the estimated cost is calculated using the following equation:

$$C_{estimated} = 2 \times \left(C_{labor} + C_{tubing} + C_{fixed}\right) \tag{6.4.3}$$

The Thermo Dynamics NCHE, using the above relation costs $400.00. Figure 6.4.1 presents the estimated heat exchanger costs as a function of heat exchanger length and number of coils.

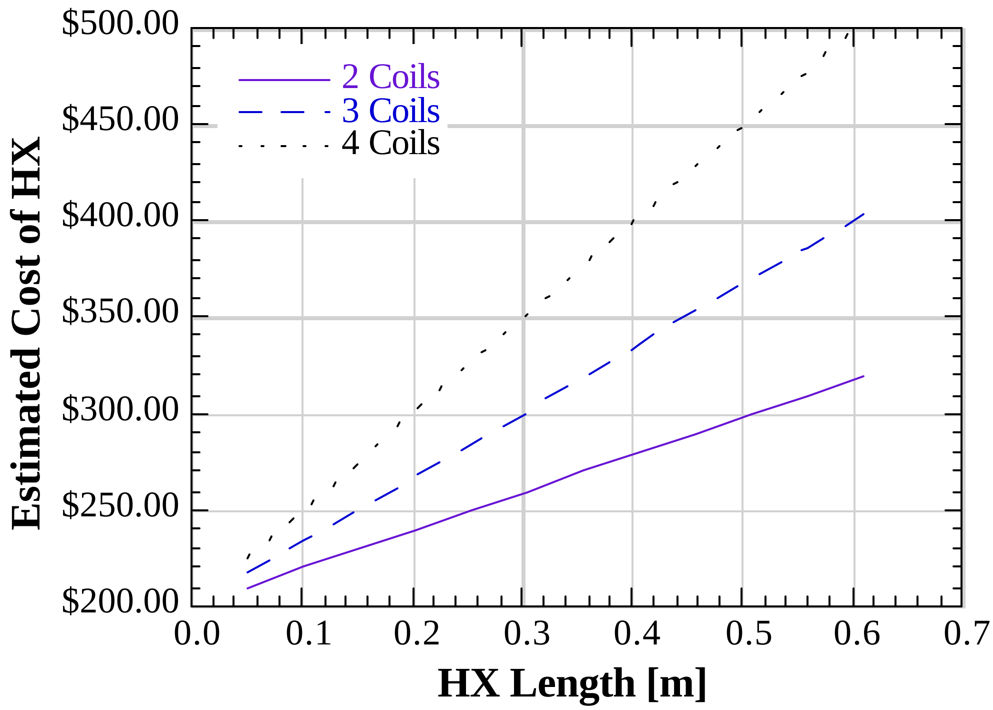

Figure 6.4.1 Estimated heat exchanger cost as a function of heat exchanger length and number of coils.

6.4.2 Optimization Simulations and Results

Using an S_T of 9.5 mm and 6.35 mm diameter tubes, number of coils and the heat exchanger length were varied for simulations in order to generate a solar fraction and an estimated heat exchanger cost. These estimated costs and solar fractions were compared to the standard (4 coil, 0.4064 m) model's cost of $400.00 and solar fraction for the given set of system parameters. It was found that the more economically efficient heat exchangers were smaller than the Thermo Dynamics model's specifications of 0.4064 m and 4 coils. Figure 6.4.2 plots the solar fraction as a function of heat exchanger length and number of coils.

From Figure 6.4.3 the optimal NCHE configuration can be selected as that heat exchanger that will yield the greatest ΔLCS. A 2 coil, 0.45 m heat exchanger should save the consumer \$54 over ten years. As shown in Figure 6.5.4, a 2 coil, 0.45 m heat exchanger would cost approximately \$290, or about \$110 less than the standard Thermo Dynamics 4 coil, 16 inch model. Note that the optimal heat exchanger design found in Figure 6.4.3 applies to the system parameters listed in Section 6.1.1. As will be shown in the following sections, variation of system parameters affects the optimal design of the heat exchanger.

