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WASTE WATER HEAT RECOVERY APPLIANCE

Final Report

Appropriate Technology Small Grants Program

Grant #DE-FG48-81R801073

ABL Corporation
Box 1918
Fort Collins, Colorado 80522

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SUMMARY

An efficient convective waste heat recovery heat exchanger was designed and tested. The prototype appliance was designed for use in laundromats and other small commercial operations which use large amounts of hot water.

Information on general characteristics of the coin-op laundry business, energy use in laundromats, energy saving resources already in use, and the potential market for energy saving devices in laundromats was collected through a literature search and interviews with local laundromat operators in Fort Collins, Colorado (Chapter 1).

Laundromat owners are interested in saving energy. Twenty-five percent of gross sales is spent on utilities for the average laundromat. Fifty percent of the gas used is for hot water. The two main problems in installation of energy saving devices are floor space and any need for modifications in the layout of the laundromat. The high percent of laundromats occupying rented building space adds to this problem. Energy saving devices such as the waste water heat recovery appliance designed in this project are easiest to install in new applications although the need for saving energy may be most accute in the less efficient existing coin-ops (Chapter 1).

A brief survey of time-use patterns in two local laundromats was conducted (Appendix D). The results were used, with additional information from interviews with owners, as the basis for the statistical model developed (Chapter 2). Two laundromat water heating system models were used, "advanced" and "conventional." In the hypothetical "advanced" model warm water is kept separate from the hot and cold (Chapter 2). Mathematical models for the "advanced" and "conventional" types were developed and the resulting computer program listed (Appendix A). Computer simulations were made using a variety of parameters; for example, different load profiles, hold-up volumes, wall resistances, and wall areas. The computer simulation results (Appendix B) are discussed with regard to the overall conclusions (Chapter 5).

Various materials were explored for use in fabricating the appliance (Chapter 3). Resistance to corrosion, workability, and overall suitability for laundromat installations were considered for each material. A fiberglass mesh and resin prototype tank was constructed. This material and metals were subsequently rejected because of the impracticability of manufacturing site specific sizes. Polypropylene plastic for the tank and PVC plastic for the heat exchanger element were ultimately chosen for construction of the system. The major advantages cited are ease of fabrication, light weight, good tensile strength, corrosion resistance, and cost effectiveness. Construction

of the tank and heat exchanger element included placement of thermocouples for test purposes (Chapter 3).

Experimental tests were performed on the prototype waste water heat recovery appliance (Chapter 4). Tests verified the operation of the flow diffuser stratification in the tank. Test results indicated an overall heat resistance for the tank walls and for the heat exchanger element wall which confirmed theoretical heat transfer values used in the simulations.

The future development of the laundromat industry is uncertain. Many approaches are being taken by laundromat operators to lower energy costs, e.g. using cold water rinses and wash. Specific recommendations for new and existing laundromats (Chapter 5) are: minimize mixing of hot and cold water, plan for waste water heat recovery when designing new laundromats, use solar and heat pumps to augment waste water heat recovery, and use life cycle costing to determine applicability.

The results of the computer simulations are discussed with schematics and recommendations for modification of laundromat heating systems to achieve greater efficiency (Chapter 5).

1. INTRODUCTION

The main goal of this project was to develop a new type of waste water heat recovery appliance particularly suited for use in laundromats and other small commercial applications. An efficient convective waste heat recovery heat exchanger was designed and tested.

Background information on coin-op laundromats and their energy use is presented in this chapter. Chapter 2 discusses computer simulations which were used in the analysis of different design parameters. Design and fabrication of a prototype waste water heat recovery appliance is described in Chapter 3. Testing of the appliance is discussed in Chapter 4. The final chapter includes discussion of major findings and conclusions. Rules of thumb for heat exchanger design and recommendations for energy efficient laundromat design are also included in Chapter 5.

BACKGROUND -- overview of the laundromat industry

The first automatic washing machine was introduced in the U.S. in 1937. By the late 1940's improved machines were available and simple dryers were on the market. The sales of automatic washers increased but did not actually overtake wringer washers until the early 1950's.

The first coin operated washing machines were installed in apartment buildings and military bases in the mid-1940's. In a typical arrangement the landlord contracted with an operator who supplied the machines and maintenance. Coin-operated laundromats were first established in the same period. One operator often had both routes and a store. The pick up and delivery of home and commercial laundry was also often part of coin-op businesses until later years when the use of trucks and higher minimum wages generally made it uneconomical ("Coin," May 1982: 25-29; "The Recorder," Vol. 34:8: 6).

The number of coin-operated laundromats in the U.S. continued to grow until the early 1970's. At this time some operators claim that there were "too many stores" for the market and that the market itself had been adversely affected by home machine use. Since that time the number of laundromats has decreased slightly in many parts of the country ("Coin," May 1982:29; "American Coin-Op," Jan 1982: 20). In 1980 there were over 25,000 laundromats in operation plus over 11,000 commercial power laundries and 7,000 hand laundries.

Assuming 25,000 laundromats there is an average of one laundromat per 9,000 people in the U.S. In Fort Collins, Colorado, there are 12 laundromats or about one per 6,000 people.

Although the total number of laundromats has declined somewhat from the mid-1970's the average floor size of

laundromats has increased 15% since 1977 and in 1981 was close to 2000 ft² ("Coin," June 1982: 24). The larger the floor area of a store the larger number of washing machines generally found in a store.

Coin-operated laundry stores represent a substantial investment. The cost of outfitting a new store runs between \$2,000 and over \$3,300 per washing machine in the store depending on the locale. New construction was down in 1981, but predictions for 1982 were optimistic with some distributors reporting a renewed interest in coin-ops by investors ("American Coin-Op," Jan 1982: 20).

Total coin-op sales were up an average 8.7% in 1981 compared to a similar increase in 1980 of 8%. However, sales for wash and dry only were up 10.5% in 1981 compared to 7.4% in 1980. Price increases in coin-ops averaged 12.1% in 1981 and 15.6% in 1980 ("Coin," May 1982: 14-16).

A major expense for many laundromat operators is rent. Only 14% of all coin-ops in the U.S. are located in buildings owned by their operators. The average store rent in 1981 was \$741.53 per month or about \$4.77/ft² per year. Of those renting, 76.6% are currently operating on a lease originally written for 5 years or less. The average lease period is 6.1 years. Half of the leases provide no time option to renew ("Coin," July 1982: 10-17).

One important characteristic of the coin-op laundry business is that most stores offer more than one service. Only 30% of all coin-op stores are just a self-service

laundry ("Coin," June 1982: 33). A store may offer drop off dry cleaning, coin-op dry cleaning, and drop off laundry in addition to the basic wash and dry. Some stores also have carwashes, showers, food, shoe repair, and other services.

One aspect of laundromat operation is the variable flow of customers. National industry efficiency ratings for 1981 show that the average laundromat is only busy (machines available in use) 16.8% of the time they are open (see Appendix D for a time-use study conducted in Fort Collins laundromats). In some areas, for example regions dependent on tourism, use patterns may be seasonal and there is also a "summer slump" phenomenon associated with vacation time ("American Coin-op," April 1982: 6).

Coin-op operators are characteristically retired or have another job according to the trade journals. Operators frequently work with their spouse in running the laundromat. Operators often own more than one laundromat. The average number of stores owned in 1979 was 2.5 and this increased to 3 per owner in 1981 ("Coin," May 1982: 25).

ENERGY USE IN LAUNDROMATS

Utility costs average 25% of the gross income for coin-op laundromats ("Coinomatic Age," Dec 1981: 14-15). Fifty percent or more of the gas used in a store is for hot water (De Roma, June 1982: 28). A typical laundromat uses about 6000 gallons or more of water during a 14-16 hour day (168,000 or more a month) and much of it is hot water. Of the total energy used in washing clothes, 95% is for hot

water (the balance for operating the motor). Switching from a warm to a cold rinse consequently saves about one-third on energy use ("Coin," July 1982: 24). Fuel costs in 1981 for laundromats went up an average 15% and the trade journals reflect a growing concern among operators with saving energy.

ENERGY SAVING MEASURES IN USE IN LAUNDROMATS

Information on energy saving devices and practices currently used in laundromats was drawn from a review of articles and advertising in one regional and three national trade journals ("Coin," "Coinomatic Age," "American Coin-op" and "The Recorder"). Laundromats frequently offer services in addition to the self-service wash and dry and consequently some of the energy saving devices in use which apply to dry cleaning are included in the following list.

I. General Energy Saving Practices

The two most basic and least costly practices in terms of up front expenditures are to improve routine maintenance and to modify hot water settings and existing machine cycles to optimize energy savings.

A. Maintenance

Steam, condensate and hot water lines are more efficient if kept in good repair and if corrosion is reduced. Dryer vents and filters should be kept clean. Timers can be used to regulate the turning on and off of water heaters, signs, and lights which saves energy in off-hour periods.

B. Settings and cycles

Rinse cycles can be permanently set so that they use only cold water. Pre-wash cycles can be bypassed to reduce warm water use.

II. Energy Saving Equipment and Materials

Energy saving techniques for laundromats can be divided into four categories. First, insulation can be added. Second, there are new machines being manufactured which have energy saving features built into the washers and dryers. Third, there are individual devices which can be used with existing equipment to improve efficiency. Fourth, there are whole systems which may include a combination of energy efficient options from the first three categories.

A. Insulation

The hot water system can be heavily insulated (pipes and tank) to reduce undesired heat loss. The building itself can be better insulated in many cases to reduce space heating and cooling loads.

B. New Machines

The majority of new equipment ads and manufacturers' specifications stress improved energy savings on new washers and dryers. For example, the Wascomat Generation 4 front-loading washer claims it can save an operator up to 56% on energy expended by reducing soak and wash times and using a cold rinse. Other companies claim similar savings for their new

washers (see Appendix E). The new 30 lb Maytag and Dexter Co. dryers have built-in heat reclaimers which recirculate 40% of the warm exhaust air and have electronic spark ignitions. New dry cleaning machines are also more energy efficient such as the Marvel EE which has a self-contained heat pump system for drying and cooling cycles.

C. Retrofitting Existing Equipment

Modifications can be made and new devices can be added to save energy with existing equipment. Dryer pilots can be replaced with electronic spark ignitions which save an estimated 2%. Air to air heat exchangers can be added to individual dryers to preheat incoming air without moisture buildup. Hot air recycle systems claim to decrease bills by 15% and have an average payback period of less than one year. Pipes can be rerouted to preheat the wash water with heat from dryer exhaust cutting costs for heating hot water by 8% ("American Coin-Op," March 1982: 18). There is one waste water heat reclaimer currently on the market. The Franklin Heat Reclaimers Inc. Model 800 is designed to reduce gas bills for hot water by approximately 20%. The device is a counterflow heat exchanger and in 1981 cost \$2,500 to \$3,500 depending on the size and type of installation.

D. Systems

One of the most common system applications is solar. Solar systems have been installed on both new and existing laundromats. Solar applications usually involve buying or leasing an array of collectors which preheat water stored in large tanks. For example, a California laundromat solar system provides 78% of the energy to heat 4,000 gallons of hot water per day. Two additional storage tanks hold 2,000 gallons for evening use. It is estimated that this leased system will save \$165,000 (42%) by the end of the seven year lease period ("Recorder," 34:4: 14-18). Another solar system, in Wisconsin, reduced hot water bills by 15%. The owner estimated a payback period of 8-10 years for the system he had installed. He also installed air to air heat exchangers on some of the dryers for additional energy savings ("American Coin-Op," July 1982: 6-10). Another type of system is one used in a hot climate where an air conditioning system was modified by installation of a heat transfer system in the freon line between the compressor and the condenser. The system heated water to well above 120° F. This water was then mixed in with cold water for washer use. In addition to saving on hot water bills there was a 10% savings on electricity resulting from

reduction of head pressure on the air conditioner ("American Coin-Op," July 1982: 353-36).

In a fourth example of larger systems for conserving energy a new coin-op laundry in Minnesota was built with an adjoining car wash. The building was well insulated and the exhaust from 30 lb steam dryers heated the carwash and laundry area in winter. The system used incoming cold water to cool the building in summer. Utility costs were 20% of gross sales which is good for a cold climate ("American Coin-Op," May 1982: 20-25).

POTENTIAL MARKET FOR ENERGY SAVING DEVICES IN LAUNDROMATS

The potential for waste heat recovery devices in the laundromat business is dependent partially on the future market for laundromats in the U.S., the price of energy, and on the characteristics of coin-op stores and operators.

A number of factors suggest an optimistic future for laundromats. First, in the present period of recession fewer people are able to afford home laundry equipment. One indication of this was a 24% drop in manufacturers' shipments of washing machines in 1981 ("Coin," May 1982: 25). In addition a 1981 study in the Washington D.C. area indicated that the cost of doing laundry at home has increased 59% since 1975 (assuming six loads a week) ("Coin," July 1982: 24-26). This study showed that laundromats are 25% more expensive than doing laundry at home (assuming three loads

and no transportation cost). However, if the investment in home equipment is considered with respect to an average 11 year operating life the coin-op may be an appealing alternative for many people. On the less optimistic side, a recent survey in Canada and the U.S. indicated that 5% of respondents expect to decrease their use of laundromats. Sixty-nine percent said they had never used a coin-op ("American Coin-Op," July 1982: 34).

One factor which may affect the market for laundromats is the increase in single parent families in the U.S. From 1970-1981 the number of single parent families in the U.S. doubled. This has affected the increased demand for day care, convenience goods, and services including laundromats ("Coin," August 1982: 151).

There are several major factors affecting adoption of waste heat recovery devices in laundromats. On the plus side operators are concerned about saving energy because they spend a large portion of their gross sales on utility costs. Operators in the past have stereotypically been rather unsophisticated with respect to financial management and awareness about implementing energy conservation measures. This may be changing as there are indications that more professional investors and full-time operators are running coin-op stores.

One critical factor is that 86% of all laundromats operate in rented spaces. A renter may be less likely to invest in adding energy saving devices if they have a

limited lease or if they are unable to make the necessary modifications in the building to accommodate a waste heat recovery system. There is some indication that more laundromats will be in operator owned buildings in the near future due to growing problems with renewing leases and rising rents. For both renters and owners, however, the cost of retrofitting and the space requirements for installing a waste heat recovery system may limit the market unless higher energy costs outweigh these considerations. New laundromats are potentially an excellent market for heat recovery systems because placement of equipment can be included in the initial layout.

2. COMPUTER SIMULATION

BACKGROUND

Models have been formulated for simulating the dynamic behavior of washing machines and laundromat use and for simulating the waste water heat recovery appliance. These models have been coded in FORTRAN (Appendix A) so that the performance of any particular heat exchanger design can be evaluated for any particular laundromat environment. The performance is characterized by a heat recovery fraction (f) equal to the heat recovered divided by the auxiliary heat and heat recovered. Many simulation runs have been made with different heat exchanger and laundromat parameters with the following objectives in mind:

- 1) to better understand how heat exchanger performance is related to the different parameters and combinations of parameters,
- 2) to see if a good heat exchanger design can be selected "by inspection" of the simulation results, and
- 3) if a good design can not be selected by inspection, to have at least generated points on the performance curve that can be used to select an optimal heat exchanger design by a numerical, non-linear optimization technique (the set of laundromat and heat exchanger parameters used in a given simulation run, together with the resulting heat recovery fraction, constitutes a "point" on the performance curve).

Some of the results of the simulation runs are displayed graphically and discussed in Chapter 5. All of the results are tabulated in Appendix B.

WASHING MACHINE AND LAUNDROMAT USE MODEL

Perhaps the most intractable problem that arises in the development of a waste water heat recovery appliance is the nature of laundromat usage. Some customers patronize their favorite laundromat at seemingly random times, while others prefer to wash their clothes on a particular day of the week and at a particular time of day. The number of washing machine loads varies from customer to customer and from visit to visit. Thus a laundromat use model must have both deterministic and stochastic components.

The deterministic component of (i.e. systematic) variation in load starts over the day is modeled as a series of connected ramp functions. That is, the profile is a spline function, $s(t)$, with straight line segments. The spline points are defined at each hour, on the hour. The profile used for the simulation is depicted in Figure 2.1. The spline points are listed in Table 2.1. The spline point values listed are relative values which are corrected during program initialization by the factor, m :

$$m = \frac{1}{16} \left[\frac{X(7) + X(23)}{2} + \sum_{i=8}^{22} X(i) \right]$$

The resulting spline function is given by

$$s(t) = m (t - \lfloor t \rfloor) X[\lceil t \rceil] = (\lceil t \rceil - t) X[\lfloor t \rfloor]$$

Table 2.1 Spline points used to define deterministic component of frequency of load starts over the day

Hour (i)	Relative Frequency of Load Starts $x(i)$
7	1.1
8	1.2
9	1.4
10	1.5
11	1.3
12	.9
13	.7
14	1.0
15	1.3
16	1.1
17	.8
18	.4
19	.7
20	1.2
21	1.1
22	.7
23	.3

Figure 2.1 Relative frequency of load starts (deterministic component).

where $\lfloor t \rfloor$ is time, in hours, rounded down (i.e. $\lfloor 14.57 \rfloor = 14$) and $\lceil t \rceil$ is time, in hours, rounded up (i.e. $\lceil 14.57 \rceil = 15$). Thus the spline function, $s(t)$, is standardized in the sense that

$$\frac{1}{23 - 7} \int_{7 \text{ h}}^{23 \text{ h}} s(t) dt = 1.0$$

That is, the daily average frequency of load starts is one per hour.

The stochastic (i.e. random or "chance") variation in load starts is modeled as a slightly modified discrete Poisson process. The Poisson model generally applies to random variables that can take on any positive value, such as the number of manufacturing defects per square yard of cloth, or the number of hail storms per year at a given location.

The Poisson distribution, given by

$$P(x) = \frac{e^{-q} q^x}{x!}$$

is plotted in Figure 2.2. One of the properties of this model is that if x is the rate of load starts (e.g. load starts per hour) then q is equal to the average rate of load starts (i.e. q may be estimated by $\bar{x}$; it happens also that q equals the variance of x).

A random variable with a distribution approximating the Poisson may be generated by the following algorithm

1. $x = 0$
2. $i = 0$

Figure 2.2 Poisson distribution.

3. if $U[0,1] < r$ then set $x = x + 1$
4. set $i = i + 1$
5. if $i < n$ go to 3.

where

$U[0,1]$ is a random number between 0 and 1 with a uniform distribution, and

$r = p/n$ is the probability of a load start during a very short time interval, $1/n$.

The larger n , the better the approximation to a true Poisson distribution. However, since time is discretized in the numerical simulation, and since the maximum number of load starts in a single time interval can never exceed the number of washing machines installed in the laundromat, a very large n is not necessary. With a discrete time unit of Δt , the algorithm remains the same but the definition of r becomes:

$$r = \Delta t \, p/n$$

where n is the number of machines installed (the maximum possible number of simultaneous load starts), and

$$p = \bar{a} \cdot s(t)$$

where $\bar{a}$ is the daily average of load starts per hour. The algorithm is then run for each discrete time interval (typically one half minute) rather than for each hour.

A set of parameters has been devised to describe the laundromat equipment. The laundromat modeled in this study has one type of washing machine. The washing machine characteristics and use parameters are as follows:

number of types of wash cycles -	3
fill capacity - (20 gallons)	166 lbm of water
number of fill/drain phases per cycle -	2

Washing machine and use parameters for two types of laundromats, "advanced" and "conventional" are shown in Table 2.2. Block diagrams of the "advanced" and "conventional" laundromats are shown in Figure 2.3. The distinction between the "advanced" and "conventional" laundromats will be discussed at length in Chapter 5, however the concept is important enough to warrant a preliminary explanation at this juncture.

The term "advanced" is used to describe a laundromat system in which mixing of the hottest and coldest water, to obtain water at an intermediate temperature, never occurs. The simplest possible "advanced" laundromat model, a model in which there is no intermediate temperature and, therefore, no mixing whatsoever of hot and cold water, has been simulated to allow comparison of the heat recovery fraction achievable when thermodynamic degradation is totally absent with that achievable in an otherwise comparable "conventional" system.

For the purpose of supplying parameters to the simulation model the "high" temperature for the "advanced" laundromat is taken to be the same as the "mix" temperature for the "conventional" laundromat, i.e. 90° F. The high

Table 2.2 Use parameters for the two types of simulated laundromats

Characteristics

Cycle Type*

	Cycle 1		Cycle 2		Cycle 3	
<u>"Advanced" Laundromat:</u>						
Use probability	.25		.50		.25	
**Fill/drain phase #	1	2	1	2	1	2
Fill Temperature	High	High	High	Low	Low	Low
Fill-to-drain time in minutes	10	4	10	4	10	4
Time to next fill in minutes	3	-	3	-	3	-
<u>"Conventional" Laundromat:</u>						
Use probability	.25		.50		.25	
**Fill/drain phase #	1	2	1	2	1	2
Fill Temperature	High	Mix	Mix	Low	Low	Low
Fill-to-drain time in minutes	10	4	10	4	10	4
Time to next fill in minutes	3	-	3	-	3	-

*Cycle types 1, 2 and 3 are generally designed for "white," "dark," and "delicate" fabrics, respectively.

**For the type of machine modeled, phase #1 is the wash phase and phase #2 is the rinse phase.

(a) Conventional

(b) Advanced

Figure 2.3 Block diagrams of the "conventional" and "advanced" laundromats as modeled by computer simulation.

temperature for the "conventional" laundromat is 130° F.

The low temperature for both laundromats is 50° F.

It should be noted that so long as the mix temperature is the average of the high and low temperatures, the actual values of the temperatures used in the simulations do not affect the resulting f-values. This constraint on mix temperature will generally hold true since the warm fill, in the "conventional" laundromat, is obtained by mixing equal parts of hot and cold water.

HEAT EXCHANGER MODEL

Temperature distributions in the heat exchanger are approximated by dividing the waste water side and supply side temperatures into N nodes as shown in Figure 2.4. The nomenclature used for this model is as follows:

- system variables and parameters -

A = heat exchanger surface area per node

M_s, M_w = mass (per node) of supply and waste water held up in the heat exchanger

m_s = mass of supply water passed through heat exchanger during a fill operation

m_w = mass of waste water passed through heat exchanger during a drain operation

T_s = supply water inlet temperature

$\bar{T}_s = \frac{1}{N} \sum_{i=N+1-i}^N T_i$ is the average supply side node temperature

T_w = waste water inlet temperature

$\bar{T}_w = \frac{1}{N} \sum_{i=1}^N T_i$ is the average waste water node temperature

Figure 2.4 Stratification model.

- water properties -

c_p - specific heat

k - thermal conductivity

α - thermal diffusivity

ρ or $\rho(T)$ - density at specified temperature

μ - viscosity

The governing equations for drain operations are adapted from Duffie and Beckman (2nd edition, pp. 333 and 335):

$$M_W \Delta T_i = F_i m_W (T_W - T_i) + m_i (T_{i-1} - T_i) \quad (1a)$$

for $i = 1$ to N , where

$$T_0 = T_1, \\ F_i = \begin{cases} 1 & \text{if } i = 1 \text{ and } T_W > T_i \\ 1 & \text{if } T_{i-1} \geq T_W > T_i \\ 0 & \text{otherwise} \end{cases}$$

and

$$m_i = m_W \sum_{j=1}^{i-1} F_j$$

The new storage state is then given by

$$T_i = T_i + \Delta T_i \text{ for } i = 1 \text{ to } N \quad (2)$$

Equation (1a) applies as written above unless $m_W > M_W$ in which case the new storage state is calculated in several steps as follows

1. set $x = m_W$ and $m_W = M_W$
2. evaluate (1) and (2)
3. set $x = x - M_W$
4. if $x > M_W$ go to 2. and repeat
5. set $m_W = x$
6. evaluate (1) and (2) one final time.

The governing equations for fill operations are the same except that m_s and M_s are substituted for m_w and M_w and equation (1) is simpler:

$$M_s \cdot \Delta T_i = m_s (T_{i-1} - T_i) \quad (1b)$$

for $i = 1$ to N , where

$$T_0 = T_s$$

because the flow always passes through all N nodes on the supply side of the heat exchanger. When m_s is greater than M_s the six step algorithm is executed with the substitution of m_s for m_w and M_s for M_w . Note that the supply side nodes are numbered from bottom to top while the waste side nodes are numbered from top to bottom.

The governing equations during relaxation periods are

$$\Delta T_i = (T_i - T_{N+1-i}) (1 - \exp(-UA\Delta t / (M_s + M_w)c_p))$$

$$T_i = T_i - \Delta T_i (1 - \frac{M_w}{M_s + M_w})$$

$$T_{N+1-i} = T_{N+1-i} + \Delta T_i (1 - \frac{M_s}{M_s + M_w})$$

for $i = 1$ to N , where

T_i are waste side node temperatures,

T_{N+1-i} are supply side node temperatures,

and

$$\frac{1}{U} = \frac{1}{h_w} + \frac{1}{h_s} + R_w$$

is the sum of the resistance of the heat exchanger wall with the supply and waste side boundary layer resistances. The values of h_w and h_s are computed according to the relation plotted in Figure 2.5 (Altas, 1979, p. 2.50) where x is the height above the base of the heat exchange module and the

Figure 2.5 Local heat transfer coefficients of laminar and turbulent free convection.

point of transition from laminar to turbulent free convection is based on the Raleigh number as follows:

$$X_t = \left[\frac{10^9 \mu \alpha}{g |\rho (\bar{T}_s) - \rho (\bar{T}_w)|} \right] = 1000 \left[\frac{\mu \alpha}{g |\rho (\bar{T}_s) - \rho (\bar{T}_w)|} \right]^{1/3}$$

SIMULATION PROCEDURE

The system was simulated with various combinations of laundromat and heat exchanger parameters. For example, computer runs were made with average frequencies of load starts of 5 and 10 per hour. Some runs were made with a flat load-start profile (i.e. the same number of load starts, on the average, from one hour to the next) while other runs were made with the profile described in the earlier section on the laundromat use model. Some runs were made with the "conventional" laundromat machine cycle characteristics while others used the machine cycles described for "advanced" laundromats.

Four different parameters were varied for the heat exchanger design: total hold-up; ratio of supply to waste water hold-up; thermal resistance of the heat exchanger wall; and area of the heat exchanger wall. The values used were:

Total hold-up (btu/F) - 100, 200, 400, 800, 1600, 3200

Supply/waste hold-up ratio - 1.0, 0.8, 0.5, 0.25

Wall resistance (ft²-F/btuh) - 0.013, 0.00023, 0.000021
(pvc) (steel) (copper)

Wall area (ft²) - 50, 100, 200, 400, 800

Most runs were made with seven nodes, however, some runs were made using 1, 3 and 20 nodes. In addition, some infinite NTU, 20 node runs were made to examine the effects of hold-up volume separately and to gain a better understanding of the "advanced" versus the "conventional" laundromat strategy.

In computing the surface area for heat transfer, the wall thickness of the heat exchanger tubes is assumed to be negligible. The surface areas on the supply and waste sides are considered equal even though the surface area is actually slightly less on the supply water side.

When the temperature difference between the supply and waste water sides is quite small, the Raleigh number (a measure of the ratio of buoyant to viscous forces on which the rate of free convection depends) becomes small and conduction heat transfer begins to predominate over free convection. The tube diameter (a measure of the distance over which conduction heat transfer must act) must be specified to compute the conductance on those occasions when the heat exchanger is operating in such a low Raleigh number regime. The tube diameter is converted to an equivalent distance between parallel plates. The equivalent plate separation distances used in the simulations are listed in Table 2.3.

The results of the computer simulations are presented and discussed in Chapter 5.

Table 2.3   Equivalent plate separation distances for heat exchanger tubes made of PVC, stainless steel, and copper

Tube Material	$R_{\text{wall}} \frac{\text{ft}^2\text{-F}}{\text{Btu h}}$	$d_s(\text{ft})$	$d_w(\text{ft})$
PVC (2")	.013	.158	.3
S. Steel (1")	.00023	.07125	.15
Copper (3/4")	.000021	.057	.1

d_s is the distance between the walls of the supply side

d_w is the distance between the walls of the waste side

3. DESIGN

MATERIALS

Materials selected for use in a waste water heat recovery system must be able to withstand the corrosive nature of laundromat waste water and provide flexibility for the construction of individually sized installations. Polypropylene plastic was found to be well suited for this use.

Initial consideration was given to metals. Steel is fairly easy to fabricate but easily corrodes requiring a coating which needs to be reapplied over time. Stainless steel has excellent corrosion resistance but is high in cost and difficult to machine. Aluminum corrodes very easily. Copper is expensive.

Non-metal materials were also considered. Wood (redwood, cedar, cyprus) has good corrosion resistance but really has to be formed in a round shape to hold water effectively. Round tanks are not very practical for laundromat owners who are usually short of storage space for bulky appliances. Fiberglass mesh and resin was initially thought to be good tank material. A prototype tank was fabricated out of fiberglass. This type of tank was rejected because of the necessity for either a mold or rigid support around it which makes it less practical for fabrication of variable sizes.

The use of plastics for tanks is relatively new (since the past 30 years). Plastics have the advantage of

- ° ease of fabrication and machining
- ° light weight
- ° good tensile strength
- ° cost effective
- ° corrosion resistant to waste water

The problem with most plastics is temperature sensitivity. Most plastics start to soften around 120-140° F (the maximum temperature of hot waste water from a laundromat). However, polypropylene plastic will withstand greater temperature and does not suffer heat distortion until over 210° F (at 66 psi). A description of physical properties for the plastics used in the exchanger can be found in Appendix C.

DESIGN

The design of the heat exchanger is shown in Figures 3.1 to 3.3. The tank walls are constructed of 0.25 inch thick polypropylene (Boltaron 5500). Rectangular polypropylene tanks are usually supported from the inside by welding plastic ribbing around the inside perimeter of the tank. Further support is usually added by cross pieces (in tension) attached at the tank sides and running through the middle of the tank. The inside support system would not work in this case since a fairly large exchanger element needed to be inserted inside the tank. Also, the internal support system would hamper cleaning of the tank which is not a problem in solar water storage tanks where clean water is the storage medium. External plastic ribbing support

Figure 3.1 Top view of exchanger.

Figure 3.2 Cut-away view of exchanger showing heat exchanger element.

Figure 3.3 Exchanger element details.

could have been used but would have occupied considerably more space. Consequently, an external steel angle support system was designed to reduce bulging in the walls of the tank similar to bands in a wood tank but rectangular as shown in Figure 3.4. The steel angle was sized and placed in relation to the height of the tank and calculated to minimize the deflection of the 0.25 inch thick polypropylene to 0.5 inches. The steel angle iron was spaced vertically with continuously threaded 0.375 inch diameter bolts. This design allowed for use of the steel support to hold the plastic (much like a form) during manufacture. Threaded bulkhead fittings were designed for use in the tank wall to allow easy disassembly of the parts and removal of the heat exchanger element.

The original design of the heat exchanger element (the heat exchanger surface inside the tank) was a roll-bond type solar collector absorber plate to be used as the heat exchange surface. This design was changed to a plastic heat exchanger element since the roll-bond product ceased to be manufactured this year, the use of a plastic exchanger element design avoids the corrosion problem associated with metal solar absorber plates and is lower in cost. Polyvinylchloride (PVC) pipe was chosen due to its availability and ease of fabrication. The PVC heat exchanger element is shown in Figures 3.5, 3.6, and 3.7. The spiral loop design was chosen to achieve a maximum hold-up volume and surface

Figure 3.4 Steel frame used as mold for plastic tank walls.

Figure 3.5 Assembled heat exchanger element banded for gluing.

Figure 3.6 Hydrostatic test of heat exchanger element.

Figure 3.7 Placing heat exchanger element in tank.

area for the cold water side of the exchanger relative to the tank hold-up volume.

FABRICATION

A steel support system was manufactured from 1.0 inch steel angle and spaced with the 0.375 inch diameter continuous bolts: Figures 3.2 and 3.8. This open steel frame was steel welded at every other corner in a right angle jig to assure squareness when used as a support for plastic welding the tank sides. The unwelded corners allowed for ease of assembly and disassembly which would be of greater concern with larger tanks. The completed steel support is shown in Figure 3.8. The plastic tank corners were first tack welded (plastic) on the inside to provide temporary seam integrity. The outside corners were beveled inward with a router to 45 degrees. Joining the two pieces of plastic (each with a 45 degree cut) formed a 90 degree channel for welding. The channel was heated to 500-550° F with an electric welding torch while molten 0.15625 diameter polypropylene welding rod was heated and fit to the channel (Figures 3.9 and 3.10). Three to five passes were made in each channel with the welding rod to form a contiguous bond.