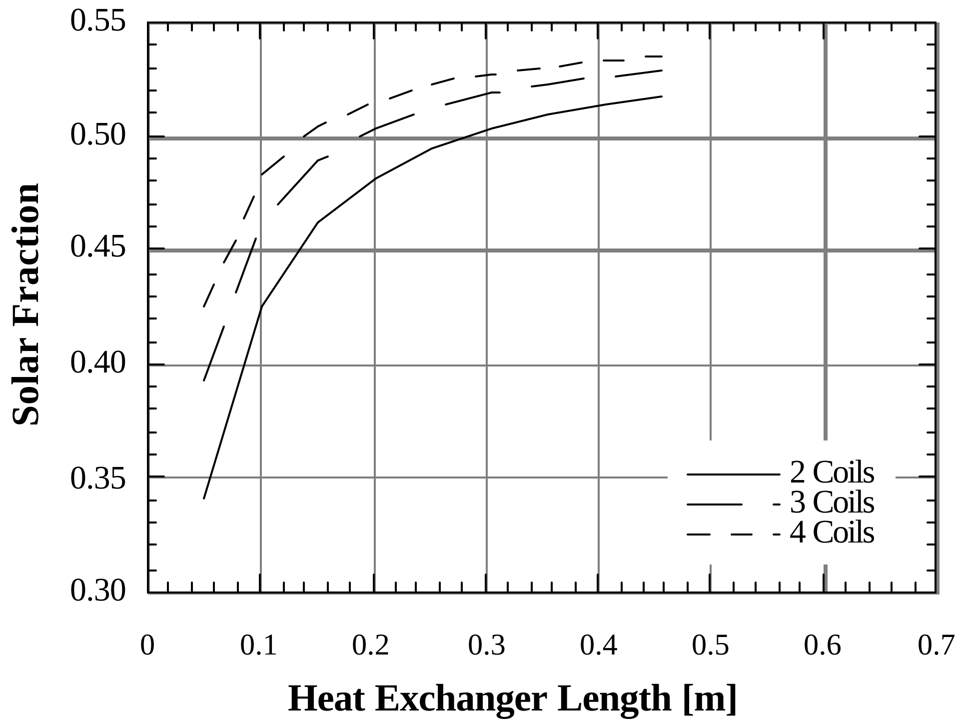

Figure 6.4.2 Solar fraction as a function of heat exchanger length and number of coils.

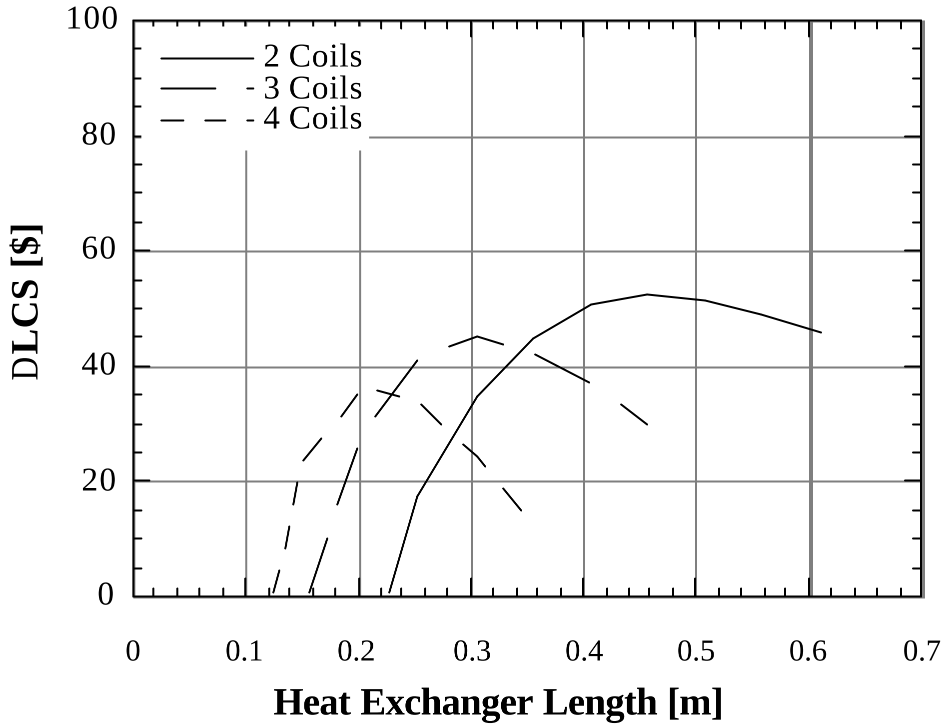

Figure 6.4.3 ΔLCS as a function of heat exchanger length and number of coils. For the given system parameters detailed in Section 6.1.1, the optimum heat exchanger configuration is 0.45 m long and contains 2 coils.