To assure structural integrity of the tank the tank was filled with 130° F water and allowed to sit for 4 hours without the steel support. Without the steel the tank started assuming a rounded shape as shown in Figure 3.11.

The heat exchanger element was fabricated from polyvinylchloride (PVC) piping material. The completed

Figure 3.8 Steel support with three tank sides in place.

Figure 3.9 Welding heat exchanger element support.

Figure 3.10 Using plastic welder to weld tank cover.

Figure 3.11 Testing structural integrity of tank without steel support structure.

exchanger element is shown in Figure 3.5 held together by a nylon band. PVC glue was used to hold each section of the element to the others. The band provided the pressure to assure a good bond. The heat exchanger element was then hydrostatically tested at 70 psi for 4 hours as shown in Figure 3.6. Water was used for the pressure test since the potential energy of compressed air would cause an explosion if the pressure test failed.

The heat exchanger appliance was then completed by the addition of the bulkhead fittings, drain pipe, diffuser, and insertion of the exchanger element (Figure 3.12). Plastic 1.50 inch I.D. bulkhead fittings were used for the drain and diffuser pipe. This method allowed for easy disassembly of the exchanger and simple pipe connections on the outside of the tank for the waste water input and output. Holes were made in the tank for the exchanger element bulkhead fittings. Figure 3.7 shows the exchanger element being lowered into the tank. The two holes for the exchanger element bulkhead fittings are visible at the bottom of Figure 3.7. The exchanger element was held in the tank by element supports (Figure 3.2) which were plastic welded to the sides of the tank. The use of the element supports allowed a 6.0 inch clear channel underneath the exchanger element for low temperature water to flow from the bottom of the flow diffuser to the drain pipe inlet (located near the bottom of the tank).

Figure 3.12 Heat exchanger assembly.

For test purposes, thermocouples were placed at the locations (designated "TC") shown in the sketch in Figure 3.13 and 3.1. Flexible clear polyvinyl hose was used to connect the heat exchanger element to the 0.75 inch I.D. bulkhead fittings. This method allowed placement of thermocouples TC3 and TC2 (Figure 3.13) inside of the cold water inlet and heated water outlet pipes. Two other thermocouples, TC4 and TC5, were placed at the top of the tank (on the flow diffuser) and bottom (on the 2.0 inch diameter drain pipe). The completed heat exchanger is shown in Figure 3.14 with the four internal thermocouples attached and the heat exchanger element in place. A polypropylene cover was fabricated to fit over the top to reduce heat loss from the tank due to evaporation. The cover is shown during initial testing resting against the wall to the right of the exchanger in Figure 3.15.

The exchanger design and fabrication has been described in this chapter. Use of plastics allowed for a design which can be easily adapted for laundromat installations requiring different sized tanks and heat exchanger elements. The use of plastics also minimizes corrosion of the components, a common problem with exchangers constructed out of metal.

Figure 3.13 Heat exchanger test set-up.

Figure 3.14 Heat exchanger with thermocouples attached.

Figure 3.15 Heat exchanger test.

4. TESTING

Various experimental tests were performed on the heat exchanger design described in Chapter 3. First, the parameters for the exchanger (e.g. volumes and surface areas for the exchanger element) were calculated for use in comparing theoretical and experimental results. Next, tests to determine the heat transfer coefficients (R-values) for the exterior of the tank and the heat exchanger element were performed and compared to theoretical results.

EVALUATION OF PARAMETERS

The heat exchanger element was assembled with polyvinylchloride (PVC) pipe as shown in Figures 3.1, 3.2, 3.3, and 3.7 (Chapter 3). Standard one inch nominal class 200 (200 psi rated at 70° F) was joined with schedule 40 one inch PVC fittings. The components were assembled using PVC cleaner and glue. The configuration illustrated by Figure 3.2 was used to optimize space in the exchanger and attempt to get the maximum internal volume and surface area. The internal hold-up volume was figured as follows:

$$\text{Internal Area of Tube} = \frac{(\pi)D^2}{4} = \frac{(\pi)(1.169)^2}{4} = 1.073 \text{ sq. in.}$$

$$\text{Internal Area of Elbows} = (1.1 \text{ in.} \times 2) \times 58 \text{ elbows} = 128 \text{ in.}$$

$$128 \text{ in.} \times \text{Area} = 137 \text{ cu. in.}$$

$$\text{Internal Area of Tubes} = (14+15 \text{ tubes}) \times 24 \text{ in. per tube} = 696 \text{ in.}$$

$$696 \times \text{Area} = 747 \text{ cu. in.}$$

Waste water volume (with exchanger in):

$$\begin{aligned}\text{TANK} &= 21.5 \times 27.5 \times 30.125 = 1781 \text{ cu. in.} \\ &= 10.3 \text{ cu. ft.} \\ &= 77.1 \text{ gallons}\end{aligned}$$

External volume of heat exchanger element:

$$\begin{aligned}\text{Tubes} &= 29 \text{ tubes} \times (24 - 2.2) = 632 \text{ in.} \\ &= 632 \text{ in} \times \frac{(\pi)D^2}{4} = (632)(\pi)(1.315)^2/4 = 858 \text{ cu. in.}\end{aligned}$$

Elbows

Outside volume of one elbow

$$\frac{(2.65)(\pi)(1.65)^2}{4} + \frac{(1.0)(\pi)(1.65)^2}{4} = 7.80 \text{ cu. in.}$$

$$58 \text{ elbows} \times 7.80 = 453 \text{ cu. in.}$$

Total external volume

$$\begin{aligned}453 + 858 &= 1311 \text{ cu. in.} \\ &= 0.758 \text{ cu. ft.} \\ &= 5.67 \text{ gallons}\end{aligned}$$

Total internal volume of the heat exchanger element is:

$$\begin{aligned}(747 + 137) &= (884 \text{ cu. in.})/1728 \text{ cu. in./cu. ft.} \\ &= 0.5116 \text{ cu.ft.} \times 7.48 \text{ gal./cu. ft.} \\ &= 3.83 \text{ gallons}\end{aligned}$$

The external surface area of the heat exchanger element is:

$$\begin{aligned}\text{TUBES} &= 29 \text{ tubes} \times (24 - 2.2) = 632.2 \text{ inches} \\ &= 632.2 \text{ in.} \times (\pi)(1.315) = 2612 \text{ sq. in.}\end{aligned}$$

Surface area of one elbow

$$(2.65)(\pi)(1.65) + (1.0)(\pi)(1.65) = 18.92 \text{ sq. in.}$$

$$\text{ELBOWS} = 58 \text{ elbows} \times 18.92 \text{ sq. in.} = 1097 \text{ sq. in.}$$

$$\text{Total outside surface area} = 1097 + 2612$$

$$= 3709 \text{ sq. in.}$$

$$= 25.76 \text{ sq. ft.}$$

The internal surface area is:

$$\text{TUBES} = (29 \text{ tubes})(24 \text{ in. per tube})(1.169)(\pi) = 2556 \text{ sq. in.}$$

$$\text{ELBOWS} = (58 \text{ elbows})(1.1)(2)(1.169)(\pi) = 469 \text{ sq. in.}$$

$$\text{Total inside surface area} = 3025 \text{ sq. in.}$$

$$= 21.0 \text{ sq. ft.}$$

The external volume of heat exchanger element was also calculated experimentally by displacement of water.

$$\text{Water displaced} = 2.125 \times 21.5 \times 27.5 = 0.727 \text{ ft}^3$$

$$= 5.44 \text{ gallons}$$

which compares favorably with the 5.67 gallons found theoretically.

Heat exchanger element parameters summary:

	<u>INTERNAL</u>	<u>EXTERNAL</u>
VOLUME	0.5116 cu. ft. (3.83 gal)	0.758 cu. ft. (5.67 gal)
SURFACE AREA	21.0 sq. ft.	25.76 sq. ft.

The average thickness of the exchanger element wall can be calculated:

$$\text{TUBES} = 2612 \text{ sq. in.} \quad \frac{2612}{3709} = 0.70$$

$$\text{ELBOWS} = 1097 \text{ sq. in.} \quad \frac{1097}{3709} = 0.30$$

$$\text{TOTAL} = 3709 \text{ sq. in.}$$

AVERAGE EXCHANGER ELEMENT WALL THICKNESS =

$$[(1.65 - 1.169) \times 0.30 + (1.315 - 1.169) \times .70]/2 = 0.123 \text{ in.}$$

FLOW DIFFUSER

Shown in Figure 4.1 is the apparatus test set-up to test the flow diffuser. A one inch pipe entered the tank and connected to the diffuser, Figure 4.1 and 4.2(a). The pipe was machined with 0.3125 inch diameter holes spaced every inch with alternating pairs of holes oriented at 180 degrees to preceding holes. The total of 60 holes yielded an area of 4.6 square inches and another 0.785 square inches for the open end of the diffuser for a total of 5.385 square inches of diffuser open area. The outlet drain shown in Figure 4.1 was 2.0 inches in diameter and arranged so that the level of the water in the tank remained at 31 inches in depth. Type T thermocouples were constructed and placed in a one inch diameter PVC pipe as shown in Figure 4.2(b). A total of 8 thermocouples were installed every four inches starting two inches from the bottom of the tank.

Results of the first test are tabulated in Figure 4.3 and plotted in Figure 4.4. The initial starting condition was State 1 with an average temperature of 71.75° F as shown in Figure 4.3. The temperature profile is shown in Figure 4.4. The tank was well mixed except for the top two nodes thermocouples which indicated a slightly higher

Figure 4.1 Test set-up showing flow diffuser.

Figure 4.2a Flow diffuser cross section.

Figure 4.2b Attaching thermocouples to holder for diffuser/
stratification tests.

TANK

NODE #	T ₁ (F)	T ₂ (F)	T ₃ (F)	T ₄ (F)
8	77	110	108	98
7	73	110	104	98
6	71	108	100	98
5	70	103	97	97
4	71	92	91	96
3	70	75	79	92
2	71	71	67	85
1	71	71	59	80
T _{AVE} (F)	71.75	92.5	88.1	93
E _b , Energy (kBtu)	35.7	41.0	39.0	41.1
ΔE (E _{out} - E _{in})		9.3	-2.0	2.1

Flow

T _{in}	130	48	85
T _{out}	70	63	69
ΔT (T _{out} - T _{in})	-60	+15	-16
Rate (gpm)	398	327	306
gpm	26	3.1	3.4
E _f , Energy (kBtu)	-8.6	2.1	-2.2

ENERGY BALANCE

E _f + E _b (kBtu)	1.7	1.1	1.1
2 E _f + E _b / (E _f + E _b)	8%	5%	5%

MISSTank

$$\text{Volume of Tank} = \frac{(22.375 \text{ in})^2}{4} \pi \times 31 = 7.05 \text{ ft}^3 \times 7.48 \frac{\text{gal}}{\text{ft}^3} = 52.76 \text{ gal} \sim 53 \text{ gal}$$

$$\text{mass} = 8.34 \frac{\text{lb}}{\text{gal}} \times 53 \text{ gal} = 442 \text{ lbs}$$

$$C_p = 1.0 \frac{\text{Btu}}{\text{lb} \cdot \text{F}}; \text{ Energy} = E = m C_p T \text{ (relative to } 0^\circ \text{F)}$$

Flow

$$V_o = \frac{(15.625 \text{ in})^2}{4} \pi \times 20.65 \text{ in} = 2.294 \text{ ft}^3 \times 7.48 = 17.13 \sim 17.1 \text{ gal} = 143 \text{ lbm}$$

Figure 4.3 Stratification test - flow diffuser - Test 1.

Figure 4.4 Flow diffuser - Test 1.

temperature. From this initial condition, at State 1, 17.1 gallons of 130° F water were introduced through the diffuser and simultaneously withdrawn from the bottom of the tank into a holding vessel. The water in the holding vessel was well mixed and at a temperature of 70° F. The theoretical energy stored in the tank was calculated by subtracting the sensible heat energy of the water added to the tank from the energy of the water taken from the tank. As shown in Figure 4.3 the energy from the flow through the tank is calculated at -8.6 kBtu (units of thousand British Thermal Units relative to zero degrees Fahrenheit). The actual energy stored in the tank was calculated by finding the arithmetic mean tank temperature before and after the test and calculating the net energy storage by the tank. In Figure 4.3 the energy at State 1 was 31.7 kBtu and at State 2 it was 41.0 kBtu for a difference of 9.3 kBtu. Theoretically the energy into the system is equal to the energy out of the system. For the first test the sum of the energy in and energy out was 0.7 kBtu as shown in Figure 4.3. This represents an experimental error of about 8%.

The temperature distribution of the water in the tank after adding the 17.1 gallons of 130° F water is shown in Figure 4.4 as State 2 (2 with a circle around it). As indicated by the change of the temperature distribution of the tank from State 1 to State 2, the majority of the energy was stored in the top part of the tank (from 14 inches to 30

inches). Little change in temperature occurred at the bottom of the tank.

The results of the next test (from State 2 to State 3) are also plotted in Figure 4.4. For this test 48° F water was added. Stratification in the tank was still maintained with a nearly linear temperature profile from the bottom to the top of the tank (60° to 110° F, difference of 50° F). Water at 85° F was then added which significantly decreased the stratification (as would be expected) to 18° F. Further tests were performed and are tabulated in Figure 4.5 and plotted in Figures 4.6 and 4.7.

SITE TEST

An apparatus was constructed in a laundromat to determine characteristic waste water temperature distribution over time. A 4 inch drain pipe from 20 Maytag A17 and A18 washing machines emptied into an 18 inch diameter by 26 inch high tank, 28.6 gallons, shown in Figures 4.8(a) and 4.8(b). A submersible sump pump (48.3 gpm) pumped water from the machines to the diffuser. The diffuser was 4 inches in diameter with approximately 100 (0.3125 inch) diameter holes machined in the wall of the diffuser for a total flow through area of 20.3 square inches (7.7 sq. in. for holes and 12.6 sq. in. for the end of the diffuser, which was about 2.0 inches from the bottom of the tank). If the flow rate from the machines exceeded the flow rate capacity of the sump pump an overflow was provided to another container which also received the return from the

TANK

NODE # (°F)	T ₁	T ₂	T ₃	T ₄	T ₅	T ₆	T ₇	T ₈	T ₉
8	73	101	98	95	96	114	111	102	120
7	73	100	95	95		114	108	100	120
6	73	100	94	95		113	106	98	119
5	72	99	92	94		112	104	97	116
4	70	97	90	93		112	102	96	102
3	70	94	88	92		110	100	95	97
2	70	92	78	92		108	96	79	96
1	70	86	69	91		105	75	67	95
T _{avg} (°F)	71.4	96.1	88	93.4	96	111.0	100.0	91.8	108.1
MASS (lbs)	442	442	442	442	442	442	442	442	442
Energy (kbtu)	31.6	42.5	38.9	41.3	42.4	49.1	44.2	40.6	47.8
E _t ΔE (E _{i+1} - E _i)		10.9	-3.6	2.4	1.1	6.7	-4.9	-3.6	7.2

FLOW

T _{IN}	141	48	102	101	142	48	49	134
T _{OUT}	72	74	82	94	97	83	70	77
ΔT (T _{OUT} - T _{IN})	-69	26	-20	-7	-45	35	21	-57
Time (s)	179	182	173	153	165	180	1020	1156
Vol (gal)	18.6	17.8	17.8	17.8	17.8	17.8	14.3	17.0
- gpm	6.3	5.9	6.2	7.0	6.5	5.9	8.4	8.8
MASS (lbs)	156	149	149	149	149	149	119	142
E _f ENERGY (kbtu)	-10.7	+3.9	-2.9	-1.0	-6.7	5.2	2.5	-8.1

ENERGY BALANCE

E _f + E _t	.2	.3	.5	.1	0	.3	1.1	.9

MISC

Volume of Tank = 53 gal = 442 lbm

$$T_{amb} \approx 73^{\circ}\text{F}$$

Figure 4.5 Flow diffuser - Test 2.

Figure 4.6 Flow diffuser - Test 2.

Figure 4.7 Flow diffuser - Test 2 (continued).

Figure 4.8a Laundry site test set-up.

Figure 4.8b Drainage system at test laundromat

tank via a 4 inch diameter pipe. Type T thermocouples were constructed in the same arrangement as in the flow diffuser tests, Figures 4.1 and 4.2(b).

Results of the site tests are plotted in Figures 4.9 and 4.10. The small arrows labeled 'white,' 'colors' refer to machine settings at the time the machines were turned on. The dashed line represents the temperature of the inlet water to the diffuser. The two solid lines represent the temperatures at the bottom (node 1) and the top (node 8) of the tank measured by the thermocouples. As noted in Figures 4.9 and 4.10 the inlet temperatures vary from about 60° to 120° F, while the temperature of the tank oscillated around 85° F.

As indicated by the floor diffuser tests a varying degree of thermal stratification is possible depending upon the design of the diffuser, flow rates, and temperatures. While the diffuser does help stratify the water in the tank a more important benefit is the dumping of cold water directly to the drain pipe along the bottom of the tank. For example, if the waste water in the tank is stratified and cold rinse water exits from some machine to the heat exchanger not only should the cold water exit out the drain but additionally the intrusion of the cold water should not upset the existing thermal stratification in the tank. This "switching" action is illustrated in Figure 4.9 at about 17 minutes. The cold water inlet dropped from over 90° F to a little above 60° F. However, while the temperature at the

Figure 4.9 Plot of results of Site Test 1.

Figure 4.10 Plot of results of Site Test 2.

bottom of the tank fell to the inlet temperature (which would be expected) the temperature at the top of the tank did not drop below about 73° F. In the same Figure (4.9) the effect of the diffuser in thermal stratification is illustrated from about 27 to 35 minutes. The inlet water temperature increased, followed by a rise in the temperature at the top of the tank. Therefore, the use of a diffuser not only helps to maintain thermal stratification in the tank but helps prevent tank mixing when cold water is introduced to the tank.

OVERALL HEAT LOSS COEFFICIENT

The heat exchanger tank, when full of water at a temperature greater than the surrounding environment temperature, will lose heat to the environment consequently lowering the temperature of the water in the tank. The rate that the temperature in the tank declines is a function of the environment temperature, temperature and thermal capacitance of the water in the tank, and the thermal properties of the tank material (overall heat transfer coefficient of the tank). This test was to find the value for the overall heat transfer coefficient of the tank. The water in the tank was initially at temperature T_0 and the surrounding air temperature at T_a . The change of the internal temperature of the tank is proportional to the net heat flow from the tank to the environment.

$$-CdT = UA (T - T_a)dt$$

where C = thermal energy stored in the tank (Btu/F)

T = average temperature of the tank (F)

U = overall heat loss coefficient (Btu/sq. ft. F)

A = surface area of the tank (sq. ft.)

T_a = environment temperature (F)

t = time (hours)

Introducing the initial temperature as T_0 and limits of a temperature T at time t the above equation can be integrated.

$$\ln \left[\frac{T - T_a}{T_0 - T_a} \right] = \frac{-UAt}{C}$$

Therefore,

$$U = -\ln \left[\frac{T - T_a}{T_0 - T_a} \right] \frac{C}{At}.$$

The following experimental results were recorded and are shown in Table 4.1.

A plot of the temperature history for the tank on a log-linear scale is shown in Figure 4.11. Using numerical values obtained in time Steps I and II,

$$U = -\ln \left[\frac{103.4 - 72.5}{129.7 - 72.5} \right] \frac{C}{A (12.17)}$$

$$U = 0.0506 \frac{C}{A}$$

where $C = (600 \text{ lbm}) \left(1.0 \frac{\text{Btu}}{\text{lbmF}} \right) = 600 \text{ Btu/F}$

$$A = 34 \text{ ft}^2$$

Table 4.1 Experimental test results to determine overall heat transfer coefficient of tank

Step	t (hours)	T _{top} (F)	T _{bot} (F)	T (F)	T _a (F)	T-T _a (F)
1	0.00	129.9	129.5	129.7	70.5	59.2
2	1.30	126.1	124.9	125.5	73.1	52.4
3	1.42	125.7	124.9	125.5	72.3	53.0
4	1.53	125.3	124.0	125.3	72.3	52.4
5	2.15	124.4	122.8	124.7	71.4	52.2
6	2.53	123.6	121.5	123.6	71.8	50.8
7	4.05	119.4	117.3	118.4	72.7	45.7
8	10.80	106.6	103.6	105.1	73.1	32.0
9	11.25	106.2	102.8	104.5	73.1	31.4
10	11.88	104.5	102.4	103.5	73.6	30.0
11	12.17	104.9	101.9	103.4	73.6	29.8

average T_a = 72.5 F

Figure 4.11 Average internal temperature of tank over time - UA test.

$$U = \frac{0.0506 (600)}{34} = 0.89 \text{ Btu/hr-sq.ft.-F}$$

$$UA = (0.89)(34) = 26.7 \text{ Btu/hr-F}$$

The R-value can be calculated from the U value by taking the reciprocal.

$$R = \frac{1}{U} = 1.12 \text{ hr-sq.ft.-F/Btu}$$

The manufacturer's heat transfer specification for the tank wall material (Boltran polypropylene 5500 0.25 inches thick) is listed as 0.83 Btu/hr-sq.ft.-F/in. Dividing by the thickness gives 3.32 Btu/hr-sq.ft.-F (an R-value of 0.301 hr-sq.ft.-F/Btu). Thus, the experimental R-value is about four times the manufacturer's R-value for the material alone. The manufacturer's R-value is less since it does not account for the boundary layer film resistance of the water on the inside, the air on the outside, and the thermal radiation losses of the tank. The thermal resistance of the tank walls can be easily increased by the addition of two inch thick styrofoam to the outside of the tank walls.

HEAT EXCHANGER EXPERIMENTAL TESTS

Experimental tests were run to determine the heat transfer coefficient of the heat exchanger element wall. Initial tests were run in a laundromat (Figure 4.12) but difficulty in establishing constant conditions and known flow rates led to testing in a more controlled environment. The tank was filled with hot water (77 gallons) and water

Figure 4.12 Heat exchanger in laundromat utility room.

was allowed to flow through the interior of the exchanger element at a constant flow rate. The results are tabulated in Tables 4.2 to 4.7. From Table 4.7 the overall heat transfer coefficient (conductance, U) is calculated to be 28.0 Btu/hr-sq.ft.-F which corresponds to an R-value of 0.036 hr-sq.ft.-F/Btu. The R-value for 2.0 inch diameter schedule 40 pipe (thickness = 0.164 inches) is 0.013 hr-sq.ft.-F/Btu (from Chapter 2). The average thickness of the exchanger tube wall is 0.123 in. (from the first section in this chapter). Therefore, the theoretical thermal resistance for class 200 1.0 diameter pipe is 0.0098 hr-sq.ft.-F/Btu ($0.013 \times 0.123 / 0.164$). Subtracting the theoretical resistance from the experimental value yields the boundary layer resistance of $(0.036 - 0.0098) = 0.026$ hr-sq.ft.-F/Btu compared to a theoretical boundary layer resistance of 0.020 hr-sq.ft.-F/Btu. Therefore, most of the resistance to heat transfer is due to the water boundary layer film on the inside and outside of the heat exchanger element as noted by the simulation results in Chapter 2.

In summary, the heat transfer resistance (R-values) was calculated for the tank wall and the heat exchanger element wall. The experimental R-value for the tank wall (heat loss from the inside of the tank to the surrounding environment) was measured to be 1.12 hr-sq.ft.-F/Btu. The theoretical wall material R-value was calculated to be 0.301 hr-sq.ft.-F/Btu. The difference is due to the boundary layer resistances. The experimental steady state UA value

Table 4.2 Nomenclature to Tables 4.3 to 4.8

TIME	HOURS MINUTES.SECONDS
TC1	Ambient Temperature surrounding tank, about 12 inches from tank, 18" from bottom of tank
TC2	Inside 3/4 inch I.D. Tube heat exchanger element input (cold water in)
TC3	Inside 3/4 inch I.D. Tube heat exchanger element output (heated water out)
TC4	Inside 1" PVC diffuser pipe 28" from bottom of tank
TC5	Inside 2" PVC drain pipe 5" from bottom of tank
t	Time in seconds
T _{tank}	Average temperature of tank (TC4 + TC5) / 2

Note: All thermocouple readings are in degrees F.

Table 4.3 Heat exchanger element heat transfer Test No. 1

TEST 1

<u>TIME</u>	<u>TC1</u>	<u>TC2</u>	<u>TC3</u>	<u>TC4</u>	<u>TC5</u>	<u>t</u>	<u>Ttank</u>
0456.06	68.3	49.8	66.0	135.3	132.4	0	133.9
0458.00	68.7	50.3	74.9	134.9	131.1	114	133.0
0459.12	<u>69.1</u>	<u>51.2</u>	<u>80.6</u>	<u>134.1</u>	<u>130.3</u>	<u>186</u>	<u>132.4</u>
Averages	68.7	50.4	73.8	134.8	131.3		133.1

$$\Delta T = (73.8 - 50.4) = 23.4 \text{ F}$$

$$\text{Flow Rate} = 4/186 = 1.29 \text{ gpm}$$

$$\begin{aligned} \text{Heat Exchanged} &= (4.0)(8.34)(3600.)(23.4)/186. = 15108 \text{ Btuh} \\ &= 15108/(25.76^*)(23.4) = 25.06 \text{ Btuh/sq.ft.-F} \end{aligned}$$

*External heat exchanger element surface area

Table 4.4 Heat exchange element heat transfer Test No. 2

TEST 2

<u>TIME</u>	<u>TC1</u>	<u>TC2</u>	<u>TC3</u>	<u>TC4</u>	<u>TC5</u>	<u>t</u>	<u>Ttank</u>
0505.20	73.1	52.1	81.1	131.1	127.4	0	129.3
0507.15	74.5	52.1	80.2	130.7	126.5	115	128.6
0508.30	<u>74.5</u>	<u>52.1</u>	<u>80.2</u>	<u>130.3</u>	<u>126.1</u>	<u>190</u>	<u>128.2</u>
	74.0	52.1	80.5	130.7	126.7		128.7

$$\Delta T = (80.5 - 52.1) = 28.4 \text{ F}$$

$$\text{Flow Rate} = (4)(60)/(190) = 1.26 \text{ gpm}$$

$$\begin{aligned} \text{Heat Exchanged} &= (4.0)(8.34)(3600)(28.4)/(190.) = 17951 \text{ Btuh} \\ &= (17951)/(25.76)(23.4) = 29.8 \text{ Btuh/sq.ft.-F} \end{aligned}$$

Table 4.5 Heat exchanger element heat transfer Test No. 3

TEST 3

<u>TIME</u>	<u>TC1</u>	<u>TC2</u>	<u>TC3</u>	<u>TC4</u>	<u>TC5</u>	<u>t</u>	<u>Ttank</u>
0517.10	70.5	53.5	79.3	127.0	122.8	0	124.9
0519.18	71.8	53.9	79.3	126.5	121.5	128	124.0
0520.19	<u>72.7</u>	<u>53.5</u>	<u>79.3</u>	<u>126.1</u>	<u>121.5</u>	<u>189</u>	<u>123.8</u>
	71.7	53.6	79.3	126.5	121.9		124.2

$$\text{delta T} = (79.3 - 53.6) = 25.7 \text{ F}$$

$$\text{Flow Rate} = (4)(60)/(189) = 1.30 \text{ gpm}$$

$$\text{Heat Exchanged} = (4)(8.34)(3600)(25.7)/(189) = 16331 \text{ Btuh}$$

$$= (16331.)/(25.7)(25.76) = 24.7 \text{ Btuh/sq.ft.-F}$$

Table 4.6 Heat exchanger element heat transfer Test No. 4

TEST 4

<u>TIME</u>	<u>TC1</u>	<u>TC2</u>	<u>TC3</u>	<u>TC4</u>	<u>TC5</u>	<u>t</u>	<u>Ttank</u>
0527.13		51.2	77.6	123.6	117.7	0	120.65
0528.25		51.2	77.1	123.2	117.3	128	120.25
0529.40	<u>68.3</u>	<u>52.1</u>	<u>76.2</u>	<u>122.8</u>	<u>116.8</u>	<u>147</u>	<u>119.8</u>
Averages	68.3	51.5	77.0	123.2	117.3		120.2

$$\text{delta T} = (77.0 - 51.5) = 25.5 \text{ F}$$

$$\text{Flow Rate} = 2.0 \text{ gpm}$$

$$\text{Heat Exchanged} = 20833 \text{ Btuh}$$

$$= 31.7 \text{ Btuh/sq.ft.-F}$$

Table 4.7 Heat exchanger element heat transfer test summary

SUMMARY TESTS 1-4

<u>TEST NO.</u>	<u>START</u>	<u>Energy (Btuh)</u>	<u>t (s)</u>	<u>T_{tank} (F)</u>	<u>Energy Btu</u>	<u>Energy Btuh sq.ft.-F</u>
1	0456.06	15108	0	133.1		25.1
2	0505.20	17951	554	128.7	2325	29.8
3	0517.10	16331	710	124.4	3540	24.7
4	0527.13	20833	603	120.2	2735	31.7
Stop	0529.40		<u>147</u>	119.8	<u>851</u>	<u>28.8</u>
			2014s		9451Btu	28.0
			TOTAL		TOTAL	AVERAGE

Change of Energy in the Tank

$$(77.1 \text{ gal})(8.34 \frac{\text{lbm}}{\text{gal}})(1 \frac{\text{Btu}}{\text{lbmF}})(133.1 - 119.8) = 8552 \text{ Btu}$$

energy losses to environment

$$Q = UA\Delta T = (26.7 \frac{\text{Btu}}{\text{hr-F}})(126.5 - 70.0)(\frac{2014\text{s}}{3600}) = - 844 \text{ Btu}$$

$$\text{Total change of energy in Tank} = \underline{\hspace{2cm}} = 7708 \text{ Btu}$$

$$\text{Experimental error} = (9451-7708)/9451 = 18\%$$

is thus 26.7 Btu/hr-F. The overall R-value for the heat resistance for the heat exchanger element wall was found experimentally to be 0.036 hr-sq.ft.-F/Btu compared to the theoretical material R-value of 0.0098 hr-sq.ft.-F/Btu. The difference between the theoretical R-value for the material and the experimental overall R-value is attributed to the boundary layer resistance on both sides of the wall (0.026 hr-sq.ft.-F/Btu).

5. CONCLUSIONS

An overview of the laundromat industry indicates that coin-op store owners are increasingly concerned with saving energy. Nevertheless, it is difficult to predict the extent to which waste water heat exchangers will be adopted by laundromat owners in the near future. There is a tremendous potential for energy savings, however, there are some hinderances to widespread adoption of heat exchangers.

In existing laundromats there are several less costly conservation measures that can be taken to reduce energy bills. It is frequently most efficient for owners to implement these basic measures before an investment is made in a heat exchanger. One measure which directly affects the market for heat exchangers is the reduction of hot water consumption by offering cold water wash and rinse. In recent years detergents have been developed for use in cold water as well as hot wash water. Laundromats which offer some cold water only machines appear to be acceptable to patrons (Coin-Op, August 1982:20), but it is unlikely that hot water washes (for diapers, etc.) will be completely eliminated. Both new and existing laundromat owners are installing more energy efficient equipment when new machines are purchased. Systems aimed at utilizing available waste heat and solar energy are already being installed by laundromat owners which indicates that there is a market for heat recovery devices in laundromats.

This chapter discusses the main results of the research on heat exchanger design, fabrication and simulated use. The basic conclusions are listed below.

1. The potential for energy recovery is significantly reduced whenever hot and cold water is mixed by individual washing machines for a "warm fill."

2. Cold water wash can be substituted for a majority of machines in a laundromat if economic incentives are provided, e.g. the cold water wash or warm wash/cold rinse machines should cost less.

3. Plastic is well suited for fabrication of heat exchanger elements.

4. The use of polyvinylchloride plastic for a heat exchanger element results in a loss of the heat recovery fraction of only 5 to 8 percentage points as compared to using copper or stainless steel.

5. The minimum hold-up volume on the waste water side of the exchanger should be at least six times the hold-up volume of a single washing machine. Heat exchanger surface area, in square feet, should be at least twice the average hot water flow rate in gallons per hour.

6. Heat exchanger appliances are most readily installed in new laundromats or existing laundromats which have sufficient floor space and a layout which can accommodate the system.

7. Life cycle costing analysis should be used to determine the economic payback.

8. The performance of the heat recovery appliance can be improved by combining with a solar system and/or heat pump.

Discussion of these conclusions and the results they are based on appears in the following sections.