6.4.3 The Effect of System Parameters Upon Optimal Heat Exchanger Design

It was found that all of the system parameters tested had an effect upon the optimal heat exchanger design. The following graphs present the relationship between the system parameter tested and the consequent optimal heat exchanger length. In all simulation results from this work, it was found that a 2 coil heat exchanger was economically the most beneficial. Therefore only plots for 2 coils heat exchanger geometries will be presented.

Smaller collector array sizes require smaller heat exchangers. Figure 6.4.4 details the reliance of the optimal heat exchanger length upon collector area. As smaller collector arrays absorb less solar energy, a smaller heat exchanger is required to transfer the energy to the water. If the

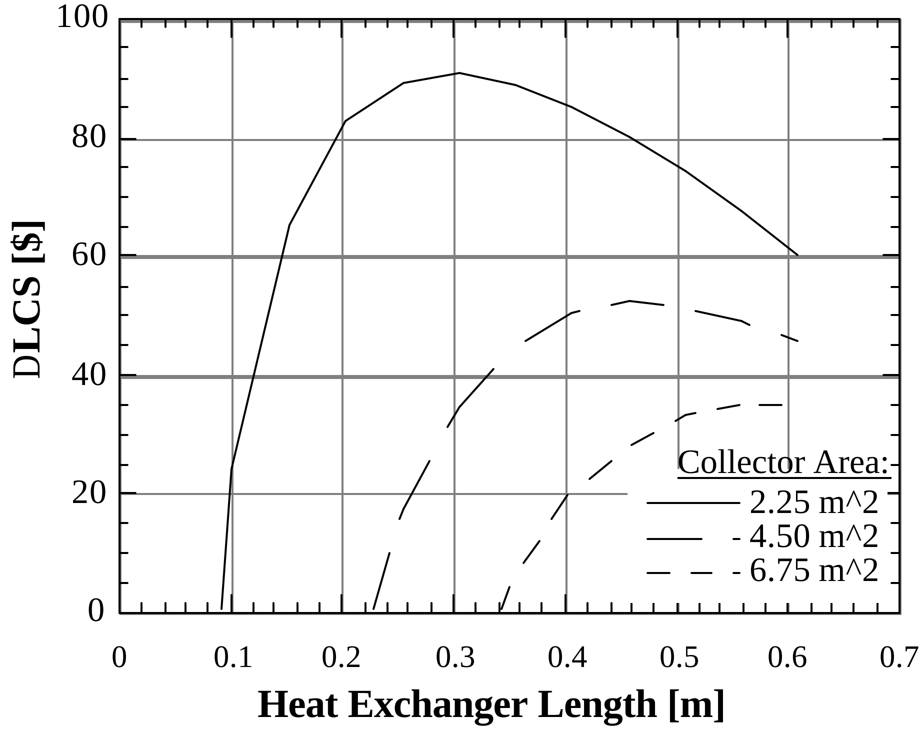

Figure 6.4.4 The effect of variation of collector area upon choice of optimal heat exchanger length for 2 coil heat exchanger design.