COMPUTER SIMULATION RESULTS

The purpose of the first simulation was to investigate the sensitivity of heat recovery fraction to total hold-up volume and hold-up volume ratio. However, and more importantly, the exercise also led to further thought about the thermodynamics of laundromat operations.

The thermodynamic effect of concern here is the mixing of hot and cold water by individual washing machines whenever a "warm fill" is called for. A simple example will serve to illustrate that while no energy is lost in this process, a lower thermodynamic state results which reduces the potential for energy recovery.

Consider 100 pounds of waste water at 150° F and 100 pounds at 50° F. Taking 50° F as the zero reference state, the hot water represents 10,000 Btu and the cold water 0 Btu. If the hot and cold water are passed in separate batches through a hypothetical ideal heat exchanger (an infinite NTU ("number of transfer units," see Appendix A), counter-flow heat exchanger with perfect stratification) it is possible to heat 100 pounds of water from 50 to 150° F on the supply side of the heat exchanger. If, however, the

waste water is first mixed to obtain 200 pounds of 100° F water (still 10,000 Btu) the heat exchanger cannot heat the supply side water above 100° F. In one sense 100% heat recovery is still possible because 200 pounds of supply water could be heated to 100° F (recovering the full 10,000 Btu). However, if we consider a hypothetical laundromat in which all fills are warm fills obtained by mixing 100 pounds of hot with 100 pounds of cold water at the individual machines we see that the hot water requirement is for 100 pounds of 150° F water per fill, not 200 pounds of 100° F water. Regardless of whether or not we heat 100 or 200 pounds of water to 100° F it is still necessary to further heat 100 pounds of that water to 150° F which requires an additional 5,000 Btu per fill. Moreover, there is no use for more than 100 pounds of hot water per fill so even though it is thermodynamically possible to heat 200 pounds per fill to a temperature of 100° F the additional 5,000 Btu cannot be considered recovered energy because there is no use for it. Thus the potential for heat recovery by a heat exchanger (heat pumps are a different story) is halved when the heated water required for the application is mixed in equal parts with unheated water, resulting in waste water (the source for heat recovery) that is at an intermediate temperature.

Two different load characteristics were explored using computer simulation as discussed in Chapter 2. The results of the computer runs using the two different

characteristics provided information for overall design ideas to eliminate hot and cold mixing and thus increase the energy efficiency of laundromat operation.

As discussed in Chapter 2 the conventional load characteristic consists of a hot, warm, and cold mixture of washing machines. An "advanced" load profile either separates the hot from the warm and cold running machines or only has warm and cold machine cycles. What efficiencies can be expected by using the "advanced" design? The results of the simulation runs are tabulated in Appendix B. In Tables B.1 and B.2 are tabulated the results of a hypothetical, infinite NTU heat exchanger driven by the two different load characteristics. Table B.1 results were generated using the conventional hot, warm, and cold machines. These results are plotted in Figure 5.1. The x-axis represents the combined thermal energy storage in Btu/F of the cold water supply side of the exchanger, C_s , plus the waste water side of the exchanger, C_w . The family of curves in Figure 5.1 characterizes different ratios of C_s/C_w , the thermal storage capacity of the supply side divided by the thermal storage capacity of the waste water side. The heat recovery fraction, f , is measured by the y-axis. The heat recovery fraction is defined as the recovered heat divided by the recovered heat plus auxiliary heat. An example may illustrate how to use the figure. Consider a waste water heat recovery appliance with 200 gallons of storage on the waste water side of the exchanger and 50 gallons on the cold

Figure 5.1 Infinite NTU model hot-warm-cold driving series.

water supply side. The combined thermal energy storage is about $(200 + 50) \times 8.34 = 2000$ Btu/F. The ratio of C_s to C_w is $50/200 = 0.25$. From the x-axis in Figure 5.1 the heat recovery fraction is approximately 64%. Notice that decreasing the cold water supply side hold-up volume to 25 gallons decreases the heat recovery fraction to about 56%. For this model the highest heat recovery fraction is about 66%. Increasing the ratio of the supply volume to the waste water volume much over 0.5 has little effect on the heat recovery fraction.

Using the same idealized heat exchanger (infinite NTU model) a comparison of the two load profiles can be made. Plotted in Figure 5.2 are the results of the "advanced" warm, cold load profile. Data were taken from Table B.2 in Appendix B. The most striking fact about the comparison of Figures 5.1 and 5.2 is that the heat recovery fractions are generally higher for the warm, cold model by about 10 to 30 percentage points for combined heat capacitances of over 2000 Btu/F. Using the same example introduced earlier, for a ratio of $r = 0.5$, and total heat capacity of 2000 Btu/F the heat recovery fraction is about 86% (a 22 percentage point increase over the conventional load profile). The difference of the heat recovery fraction between the two load models is not as great when the combined heat capacitance is low (below 1000 Btu/F). Also note that the highest recovery fraction is obtained with $r = 1.0$ for the advanced load characteristic whereas the best

Figure 5.2 Infinite NTU model warm-cold driving series.

ratio for the conventional load characteristic is about $r = 0.8$. This is because there is less supply than waste water flow in the conventional system whereas the flows are equal in the advanced system of the type simulated here.

The results of the simulation runs can be presented in a different form shown in Figure 5.3. In Figure 5.3 the ratio C_s/C_w is plotted on the y-axis and $C_s + C_w$ plotted on the x-axis. Thus, for a given heat recovery fraction the best values of C_s and C_w can be chosen. Figure 5.3 illustrates that much larger heat storage, $C_s + C_w$, is required if the ratio of C_s/C_w is small (under 0.25).

Up to now we have been discussing an infinite NTU exchanger, an exchanger with idealized heat transfer. The following discussion concerns finite NTU exchangers.

A finite NTU heat exchanger has "real" parameters (e.g. heat exchanger element wall thickness and area) that characterize a particular exchanger. Various materials are used for heat exchangers. Metals usually have better overall heat transfer properties than non-metals. This is perhaps true for forced flow type heat exchangers. However, the design for the waste water heat recovery appliance considered in this report relies on large cross sectional areas and low fluid velocity (correspondingly low Reynold's number). Therefore, free convection predominates, not forced convection. The primary resistance to heat transfer in free convection heat transfer is the boundary layer on both sides of the heat exchanger element wall. Therefore,

Figure 5.3 Infinite NTU model warm-cold driving series.

the thermal resistance of the wall material is less important. The curves in Figure 5.4 illustrate the comparison of selected materials for use as a heat exchanger element.

The curves in Figure 5.4 were plotted from the data in Tables B.6, B.7, B.8 in Appendix B. The comparison is shown for copper, stainless steel, and polyvinylchloride plastic. The parameters of the exchanger chosen to be graphed are: $C_s + C_w = 800$ Btu/F, $C_s/C_w = 0.25$, seven node model for storage, 5 washer loads an hour (AVGTBL = 12), "advanced" warm, cold load profile. The heat recovery fraction is plotted in Figure 5.4 as a function of the surface area (sq. ft.) of the exchanger. As illustrated in Figure 5.4 the heat recovery fractions are nearly the same for stainless steel and copper no matter what the surface area. As can be expected, the heat recovery fraction is greater with increasing area. The curve for polyvinylchloride is nearly as good as the curves for stainless steel and copper, about 5 to 8 percentage points less heat recovery over the given range of areas (50-400 sq. ft.). This is significant. By using a less expensive and more corrosion resistant material, PVC, the heat recovery fraction only drops 5 to 8 percentage points. In examining Figure 5.4 it is apparent that a reduction in the heat recovery fraction by using PVC plastic can be compensated for by using more surface area for the heat exchanger element. For example, a copper exchanger element surface

Figure 5.4 Comparison of copper, stainless steel and PVC for heat exchanger element material.

area of 100 square feet is expected to recover about 34% of the heat. The same performance can be matched by using a PVC plastic exchanger element surface area of 180 square feet. Plastic is therefore a serious contender for heat transfer element surfaces when the dominant resistance to heat transfer between two fluids is the boundary layer in the fluids.

The performance of the waste water heat recovery appliance is affected by the amount of use the washing machines get over a given time period. We would expect that with more machines in use more hot water would be demanded, imposing a greater load on the heat exchanger. A comparison between two different use rates is plotted in Figure 5.5. The curves in Figure 5.5 were plotted from the data in Tables B.6 and B.9 in Appendix B with the following parameters: polyvinylchloride heat exchanger element, $C_s/C_w = 0.25$, $C_s + C_w = 800$ Btu/F, seven node storage model. Two curves are shown in Figure 5.5. The one labeled AVGTBL = 6 means an average time between loads of 6 minutes or 10 washing machine uses per hour. The other curve, AVGTBL = 12 corresponds to 5 washing machine uses per hour. As shown in Figure 5.5 there is little difference between the two curves. The AVGTBL = 6 curve is about 3 or 4 percentage points higher than the AVGTBL = 12 curve. Therefore, the number of machines in use, in this case 10 per hour compared to 5, has little effect on the heat recovery fraction.

Figure 5.5 Effect of average time between loads on fraction.

RULES OF THUMB

From the computer simulation results we can draw some rules for sizing the waste water heat recovery appliance. These rules are based on target heat recovery fractions of .4 for conventional laundromats and .6 for advanced laundromats. The design parameters of interest are 1) total hold-up volume, 2) ratio of supply to waste side hold-up volume, and 3) surface area. The sizing rules are listed below:

The heat exchanger element (supply side vessel) should be suspended about four inches above the bottom of the outer tank (waste side vessel) to allow cold drain water to pass through the tank without contacting the heat exchanger element or disturbing stratification. This portion of the heat exchanger hold-up volume is not active and therefore is not considered part of the hold-up volume referred to in the following statements.

Total hold-up volume should be at least 6 times the hold-up volume of a single washing machine or the hold-up volume necessary to achieve an average residence time of 20 minutes for both supply and waste streams, whichever is greater.

The ratio of supply to waste water hold-up volume should be between .6 and 1.0. A ratio of about .8 yields the best performance in the typical conventional laundromat. A ratio of 1.0 maximizes performance in the advanced laundromat.

Heat exchanger surface area, in square feet, should be about twice the average hot water flow rate, in gallons per hour, or about three fourths the waste water flow rate in gallons per hour, whichever is greater. If a metal heat exchange element (supply side vessel) is substituted for the plastic element, its area can be reduced by 40%.

DESIGN AND FABRICATION

The use of plastic for heat transfer devices is new. Discussed in this section are some conclusions, recommendations, and ideas that are drawn from the design and fabrication of the waste water heat recovery appliance. Of primary importance is the end use of the appliance. As indicated in the survey of laundromats, most laundromats are compact in floor space and utilize space as efficiently as possible. Thus, the tank needs to fit into an already tight floor plan in most instances. In addition the fact that the majority of laundromats operate in rented space means it may be difficult for owners to make modifications.

Using the fabrication method discussed in Chapter 3 allows for construction of a tank to fit the particular application. If the appliance were to be manufactured in large quantities this could be accomplished by supplying a variety of sizes. For custom quantities, the tank could be fabricated on site. The on site fabrication would also be advantageous if the door to the utility room is smaller than the proposed tank. In new installations the tank can be designed in.

The designer for a new laundromat should consider the installation of a waste water heat recovery appliance.

Some recommendations are:

1. Locate the waste water drain for the washing machines in proximity to the cold water supply and boiler. Thus, if a heat recovery appliance is to be considered at a later date, the retrofitting costs will be minimal.
2. Plan three pipes to each washer for water supply, a hot, warm, and cold. This will allow connection of three-inlet type machines when they become available and allow easy and flexible partitioning of machines into hot-warm machines and warm-cold machines without physically moving the machines.
3. Locate the waste outlet from the washing machines high enough to allow gravity flow of waste water into a waste water heat recovery appliance.

Considering the design and fabrication of the waste water heat recovery appliance itself, the following recommendations are offered.

1. Consider a tank size that will fit the application. Tall tanks, e.g. six ft. in height, can be considered if the washing machine pumps can pump the waste water that high and the tank can be serviced (the heat exchanger element removed).
2. The fit of the exchanger elements into the tank needs to be considered when designing the tank size

to provide for the best hold-up ratio and surface area for the heat exchanger element.

3. All components, tank and heat exchanger element, should be hydrostatically pressure tested before use in a laundromat. Many laundromats are unattended and an unexpected leak would be awkward.
4. In construction of the polypropylene tank, forms could be used to decrease construction time. If custom forms were used the steel frame would not have to be self supporting. Continuously threaded rods which acted as spacers for the angle iron could be replaced with standard nuts, bolts, and lockwashers.
5. The heat exchanger elements could be designed in modules with header connections. This would allow standardization of the heat exchanger elements lowering large quantity production costs. The number of modules could then be varied for different tank sizes.

APPLICATION AND ECONOMICS

An example will be given here to illustrate the application of the waste water heat recovery appliance in a laundromat. Consider a laundromat that uses 1000 gallons of 130° F of hot water over a 16 hour period and pays \$5/MBtu for fuel. The estimated yearly energy cost is $(1000 \text{ gal/day})(130 \text{ F} - 50 \text{ F})(8.34 \text{ lbm/gal})(365 \text{ days/yr}) (1.3 \text{ furnace}$

efficiency)($\$5/\text{MBtu}$) = $\$1600/\text{year}$. A small exchanger designed with characteristics similar to those described in Chapters 3 and 4 might have a tank hold-up volume of 90 gallons ($C_w = 750 \text{ Btu/F}$) and an exchanger element size of 156 square feet area and 30 gallons hold-up volume ($C_s = 250 \text{ Btu/F}$). Therefore, $C_s + C_w = 1000 \text{ Btu/F}$ and $C_s/C_w = 0.35$. From Table B.11 a heat recovery fraction is estimated to be 0.36 for an exchanger with these parameters. An exchanger of this size could be fabricated and sold to the end user for about $\$2700$. Adding $\$100$ for shipping and $\$400$ for installation the cost to the laundromat owner would be $\$3300$. The savings from using the appliance is expected to be $(0.36)(1600) = \$575$ per year. Thus, excluding tax credits the exchanger would have a simple payback of $3300/575 = 5.7$ years. This is shorter than the payback for most solar hot water heating systems. The discounted payback can also be found by using the nomograph in Figure 5.6. Assuming a 12% discount rate, 25% tax bracket, 18% estimated year fuel price increase, and a 20 year life cycle for the appliance, the discounted payback is about 4.6 years. Tax incentives make the investment look even more attractive. If we assume a combined (state and federal) investment tax credit for energy property of 45% then the investment cost can be recalculated to $(3300 - (3300)(0.45)) = 1815$. The simple payback can also be recalculated $1815/575 = 3.2$ years. Thus, tax incentives and use of life cycle costing principles are important in any economic analysis.

Figure 5.6 Life cycle costing nomograph showing discounted payback period.

The outcome is also sensitive to the price of fuel which was assumed here to be \$5/MBtu but is rapidly rising in price and is higher in different geographic areas.

RECOMMENDATIONS FOR ENERGY EFFICIENCY IN LAUNDROMATS

A number of recommendations can be made regarding the interaction of the washing machines, boiler, waste heat recovery appliance, and other heat sources. These recommendations are based on the undesirability of mixing hot and cold to obtain warm water. Such mixing reduces the potential for waste heat recovery and also reduces the efficiency of such alternative heat sources as solar water heaters and heat pumps.

One possible way to reduce mixing is to use washing machines outfitted with hot, warm, and cold water inlet connections and control valves in place of the usual hot and cold inlets. Figure 5.7 shows this arrangement. Tempering valves mix a small amount of either hot or cold water with the preheated water leaving the waste heat exchanger to obtain warm water at the required set temperature. (The tempering valve that admits hot water should be set about 5° F lower than the tempering valve that admits cold water to prevent the tempering valves from working against each other.)