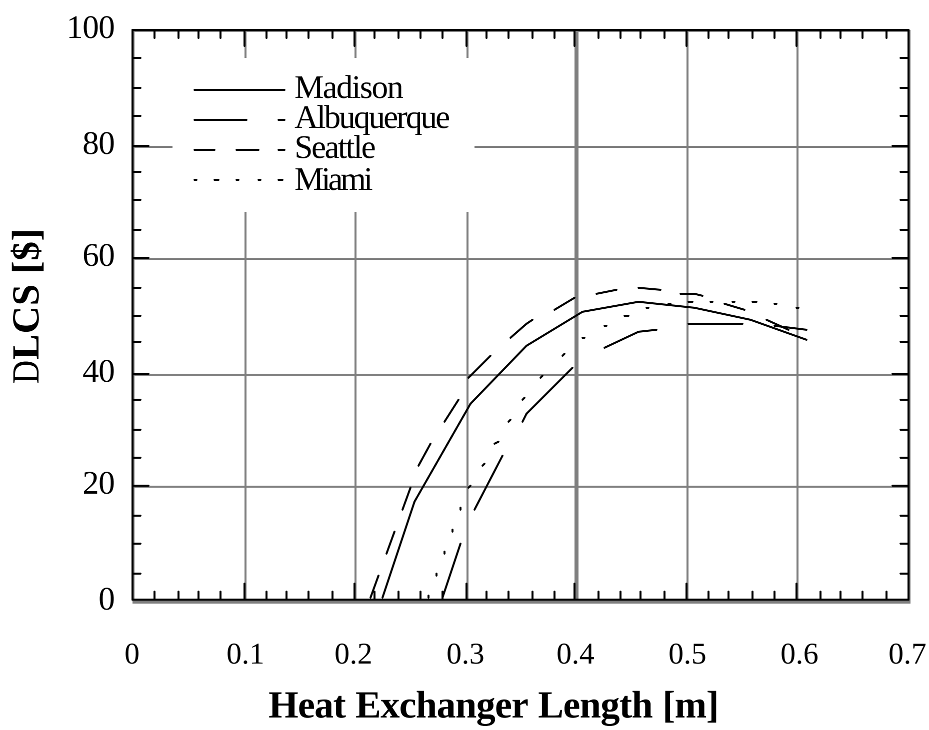

Figure 6.4.5 The effect of location upon choice of optimal heat exchanger length for the 2 coil heat exchanger design.

0.45 m optimal heat exchanger length found in Section 6.4.2 were chosen, for the small collector array, approximately $30 additionally could be saved by the consumer. By choosing the optimal heat exchanger length of 0.3 m for the small collector array, an additional $10 could be saved.

Warmer climates also require larger optimal heat exchanger designs. As is shown in Figure 6.4.5, Miami and Albuquerque require heat exchanger lengths from 0.54 to 0.58 m whereas Seattle and Madison require shorter heat exchanger lengths of 0.45 m. As the collectors in warmer climates take in more thermal energy, a longer heat exchanger design is required to maximize economic performance. However, it must be noted that if a heat exchanger length of 0.45 m was chosen rather than 0.54 or 0.58 m for Miami or Albuquerque, the difference in consumer savings over the period of analysis is less than $5.

As is shown in Figure 6.4.6, the magnitude of the daily hot water draw also significantly affects optimal heat exchanger length. Larger water draws require larger heat exchangers. As systems with small draws require smaller heat exchangers, an optimally designed heat exchanger can save more money for the consumer. For the case with smaller draws, the difference in *DLCS* for the 0.45 m heat exchanger and a smaller 0.35 m heat exchanger, amounts to $5.

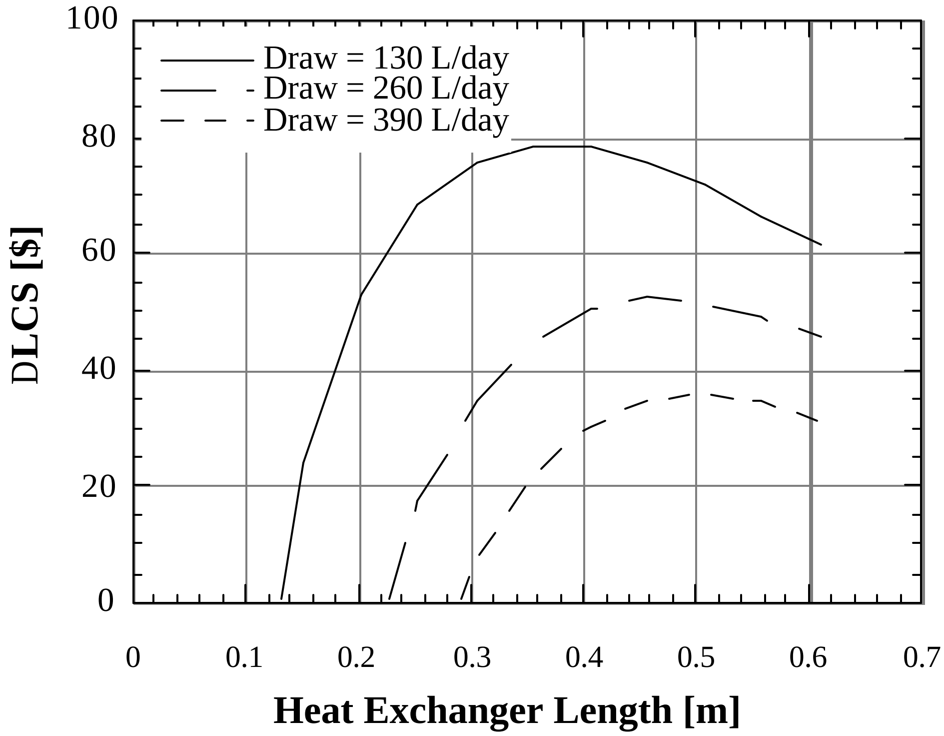

Figure 6.4.6 The effect of magnitude of water draw upon choice of optimal heat exchanger length for the 2 coil heat exchanger design.