Since washing machines are not currently manufactured with a third warm water inlet an alternative scheme to reduce mixing, shown in Figure 5.8, has been

Figure 5.8 A practical implementation of the "advanced" laundromat.

devised. Here the machines are segregated into one group of machines offering hot or warm wash and warm rinse cycles and a second group offering warm or cold wash and cold rinse cycles. This arrangement not only reduces mixing but also provides the laundromat owner a convenient mechanism for charging hot water users more, thus encouraging users to conserve hot water. Given that hot water heating constitutes about 12% of total (both owning and operating) laundromat expenses, that washing machines produce about 50% of gross laundromat income, and that water heating requirements of the high temperature machines would be about three times the requirements of the low temperature machines, a 15 to 20 percent higher price for the high temperature machines is justified. Conventional machines can be easily modified, either in the field or in manufacture, by simple design modifications, to draw water from only the hot or cold inlet, never both simultaneously. The high temperature machine will draw warm water from the cold inlet whenever a warm or cold fill is called for; the hot inlet will admit hot water only when a hot fill is called for, as usual. The low temperature machine will draw warm water from the hot water inlet whenever a warm or hot fill is called for; the cold inlet will admit cold water only when a cold fill is called for, as usual.

The idea of having a water heating plant that supplies hot, warm, and cold water in the most thermodynamically efficient manner and of having washing

machines that never mix hot and cold water is the key to achieving maximum heat recovery potential. These practices will also increase the efficiency of solar and heat pump sources if they are properly integrated with the water heating plant.

A laundromat water heating plant that employs both waste heat recovery and solar water heating is shown in Figure 5.9. Most of the time one would expect the solar heating stage to raise the supply water to a temperature above the warm set point. Under these conditions supply water is taken from the solar and heat recovery stage outlets and mixed in the correct proportions by the second tempering valve to provide water at the warm set point temperature. However, sometimes the heat recovery stage will, by itself, raise the supply water temperature sufficiently so that cold supply water has to be mixed in by the third tempering valve. On the other hand, supply water may sometimes leave the solar stage at a temperature less than the warm set point, in which case hot water from the boiler will be mixed in by the first tempering valve. By always selecting the two supply sources (one hotter and one colder) that are closest in temperature to the warm set point the system minimizes unnecessary thermodynamic degradation and extracts the maximum possible efficiency from the solar and heat recovery stages.

One small modification could be made to slightly reduce the cost and complexity of the system and conserve

Figure 5.9 Solar heated laundromat with waste heat recovery.

auxiliary energy at the same time. The first tempering valve could be removed so that the warm supply temperature would fall below the warm set point at times when the solar and heat recovery stages fail to supply enough heat. Users would thus be discouraged from doing their laundry on cloudy days when the supply of "free" energy is low.

The solar collector and storage should be sized for a load equal to $(1-f)$, where f is the heat recovery fraction) of the total load. Also, the AUX tank needs to be only about one fourth the size of the tank required in a conventional system with the same total load because most of the water is drawn off the warm tap.

A laundromat water heating plant that employs both waste heat recovery and heat pump back-up is shown in Figure 5.10. Separate warm and hot storage tanks are used because the heat pump operates more efficiently (higher Coefficient Of Performance, or COP) when it is heating water to the warm set point than when heating water to the hot set point. The amount of water heated to the hot set point in a system like this is generally less than one fourth the amount of water heated to the hot set point in a conventional system. The AUX2 storage tank can therefore be one fourth the size, and the AUX1 tank about one half the size, of the single storage tank required in a conventional system.

The heat pump should be sized for a load equal to $(1-f)$ of the total load. A special advantage of combining the heat pump with the waste water heat exchanger is that

Figure 5.10 Heat pump heated laundromat with waste water heat recovery.

space can be provided in the bottom of the waste water heat exchanger tank for the heat pump evaporator. Higher evaporation temperatures, thus higher COP's, can generally be achieved with the water source heat pump than with a conventional air source heat pump. Another advantage is that this system approaches a thermally closed system--that is, the waste water leaves at about the same temperature as the supply water enters--which contributes very little thermal pollution to the external environment.

A modification that could be made to the heat pump system is the addition of solar collectors and substitution of a solar storage tank for AUX1. The tempering valves shown in Figure 5.9 should also be installed.

The above recommendations for laundromat system modifications are for a "typical" laundromat. Naturally, the above recommendations need to be engineered to fit a particular situation following careful analysis of the heating system in the laundromat. There are many ways to reduce energy consumption in laundromats. We have shown that the waste water heat recovery appliance is one solution to reducing the quantity of energy needed to heat hot water for washing machines. There are many other approaches. The future of the laundromat industry is in transition with many store owners renting, some being managed by investment groups, and some laundromat owners switching to all cold water washes. The need for the waste water heat recovery appliance has to be determined for each situation taking

into consideration the price of energy, government incentives, availability of capital, interest rates, laundromat heating plant layout, etc. When properly applied the waste water heat recovery appliance will "recycle" an increasingly valuable commodity - energy.

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APPENDIX A
COMPUTER PROGRAMS

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SIMULATION OF WASHING MACHINE CYCLES AND LAUNDROMAT USE

Program SEQUE generates a sequence of hot water demands and waste water discharge volumes and temperatures (the "fill/drain series") for a specified period of operation of a laundromat with specified washing machine and use parameters. The parameters that describe the washing machine cycles and the parameters that describe deterministic and stochastic components of laundromat use were enumerated in Chapter 2. The fill/drain series is the used to drive the heat exchanger model, program HXSIM.

The simulation algorithm is based on discrete time intervals (we have used half minute intervals). Events that occur within a given time interval (e.g. a load start, or the beginning or end of a phase of a particular washing machine cycle) are therefore modeled as occurring simultaneously. At each load start the phases of the particular cycle are loaded into a circular event queue. Events are sequentially unloaded between load starts and as each event is unloaded its effect on the heat recovery system is computed and the appropriate information (time, flow volume, temperature) is appended to the fill/drain series.

The essential subroutines in program SEQUE are INIT, NEWDAY, MTONXT, NXTPHS, INQUE, FINQUE, FINDQ, and DEQUE.

INIT reads the laundromat parameters, checks program restrictions on these parameters, echoes them to TAPE 4 (as the first record of the output file containing the fill/

drain series) and puts them in a form that is efficient for use by the simulation algorithm. INIT also clears and initializes certain simulation variables.

NEWDAY initializes certain simulation variables at the beginning of each day.

NXTPHS performs several tasks, some of which are accomplished by calling subroutines FINDQ, DEQUE, MTONXT, and INQUE. FINDQ removes defunct events from the circular event and "cycle active" queues and sets a flag if a user has been waiting for the type of machine that just finished. DEQUE then unloads the current event from the circular event queue. NXTPHS then determines the time to the next impending event in the event queue.

INQUE is called by NXTPHS whenever a new load start occurs before the next impending event in the event queue. INQUE merges the events of the new load cycle into the circular event queue by calls to LBUSY, INQUE1, INQUE2 and FINQUE. If no machines of the desired type are available, but not all machines in the laundromat are busy, LBUSY is called to determine if a machine of the desired type is available. If not, NXTLD and LBUSY are called repeatedly until a type of machine is selected that is available. INQUE1 AND INQUE2 are then called repeatedly to merge the fill and drain parameters for each phase of the machine cycle into the circular event queue. FINQUE is then called to insert the type and termination time of the wash cycle into the circular cycle-active queue.

MTONXT is also called by NXTPHS whenever a new load start occurs before the next impending event. After INQUE processes the new load start, MTONXT generates the time and cycle type of the next load start. Cycle type is determined by a call to function NXTLD. NXTPHS loops back to call INQUE and MTONXT if (and so long as) the next load start is immediate (recall that load starts occurring in the same discrete time interval are considered simultaneous). The tasks executed by NXTPHS are complete when the first non-simultaneous load start is generated.

Upon return from NXTPHS, the main program writes the fill/drain information to TAPE4 and loops back to call NXTPHS unless it is closing time at the laundromat. At closing time the main program loops all the way back to call NEWDAY again until the desired number of days has been simulated at which time the program stops.

SIMULATION OF THE WASTE-WATER HEAT RECOVERY APPLIANCE

Program HXSIM simulates the performance of a specified heat exchanger over a specified time period using data from one of the fill/drain series, as the forcing function. Because the laundromat simulation (SEQUE) and heat exchanger simulation (HXSIM) are separate programs, many different heat exchanger designs may be simulated using the same forcing function while the calculations that generate the forcing function are performed only once.

The forcing function is just a series of fill and drain events. Each event is characterized by its time of

occurrence and the thermal capacitance(s) of the volume(s) of water passed through the heat exchanger during the event. A temperature value is specified whenever the event includes draining activities. For convenience, the time until the next event is also specified. HXSIM simulates drain and fill actions as if they were instantaneous. During the intervals between fill/drain events, HXSIM simulates "relaxation"--i.e. the tendency for the supply and waste water temperature profiles to decay exponentially towards an average temperature profile. Thus HXSIM basically comprises a loop in which fill/drain events and relaxation periods are repeatedly and sequentially simulated.

The essential subroutines in program HXSIM are INIT, DAYSIM, NEWDAY, FILL, DRAIN, and RELAX. INIT rewinds and reads the record of laundromat parameters (the first record) from the file containing the fill/drain series (now TAPE7). The laundromat parameters are echoed to the output file (TAPE3) to completely identify the forcing function used in the simulation (the laundromat parameters are not used for any other purpose in HXSIM). The first set of heat exchanger parameters is then read from TAPE7 and the heat exchanger state (node temperatures) is initialized. On subsequent calls to INIT the echoing of laundromat parameters to TAPE3 is suppressed. TAPE7 is never rewound and execution is terminated when an end-of-file is encountered on TAPE7.

DAYSIM is called once for each simulated day. DAYSIM immediately calls NEWDAY to clear the daily performance accumulators and initialize the time clock and the average supply and waste water temperatures. DAYSIM then enters a loop which 1) reads the next fill/drain event, 2) calls DRAIN or FILL or both (order is immaterial) as required, 3) calls RELAX. The loop is repeated until the end of the fill/drain series is detected (DTIME=0). A daily performance summary is then printed and the program returns from DAYSIM to the main program.

FILL, DRAIN and RELAX simulate the fill and drain operations as described in Chapter 2. Note, however, that both the supply and waste-water side nodes are numbered from top to bottom in the program. Also note that FILL and DRAIN calculate new node temperatures in one pass regardless of whether or not the fill or drain volume exceeds the volume per node. RELAX calls RFREE2 (which in turn calls DELRHO) to calculate some free convection heat transfer parameters for subsequent use by RFREE. RELAX then calls RFREE twice in order to use different film temperatures and channel widths in the calculation of supply and waste water side heat transfer coefficients. The overall node conductance and the progress toward equilibrium (complete relaxation) are then computed. Finally, the new node temperatures, average node temperatures and amount of heat transferred are computed. The "heat transferred" accumulator is updated and control returns from RELAX to DAYSIM.


```

C ADD FINQUE,FINQO,LBUSY, 4-5-82
C REPAIR NXTPHS,INQUE(ADD INQUE1),INQUE2,MTONXT, 4-22-82
C REMOVE $=SIGNS, ADD SUBR SEPERATORS (1), CLEAN UP HERE, THERE, 5-1-82
PROGRAM SEQUE(INPUT=1Q1B,OUTPUT=1Q1B,TAPE1=1Q1B,TAPE2=1Q1B,
+TAPE3=1Q1B,TAPE8=1Q1B,TAPE9=1Q1B,TAPE5=INPUT,TAPE6=OUTPUT
+ ,TAPE4=1Q1B)
COMMON/TIME/ MTIME,NLTIME,IDQWK
COMMON/RUNPAR/ NDAYS
COMMON/SYSPAR1/TMAIN,TSET,TAMB,MOPEN,MCLOSE,MONITE
COMMON/SYSPAR2/LDTYP, TMUNIT, JCYCLE(2,20), CYCLE(20), ICYCLE(3,5,20)
COMMON/EVENTQ/ MAXQ, IC, NQEV, Q(3,60)
COMMON/ENDCYC/ NTMACH, NTBUSY, MBUSY(6), LDTYPE(30), MENDTM(30)
DIMENSION SUM(10,3)
DATA ZERC/C./
CALL INIT
DO 20 IO=1,NDAYS
CALL NEWDAY
DO 5 I=1,10
DO 5 J=1,3
5 SUM(I,J)=0.
DTIME=(MTIME-MOPEN)*TMUNIT
C WRITE(4) MOPEN,DTIME,ZERO,ZERO,ZERO
C WRITE(4,9C) MOPEN,DTIME,ZERO,ZERO,ZERO
CALL STATS(SUM,DTIME,ZERO,ZERO,ZERO)
10 MSTEP=MTIME
C CALL NXTPHS(DTIME,FTHMAS,DTHMAS,TWW)
C WRITE(4) MSTEP,DTIME,FTHMAS,DTHMAS,TWW
C WRITE(4,9C) MSTEP,DTIME,FTHMAS,DTHMAS,TWW
CALL STATS(SUM,DTIME,FTHMAS,DTHMAS,TWW)
90 FORMAT(1X,I5,F20.15,3F10.5)

C IF(NQEV.NE.C) GOTO 1C
C IF(MTIME.LT.MCLOSE) GOTO 1C
C WRITE(4) ZERC,ZERO,ZERO,ZERO,ZERO
C WRITE(4,9C) ZERC,ZERO,ZERO,ZERO,ZERO
CALL STATS(SUM,DTIME,1.,1.,1.)
CALL WSTATS(SUM)
C WRITE(4) MOPEN,ZERO,ZERO,ZERO,ZERO
C WRITE(4,9C) MOPEN,ZERC,ZERO,ZERO,ZERO
C DESIRABLE TO CALL FINDQ TC CLEAR /ENDCYC/, PRINT FINAL STATE ON TAPE9 HERE?
2C CONTINUE
STOP
END

C = = = = =
SUBROUTINE STATS(SUM,DT,F,D,TW)
DIMENSION SUM(10,3)
C WRITE(3,9C) DT,F,D,TW
90 FORMAT(" STATS..DT,F,D,TW=",4F8.3)
CALL STATS2(SUM(1,1),DT,F)
CALL STATS2(SUM(1,2),DT,D)
CALL STATS2(SUM(1,3),DT,D*TW)
C WRITE(3,91) (SUM(1,J),J=1,3)
91 FORMAT(" TIME ACCUMULATORS=",3F8.3)
RETURN
END

C = = = = =
SUBROUTINE STATS2(S,DT,THM)
DIMENSION S(10)
IF(THM.EQ.C.) GOTO 1CC
C WRITE(3,9C) THM,(S(I),I=1,10)
90 FORMAT
+ (" X,S(1-10)=",F7.0,F8.5,F8.0,F5.0,F7.3,F7.4,F10.0,F8.0,3E9.3)
S(3)=S(3)+1
S(4)=S(4)+S(1)
S(5)=S(5)+S(1)*S(1)
T=S(2)/S(1)
IF(S(1).EQ.0..AND.S(2).EQ.0.) T=0.
S(6)=S(6)+T
S(7)=S(7)+S(2)
S(8)=S(8)+T*T
S(9)=S(9)+S(2)*S(2)
S(10)=S(10)+T*S(2)
S(1)=DT

```

```

S(2)=THM
RETURN
S(1)=S(1)+DT
RETURN
END

```

```

SUBROUTINE WSTATS(S)
DIMENSION S(10,3)
CALL WSTAT2(S(1,1),"SUPPLY WATER SIDE OF HEAT EXCHANGER",4)
CALL WSTAT2(S(1,2),"WASTE WATER SIDE OF HEAT EXCHANGER",4)
CALL WSTAT2(S(1,3),"WASTE WATER ENTHALPY",2)
RETURN
END

```

```

SUBROUTINE WSTAT2(S,MSG,LEN)
DIMENSION S(10),MSG(8)
STDEV(SS,AV)=SQRT(AMAX1((SS-N*AV*AV)/(N-1),0.))
WRITE(3,*)"S(1-10)=", (S(I),I=1,10)
N=S(3)
AVDT=S(4)/N
SDDT=STDEV(S(5),AVDT)
AVFLOW=S(6)/N
AVSLUG=S(7)/N
SDFLOW=STDEV(S(8),AVFLOW)
SDSLUG=STDEV(S(9),AVSLUG)
SDMIXD=STDEV(S(10),AVFLOW*AVDT)
WRITE(3,90) (MSG(I),I=1,LEN)
WRITE(3,91)
WRITE(3,92) N,AVDT,SDDT,AVFLOW,AVSLUG,SDFLOW,SDSLUG,SDMIXD
90 FORMAT(" STATISTICS FOR ",8A10)
91 FORMAT(" N = AVDT = SDDT = AVFLOW = AVSLUG = SDFLOW = SDSLUG =",
+ " SDMIXD =")
92 FORMAT(1X,I4,2F8.4,F9.0,4F9.2)
RETURN
END