The glycol flow rate also affects the optimal heat exchanger length. As is shown in Figure 6.4.7, although glycol flow rates of 0.03 and 0.04 kg/s require the same heat exchanger lengths for optimal performance, a smaller glycol flow rate of 0.02 kg/s leads to a longer optimal heat exchanger length. However, as the difference in *DLCS* is small, an optimal length of 0.45 m is adequate for each of the glycol flow rates tested.

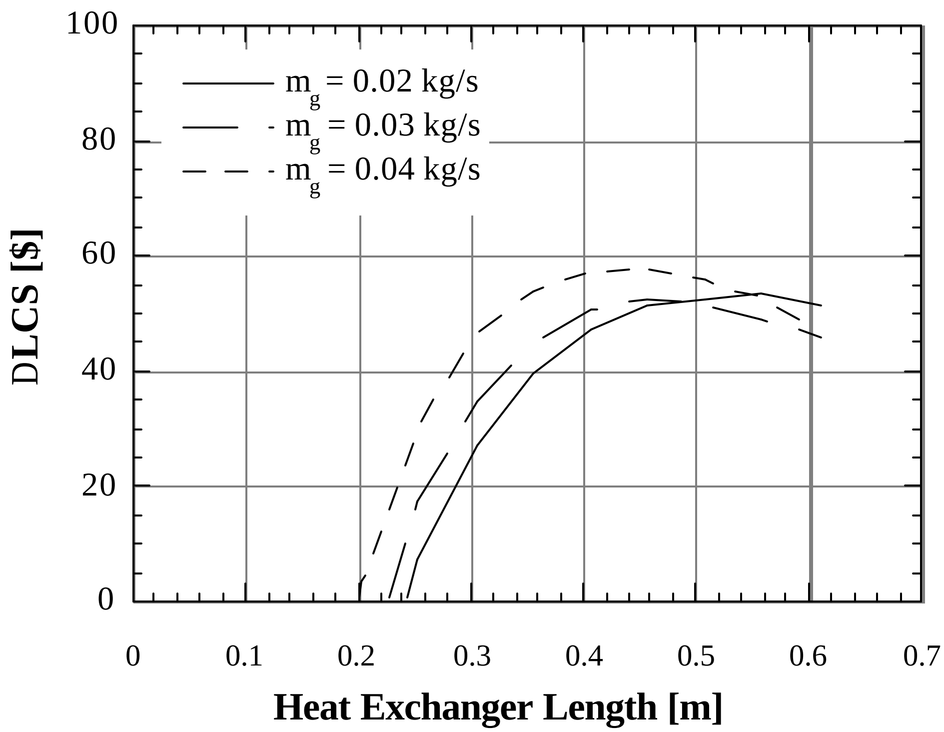

Figure 6.4.7 The effect of glycol flow rate upon choice of optimal heat exchanger length for the 2 coil heat exchanger design.

6.5.4 The Optimal Heat Exchanger Design

In Figures 6.4.3-7, the largest *DLCS* reported is approximately \$80. Over a ten year period, a modified heat exchanger design could save the consumer only up to an additional \$8 per year, which is nearly insignificant. These small *DLCS*s suggests that the Thermo Dynamics NCHE is well-designed but that some improvement can still be made upon the model.

For the simulation results presented, although each system parameter leads to different optimal heat exchanger lengths, choosing an optimal heat exchanger length of 0.45 m can be considered adequate for all the cases herein presented. The increase in *DLCS* that could be achieved by a choosing a heat exchanger sized for each particular set of system parameters is less than \$10 or 11% of the *DLCS* for every case considered over the 10 year period of economic analysis.

Consequently, the chosen optimal heat exchanger design specifications are those that were chosen in Section 6.4.2, and are listed below:

N_{coils}	=	2
$ID_{coil,1}$	=	25.4 mm
$ID_{coil,2}$	=	44.5 mm
Dtube	=	6.35 mm
Pitch	=	8.89 mm
D_{shell}	=	101.6 mm
L_{HX}	=	0.45 m
$C_{Estimated}$	=	$289.58

For the system parameters listed in Section 6.1.1, the heat exchanger delivered a solar fraction of 0.5177 while the Thermo Dynamics heat exchanger delivered a solar fraction of 0.5329. The resulting *DLCS*, based upon the economic assumptions listed, is $52.32. Over a 10 year period, the consumer should save an additional $52.32 from the SDHW system if the optimally designed heat exchanger is used in place of the Thermo Dynamics model.

The choice of economic parameters can change the optimal heat exchanger design found using this method. Therefore, although an optimal design is presented, it should be understood that the design depends upon a particular set of economic assumptions.