```

```

SUBROUTINE DBUGQ(IU)
COMMON/ENDCYC/ NTMACH,NTBUSY,MBUSY(6),LDTYPE(30),MENDTM(30)
COMMON/MACHIN/ NMCHTP,NMACH(6),ILDTP(6),MCHTYP(20)
COMMON/TIME/ MTIME,NLTIME,IDQWK
COMMON/SYSPAR2/LDTYPS,TMUNIT,JCYCLE(2,20),CYCLE(20),ICYCLE(3,5,20)
DIMENSION NL(30),IPRNT(36)

```

THIS IS NOT AN ELEGANT WAY TO DO IT, BUT HERE GOES...

```

DO 10 I=1,LDTYPS
10 NL(I)=0
DO 20 I=1,NTBUSY
20 NL(LDTYPE(I))=NL(LDTYPE(I))+1

```

```

L=LDTYPE COUNTER
M=MCHTYPE COUNTER
IP=PRINT FIELD COUNTER (=L+M)

```

```

L=0
M=1
IP=1
IPRNT(IP)=NMACH(M)

```

```

30 L=L+1
IP=IP+1
IPRNT(IP)=NL(L)
IF(L.NE.ILDTP(M)) GOTO 40
M=M+1
IP=IP+1
IPRNT(IP)=NMACH(M)
40 IF(L.LT.LDTYPS) GOTO 30

```

```

IP=IP-1
WRITE(IU,99)MTIME,(IPRNT(I),I=1,IP)
WRITE(6,99)MTIME,(IPRNT(I),I=1,IP)
99 FORMAT(1X,I4,42I3)
RETURN
END

```

```

C SUBROUTINE NEWDAY
COMMON/MEASR/ QAT,QST,QWT,QXST,QXWT,QATT,QSTT,QWTT,QXSTT,QXWTT
COMMON/EVENTQ/ MAXQ,IC,NQEV,Q(3,60)
C COMMON/TNODES/TS(21),TW(21)
COMMON/SYSPAR1/TMAIN,TSET,TAMB,MOPEN,MCLOSE,MONITE
COMMON/SYSPAR2/LDTYPS,TMUNIT,JCYCLE(2,20),CYCLE(20),ICYCLE(3,5,20)
C COMMON/XPART1/NODES,CSN,CWN,FS,FW,RWALL,ANODE,XL,CHNLS,CHNLW
COMMON/TIME/ MTIME,NLTIME,IDOWK
COMMON/NLPAR/ NLTY
COMMON/ENDCYC/ NTMACH,NTBUSY,MBUSY(6),LDTYPE(30),MENDTM(30)
COMMON/MACHIN/ NMCHTP,NMACH(6),ILDTP(6),MCHTYP(20)
COMMON/SUBCLK/ MMTIME
DATA MAXQ/60/

C ZERO ACCUMULATORS FOR AUXILIARY (QAT) AND RECOVERED (QRT) HEAT
QAT=0.
QST=0.
QWT=0.
QXST=0.
QXWT=0.

C S E T I N I T I A L S U P P L Y & W A S T E W A T E R N O D E T E M P E R A T U R E S
DO 10 I=1,NODES
TS(I)=TAMB
10 TW(I)=TAMB
TSAVG=TWAVG=TAMB

C Z E R O T H E C I R C U L A R E V E N T Q U E U E
DO 110 I=1,MAXQ
Q(1,I)=0.
Q(2,I)=0.
110 Q(3,I)=0.
IQ=1
NQEV=0

C Z E R O T H E "CYCLES ACTIVE" S T A C K (/ENDCYC/ OF FINQUE)
NTBUSY=0
DO 210 I=1,LDTYPS
LDTYPE(I)=0
210 MENDTM(I)=0
DO 220 I=1,NMCHTP
220 MBUSY(I)=0

C F E T C H O R G E N E R A T E T I M E & T Y P E O F F I R S T L O A D
IDOWK=MOD(IDOWK,7)+1
MTIME=MOPEN
MMTIME=0

C MTONXT ALSO GENERATES NLTY (SEE COMMON BLOCK /NLPAR/)
MTIME=MTONXT(MTIME,IDOWK)/2+MTIME

C E N T E R P H A S E S O F F I R S T L O A D C Y C L E I N T O T H E Q U E U E
CALL INQUE(NLTY)
NLTIME=MTONXT(MTIME,IDOWK)+MTIME
WRITE(6,*)"NEWDAY..MT,NLT=",MTIME,NLTIME
RETURN
END

C FUNCTION MTONXT(MTIME,IDOWK)
C FCN SUBR TO GENERATE TIME TO NEXT LOAD START USING *AVG TIME BETWEEN
C LOADS* (TBL), A FUNCTION OF TIME-OF-DAY & DAY-OF-WEEK.
COMMON/SYSPAR3/AVGTBL,TYPREL(20)
COMMON/SYSPAR1/TMAIN,TSET,TAMB,MOPEN,MCLOSE,MONITE
COMMON/SYSPAR2/LDTYPS,TMUNIT,JCYCLE(2,20),CYCLE(20),ICYCLE(3,5,20)
COMMON/NLPAR/ NLTY
COMMON/ENDCYC/ NTMACH,NTBUSY,MBUSY(6),LDTYPE(30),MENDTM(30)
C /SUBCLK/ ALSO DECLARED IN NEWDAY TO INITIALIZE MMTIME=0
COMMON/SUBCLK/ MMTIME

C PROBABILITY OF LOAD START DURING NEXT MICROTIME INTERVAL, MMTIME:
P=1./((TBL(MTIME*TMUNIT,IDOWK)*NTMACH)
MTONXT=0

```

```

C INCREMENT MICROTIME, MMTIME, UNTIL RANF.LE.P
  IF(MMTIME.LT.NTMACH) GOTO 20
  MMTIME=0
  MTONXT=MTONXT+1
  20 MMTIME=MMTIME+1
  IF(RANF(CUM).GT.P) GOTO 10
  -----
C WHAT ABOUT WHEN MTONXT+MTIME>MCLOSE?
  -----
C NOW THAT WE HAVE TIME TIL NEXT LOAD START, GENERATE CYCLE TYPE OF NXTLD
  NLTIME=NXTLD(DUM)
  RETURN
  END
  -----
C FUNCTION TBL(TOD, IDOWK)
C RETURNS CURRENT AVG. TIME BETWEEN LOADS, A FCN OF TIME-OF-DAY AND
C TIME-OF-WEEK. FOR NOW, JUST LET TBL BE FIXED: TBL=AVGTBL.
  COMMON/SYSPAR3/AVGTBL, TYPRBL(20)
  COMMON/TBLFCN/ DTBLF(25,2), WTBLF(7)
  F(IHOUR)=PLASTH*DTBLF(IHOUR,K) + PNEXTH*DTBLF(IHOUR+1,K)
C COMPUTE INTERPOLATION WEIGHTINGS (P) AND TABLE POINTER (IHOUR)
  IHOUR=AMOD(TOD,24.)+1
  AMOD CAN BE ELIMINATED WHEN SUBR NXPBS IS CLEANED UP. OFFENDER IS LINE
  81 OF NXPBS: IF(NLTIME.GE.MCLCSE) NLTIME=MAX3(NLTIME+MAXQ,MONITE+MCPEN).
  IHOUR=MIN(IHOUR,24)
  DO THE ABOVE IN CASE TOD=24. IS PASSED INSTEAD OF TOD=0. AT MIDNIGHT
  PNEXTH=TOD-IHOUR
  IHOUR=IHOUR+1
  PLASTH=IHOUR-TOD
C COMPUTE K (K=1 FOR WEEKDAY, K=2 FOR WEEKEND)
  K=IDOWK/6 + 1
  FF=AMAX1( F(IHOUR)*WTBLF(IDOWK), .1E-20)
  WRITE(6,*)"TBL..T,IH,ID,K,FF=",TOD,IHOUR,IDOWK,K,FF,PLASTH,PNEXTH
  TBL=AVGTBL/FF
  TBL=AVGTBL
  RETURN
  END
  -----
C FUNCTION NXTLD(DUM)
C FCN SUBR TO GENERATE *CYCLE TYPE* FOR NEXT LOAD USING VECTOR OF
C PROBABILITIES OF ALL POSSIBLE CYCLES.
  COMMON/TIME/ MTIME,NLTIME, IDOWK
  COMMON/SYSPAR3/AVGTBL, TYPRBL(20)
  COMMON/SYSPAR2/LDTYPES, TMUNIT, JCYCLE(2,20), CYCLE(20), ICYCLE(3,5,20)
  FOR I=1 TO NLDTYPES (ALL DIFFERENT TYPES OF WASH CYCLE)
    JCYCLE(1,I)=NMBR OF PHASES IN THE ITH TYPE OF CYCLE
    JCYCLE(2,I)=MACHINE TYPE ON WHICH ITH TYPE OF CYCLE IS IMPLEMENTED
    CYCLE(I)=THERMAL MASS OF FILL WATER
    ( MIGHT TRY CYCLE(1,I)=MIN, CYCLE(2,I)=MAX FILLMASS..LSE POIST )
    FOR J=1 TO JCYCLE(1,I) (ALL PHASES OF ITH TYPE OF CYCLE)
      ICYCLE(1,J,I)=TIME FROM FILL TO DRAIN IN TMUNITS
      ICYCLE(2,J,I)=TIME TO NEXT FILL (.EQ.0 IF LAST PHASE)
      ICYCLE(3,J,I)=FILL TMPTUR...1=COLD 2=WARM 3=HCT
  NOTE...CAN PASS ICYCLE(1,I),CYCLE(1,I) TO, E.G., SL9R INQLE
    WITH F.P.S DIMENSIONED ICYCLE(6),CYCLE(6)
  TYPRBL IS VECTOR OF BREAK POINTS ON CUMMULATIVE PROBABILITY CURVE OF
  ALL POSSIBLE LOAD TYPES
  (CURVE IS DIVIDED INTO LDTYPES INTERVALS TO REPRESENT ALL LOAD TYPES)
  CUMM=RANF(DUM)
  DO 10 I=1,LDTYPES
  10 IF(TYPRBL(I).GE.CUMM) GOTO 20
  I=LDTYPES
  20 NXTLD=I
  WRITE(6,*)"NXTLD...NL,MTM,NTM=",NXTLD,MTIME,NLTIME
  RETURN
  END

```

```

C - - - - -
C SUBROUTINE NXTPHS(OTIME,FTHMAS,OTHMAS,TW)
C SELECTIVE SLBR THAT CALLS OTHER ROUTINES TO DETERMINE TIME/TYPE OF
C NEXT WASH CYCLE, KEEP TRACK OF WHEN ALL MACHINES OF ANY ONE TYPE ARE
C BUSY, AND, AT START OF EACH NEW LOAD, DECOMPOSE CYCLE INTO ITS
C COMPONENT FILL AND DRAIN OPERATIONS. EACH CALL TO NXTPHS RETURNS
C THE PARAMETERS OF THE CURRENT PHASE I.E. DRAIN AND/OR FILL OPERATION),
C INCLUDING THE RELAXATION TIME (TIME TO NEXT DRAIN AND/OR FILL OPERATION).
C LOGICAL LEVENT
C COMMON/EVENTQ/ MAXQ,IC,NQEV,Q(3,60)
C COMMON/SYSPAR1/TMAIN,TSET,TAMB,MOPEN,MCLOSE,MCNITE
C COMMON/SYSPAR2/LDTYPS,TMUNIT,JCYLE(2,20),CYCLE(20),ICYCLE(3,5,20)
C COMMON/TIME/ MTIME,NLTIME,IDCWK
C COMMON/NLPAR/ NLTP
C COMMON/ENCCYC/ NTMACH,NTBUSY,MBUSY(6),LDTYPE(30),MENDTM(30)
C COMMON/MACHIN/ NMCHTP,NMACH(6),ILDTYPE(6),MCHTYP(20)
C TEST FOR MACHINE(S) FINISHING IN THIS TIME SLOT... SERVICE CUSTOMERS,
C IF ANY, WAITING FOR TYPE(S) OF MACHINE FINISHING.
C CALL FINDQ
C GET AGGREGATE DRAIN/FILL PARAMETERS FOR ALL DRAIN/FILL OPERATIONS
C OCCURRING IN THIS TIME SLOT, MTIME.
C CALL DEQUE(Q(1,IQ),FTHMAS,OTHMAS,TW)
C J=MCHTYP(LDTYPS)
C IF(NQEV.LE.C.OR.NTBUSY.LE.0) WRITE(6,*)"NXP..NQ,NBSY,TM=",NQEV,
C + NTBUSY,MTIME,"LDTP,END",LDTYPE(I),MENDTM(I),I=1,NTBUSY),
C + MBUSY,(MBUSY(I),I=1,J)
C - - - - -
C REMAINDER OF SUBROUTINE DETERMINES TIME OF NEXT FILL/DRAIN BY ANY
C OPERATING MACHINE OR TIME OF NEXT LOAD START, WHICHEVER COMES FIRST.
C IF THE LATTER, FILL/DRAIN PARAMETERS OF THE NEW LOAD ARE PUT IN
C THE QUEUE AND TYPE/TIME OF SUCCEEDING LOAD START ARE GENERATED...
C EXCEPTION OCCURS IF ALL MACHINES OF MCHTYP(NLDTYP) ARE BUSY (SEE
C WHERE INQUE CALLS LBSUY).
C SET "NEXT LOAD POINTER" TO LIMIT SEARCH FOR NEXT EVENT
C NLTIME=MAXC(NLTIME,MTIME+1)
C IQNL=IQ+NLTIME-MTIME
C IF(NTBUSY.GE.NTMACH) IQNL=IQ
C IF(NQEV.GT.C) GOTO 5
C SINCE EVENTQ IS EMPTY PTR VALUE (IQ) IS ARBITRARY, BUT MUST SET IQNL=IQ:
C IQNL=IQ
C IQOLD=IQ-(NLTIME-MTIME)
C GOTO 30
C 5 IF(IQNL.GT.IQ+MAXQ) GOTO 1C
C IF(IQNL.GT. MAXQ) IQNL=IQNL-MAXQ
C IF(IQNL.LE.C) GOTO 930
C IF(IQNL.EQ.IQ) GOTO 900
C MIGHT NEED TO TST FOR EMPTY QUEUE (NQEV=0)
C SEARCH FOR FIRST NON-ZERO ENTRY IN QUEUE
C 10 IQOLD=IQ
C 11 IQ=MOD(IQ,MAXQ)+1
C IF(IQ.EQ.IQNL) GOTO 3C
C QUEUE (CIRCULAR BUFFER) OVERFLOW DIAGNOSTIC
C (SHOULD NEVER HAPPEN THAT ALL MCH BSY BUT QUEUE EMPTY)
C IF(IQ.EQ.IQOLD) GOTO 910
C IF(.NOT.LEVENT(Q(1,IQ))) GOTO 11
C - - - - -
C 30 MDTIME=IQ-IQOLD
C IF(MDTIME.LT.C) MDTIME=MDTIME+MAXQ
C IF(MDTIME.EQ.C) GOTO 940
C OTIME=MDTIME*TMUNIT
C WRITE(3,*)"IQ,IQOLD,TM,OTM,INTP,NLT,Q",IQ,IQOLD,MTIME,MDTIME
C + NLTP,NLTIME,(Q(J,IQ),J=1,3)
C MTIME=MTIME+MDTIME
C IF(IQ.NE.IQNL) RETURN
C - - - - -

```

```

C IF NEXT EVENT IS N E W L O A D S T A R T, MERGE CYCLE INTO EVENTQ
C IF(NLTIME.LT.MCLOSE) CALL INQUE(NLTYP)
C LOOKS LIKE NOLDTM=NLTIME CAN BE MOVED AHEAD OF 40 LOOP...EXECUTE JUST ONCE
  NOLDTM=NLTIME
C MTONXT ALSO GENERATES NLTYP (SEE COMMON BLOCK /NLPAR/)
  NLTIME=MTCNXT(NLTIME, IDOWK)+NLTIME
C DOES NXTLD COME SO SOON THAT IT IS IN CURRENT TIME SLOT?
  IF(NLTIME.GE.MCLOSE) NLTIME=MAXQ(NLTIME+MAXQ, MDNITE+MOPEN)
  IF(NLTIME.GT.NOLDTM) RETURN
  IF(NTBUSY.LT.NTMACH) GOTO 40
C NO MACHINES AVAILABLE...EARLIEST POSSIBLE NLTIME IS NEXT TIME STEP
  NLTIME=NLTIME+1
  RETURN
C NOTE THAT A WAITING LINE IS NOT ESTABLISHED. WE JUST KEEP PUSHING BACK THE
C TIME OF THE NEXT LOAD START. THUS WE DO NOT ACCUMULATE A RESERVE OF CUSTOMERS
C WHO WILL TAKE UP THE SLACK CAPACITY LATER. INSTEAD WE TURN THEM AWAY.
C THIS IS NOT ENTIRELY REALISTIC, BUT THE FACT THAT THE NEXT LOAD STARTS
C IMMEDIATELY WHEN A MACHINE BECOMES AVAILABLE MITIGATES. THEREFORE THE MODEL
C IS MORE REALISTIC THAN A SIMPLE WAITING LINE. IDEALLY, THE RATE OF NEW
C WAITING LINE ENTRIES WOULD BE A FUNCTION OF THE LENGTH OF THE WAITING LINE.
C
C
C
C
C 900 WRITE(6,*)"ERROR..IQNL=I",IQNL,IQ
C     WRITE(6,*)"IQ,MTIME,NLTIME,NQEV=",IQ,MTIME,NLTIME,NQEV
C     STOP
C 910 WRITE(6,*)"ERROR..QUEUE OVFLW: IQ,IQOLD=",IQ,IQOLD
C     STOP
C 920 WRITE(6,*)"ERROR...EMPTY QUEUE: IQ,IQOLD=",IQ,IQOLD
C     WRITE(6,*)"Q1=",Q(1,I),I=1,MAXQ)
C     WRITE(6,*)"Q2=",Q(2,I),I=1,MAXQ)
C     WRITE(6,*)"Q3=",Q(3,I),I=1,MAXQ)
C     STOP
C 930 WRITE(6,*)"---NXTTP..IQNL=0,IQ,OLD,MAXQ,NL,TM,NBSY,NMCH"
C     +,IQNL,IQ,IQOLD,MAXQ,NLTIME,MTIME,NTBUSY,NTMACH
C     STOP
C 940 WRITE(6,*)"---NXTTP..MDT=0,IQ,NL,OLD=",IQ,IQNL,IQOLD
C     STOP
C     END
C
C
C
C LOGICAL FUNCTION LEVENT(THM)
C DIMENSION THM(2)
C LEVENT=.TRUE.
C LEVENT=THM(1).NE.0..OR..THM(2).NE.0.
C RETURN
C END
C
C
C
C SUBROUTINE INQUE(K)
C INQUE PUTS PARAMETERS AND TIME SEQUENCE OF THE FILL/DRAIN OPERATIONS
C COMPRISING LOAD TYPE K INTO THE TIME/EVENT QUEUE. INQUE ALSO UPDATES
C THE ARRAY WHICH KEEPS TRACK OF BUSY MACHINES BY A CALL TO FINQUE.
C
C ***NOTE THAT IF ALL MACHINES THAT RUN LDTP K ARE BUSY, A NEW K WILL
C ***BE GENERATED
C
C LOGICAL LBSY
C DIMENSION QC(3,59)
C COMMON/SYSPAR1/TMAIN,TSET,TAMB,MOPEN,MCLOSE,MDNITE
C COMMON/EVENTQ/ MAXQ,IQ,NQEV,Q(3,60)
C SEE FUNCTION NXTLD FOR DEFINITION OF /SYSPAR2/ VARIABLES
C COMMON/SYSPAR2/LDTPS,TMUNIT,JCYCLE(2,20),CYCLE(20),ICYCLE(3,5,2)
C COMMON/ENDCYC/ NTMACH,NTBUSY,MBSY(6),LDTYPE(30),MENDTM(30)
C COMMON/MACHIN/ NMCHTP,NMACH(6),ILDTP(6),MCHTYP(20)
C DEFINE OVERLAPPING ARRAY SO SO THAT QO(*,0)=Q(*,1) CAN BE REFERENCED
C EQUIVALENCE (Q(1,2),QC(1,1))
C
C
C
C CANT START LOAD IF ALL MACHINES OF DESIRED TYPE ARE BUSY!
C IF(NTBUSY.GE.NTMACH) RETURN
C IF(.NOT.LBSY(K)) GOTO 5
C 1 K=NXTLD(DUM)
C IF(LBSY(K)) GOTO 1
C
C
C
C CALCULATE THERMAL MASS OF FILL AND DRAIN FLOWS FOR CYCLE TYPE K
C 5 THMF=CYCLE(K)
C THMD=THMF
C NP4SES=JCYCLE(1,K)
C MERGEQ=IQ-1
C
C
C

```

```

C MERGE PARAMETERS FOR EA PHASE OF LC CYCLE INTO CIRCULAR TIME/EVENT QUEUE
  DO 10 J=1,NPHSES
C   L PARAMETERS
C   CONVERT FILL TMPTR CODE (1=C,2=W,3=H) TO MIX RATIO: FMIX=HOT/(HOT+COLD)
  FMIX=(ICYCLE(3,J,K)-1)*.5
  FILTMP=FMIX*TSET+(1.-FMIX)*TMAIN
  CALL INQUE1(MERGEQ,ICYCLE(1,J,K),THMF*FMIX,QO(1,MERGEQ))
C DRAIN PARAMETERS
  DRNTMP=FILTMP
  CALL INQUE2(MERGEQ,ICYCLE(2,J,K),THMD,DRNTMP,QO(1,MERGEQ))
C COULD MAKE MERGEQ COMMON TO /CINQUE/
C ALSO PASS FILL/DRAIN PARAMETERS (THMF*FMIX,THMD,DRNTMP) TO INQUE1,2
  VIA /CINQUE/ TO STREAMLINE CODE GENERATED FOR CALLING SEQUENCE.
  10 CONTINUE
C -----
C INSERT TYPE & TERMINATION TIME OF THIS LOAD INTO THE *CYCLE ACTIVE* STACK
  MCYCLE=MERGEQ-IQ+1
  IF(MCYCLE.LE.0) MCYCLE=MCYCLE+MAXQ
C ALTERNATIVELY:
  MCYCLE=MOD(MAXQ+MERGEQ+1-IQ,MAXQ)
  CALL FINQUE(K,MOD(MAXQ+MERGEQ-IQ,MAXQ)+1)
  WRITE(6,*)"INQ..LDTYP,NPHSES,MCYCLE=",K,NPHSES,MCYCLE
  CALL FINQUE(K,MCYCLE)
  CALL DEBUG(8)
  RETURN
END
C =====
C SUBROUTINE INQUE1(I,IC,THM,QREC)
C SLBR TO MERGE FILL PARAMETER, THM, INTO EVENTQ (QREC(1))
  LOGICAL LEVENT
  COMMON/EVENTQ/ MAXQ,IQ,NQEV,Q(3,60)
  DIMENSION QREC(3)
C IS IT A COLD FILL (THUS THMAS OF HOT WATER=C) ?
  IF(THM.EQ.0.) GOTO 10
  IF(.NOT.LEVENT(QREC)) NQEV=NQEV+1
  TOTHM=THM+QREC(1)
C CALCULATE NEW AGGREGATE THERMAL MASS
  QREC(1)=TOTHM
C BEFORE RETURN, SET EVENTQ TEMPORARY PCINTER TO NEXT DRAIN OPERATION TO BE
  MERGED INTO EVENTQ:
  10 I=MOD(I+IC,MAXQ)
  RETURN
END
C =====
C SUBROUTINE INQUE2(I,IC,THM,T,QREC)
C SUBR TO MERGE DRAIN PARAMETERS, THM,T, INTO EVENTQ (QREC(2),QREC(3))
  LOGICAL LEVENT
  COMMON/EVENTQ/ MAXQ,IC,NQEV,Q(3,60)
  DIMENSION QREC(3)
C IS IT A NULL DRAIN ? ...SHOULD NEVER HAPPEN!!
  IF(THM.EQ.0) WRITE(6,*)"---INQ2..DTHM=",THM
  IF(THM.EQ.C.) GOTO 10
  IF(.NOT.LEVENT(QREC)) NQEV=NQEV+1
  TOTHM=THM+QREC(2)
C CALCULATE NEW AGGREGATE DRAIN TEMPERATURE AND THERMAL MASS
  QREC(3)=(QREC(2)*QREC(3)+THM*T)/TOTHM
  QREC(2)=TOTHM
C BEFORE RETURN, SET EVENTQ TEMPORARY POINTER TO NEXT FILL OPERATION TO BE
  MERGED INTO EVENTQ:
  10 I=MOD(I+IC,MAXQ)
  RETURN
END
C =====
C LOGICAL FUNCTION LBUSY(LDTYP)
  COMMON/ENDCYC/ NTMACH,NTBUSY,MBUSY(5),LDTYPE(30),MENDTM(30)
  COMMON/MACHIN/ NMCHTP,NMACH(6),ILDTP(6),MCHTYP(20)
C COULD SIMPLY PASS MCHTYP(LDTYP) INSTEAD OF LDTYP...
C ...ALL THE MORE REASON TO PUT MBUSY(6) IN /MACHIN/
  J=MCHTYP(LDTYP)
  LBUSY=MBUSY(J).EQ.NMACH(J)
  RETURN
END

```

```

C - - - - -
C SUBROUTINE FINQUE(NEWLD,MCYCLE)
C COMMON/ENDCYC/ NTMACH,NTBUSY,MBUSY(6),LDTYPE(30),MENCTM(30)
C COMMON/MACHIN/ NMCHTP,NMACH(6),ILDTP(6),MCHTYP(20)
C COMMON/TIME/ MTIME
C DATA NTBUSY/0/
C NEWEND=MTIME+MCYCLE
C IP=1
C IF(NTBUSY.EQ.0) GOTO 50
C IF(NTBUSY.LT.0) WRITE(6,*)"---FINQUE...NTBUSY=",NTBUSY
C
C COUNT ENTRIES IN STACK THAT WILL STAY PUT (THOSE THAT PRECEDE NEW ENTRY)
C DO 10 IP=1,NTBUSY
C IF(MENDTM(IP).GT.NEWEND) GOTO 20
C 10 CONTINUE
C
C IP=NTBUSY+1
C GOTO 50
C
C PUSH DOWN REST OF ENTRIES TO MAKE ROOM FOR NEW ENTRY
C 20 NTBP=NTBUSY+IP
C DO 30 I=IP,NTBUSY
C J=NTBP-I
C JP=J+1
C LDTYPE(JP)=LDTYPE(J)
C 30 MENDTM(JP)=MENDTM(J)
C
C INSERT NEW ENTRY...CONTROL **ALWAYS** PASSES HERE BEFORE RETURN FROM FINQUE.
C 50 LDTYPE(IP)=NEWLD
C MENDTM(IP)=NEWEND
C NTBUSY=NTBUSY+1
C MBUSY(MCHTYP(NEWLD))=MBUSY(MCHTYP(NEWLD))+1
C IF(NTBUSY.NE.MBUSY(MCHTYP(NEWLD)))
C CALL DEBUG(9)
C RETURN
C END
C
C - - - - -
C SUBROUTINE FINDQ
C CHECK FOR CYCLE(S) FINISHING &
C AND DISCARD DEFUNCT CYCLES FROM STACK. (WE MUST ALLOW
C FOR (NKILL.GT.1) AS .GT.1 LOADS CAN END IN SAME TIME SLOT.)
C COMMON/ENDCYC/ NTMACH,NTBUSY,MBUSY(6),LDTYPE(30),MENDTM(30)
C COMMON/MACHIN/ NMCHTP,NMACH(6),ILDTP(6),MCHTYP(20)
C COMMON/TIME/ MTIME,NTIME,ICDWK
C COMMENT OUT LATER
C IF(NTBUSY.EQ.0) RETURN
C IF(NTBUSY.NE.MBUSY(1))
C + WRITE(6,*)"FINDQ1...NTBUSY,MBUSY=",NTBUSY,MBUSY(1),"(1)"
C IF(NTBUSY.LE.0) RETURN
C
C COUNT DEFUNCT CYCLES TO BE REMOVED
C DO WHILE (MENDTM(IP).LE.MTIME)
C (MAYBE SHOULD INHIBIT NEWLOAD TIL NXT TIME SLOT BY USING .GT.MTIME)
C DO 10 IP=1,NTBUSY
C IF(MENDTM(IP).GT.MTIME) GOTO 20
C ANY REASON WE CANT IMPLEMENT A MIN TIME BETWEEN LOADS BY DELAYING
C THE REMOVAL OF DEFUNCT CYCLES HERE ???
C J=MCHTYP(LDTYPE(IP))
C MBUSY(J)=MBUSY(J)-1
C 10 CONTINUE
C
C 20 NKILL=IP-1
C IF(NKILL.EQ.0) RETURN
C
C TIME WENT IN OPPOSITE DIRECTION IN STACK, ITS BASE PTR COULD SIMPLY BE
C INCREMENTED BY NKILL AND WE COULD RETURN AT THIS POINT, BUT.....
C FOR NOW:
C NTBUSY=NTBUSY-NKILL
C
C - - - - -

```



```

C POP (AND DISCARD) DEFUNCT CYCLES
  DO 30 I=1,NTBUSY
    LDTYPE(I)=LDTYPE(IP)
    MENDTM(I)=MENDTM(IP)
  30 IP=IP+1
    CALL DBUGQ(9)
  RETURN
END
C - - - - -
SUBROUTINE DEQUE(QRECRD,FTHMAS,DTHMAS,TWW)
  LOGICAL LEVENT
  DIMENSION QRECRD(3)
  COMMON/EVENTQ/ MAXQ,IC,NQEV,Q(3,60)
  IF(LEVENT(QRECRD)) NQEV=NQEV+1
  FTHMAS=QRECRD(1)
  DTHMAS=QRECRD(2)
  TWW=QRECRD(3)
  DO 10 I=1,3
  10 QRECRD(I)=0
  RETURN
END
C - - - - -
SUBROUTINE INIT
  LOGICAL PASS1
  COMMON/SYSPAR1/TMAIN,TSET,TAMB,MOPEN,MCLOSE,MONITE
  COMMON/SYSPAR2/LDTYPES,TMUNIT,JCYCLE(2,20),CYCLE(20),ICYCLE(3,5,20)
  COMMON/SYSPAR3/AVGTBL,TYPRBL(20)
  COMMON/HXPAR1/NODES,CSN,CWN,FS,FW,RWALL,ANODE,XL,CHNLS,CHNLW
  COMMON/TBLFCN/ DTBLF(25,2),WTBLF(7)
  COMMON/RUNPAR/ NDAYS
  COMMON/ENDCYC/ NTMACH,NTBUSY,MBUSY(6),LDTYPE(30),MENDTM(30)
  COMMON/MACHIN/ NMCHTP,NMACH(6),ILDTP(6),MCHTYP(20)
  COMMON/TIME/ MTIME,NLTIME,IDOWK
  DATA PASS1/.TRUE./
C
C UNITS: BTU,LBM,FT,HR,DEGF
C
C READ/COMPUTE H X P A R A M E T E R S
C
  IF(PASS1) REWIND 2
  PASS1=.FALSE.
C
C THIS STUFF (READ/COMPUTE HX PARAMETERS) SHOULD BE MOVED TO END TO MAINTAIN
C SIMILARITY OF FORM WITH PROGRAM HXSIM (6-7-82).
  20 READ(2,*)NODES,CSN,CWN,RWALL,ANODE,XL,CHNLS,CHNLW
  IF(EOF(2).NE.0) STOP
  WRITE(6,*)NODES,CSN,CWN,RWALL,ANODE,XL,CHNLS,CHNLW
  CSN=CSN/NODES
  FS=CSN/(CSN+CWN)
  CWN=CWN/NODES
  FW=CWN/(CSN+CWN)
C
C READ/COMPUTE P L A N T P A R A M E T E R S
C
  READ(2,*) TMAIN,TSET,TAMB,MOPEN,MCLOSE
  WRITE(4,*) TMAIN,TSET,TAMB,MOPEN,MCLOSE
  IF(EOF(2).NE.C) STOP
  WRITE(6,*) TMAIN,TSET,TAMB,MOPEN,MCLOSE
C
C READ/COMPUTE W A S H I N G M A C H I N E P A R A M E T E R S
C
  READ(2,*) TMUNIT
  WRITE(4,*) TMUNIT
  MONITE=24/TMUNIT
  IF(TMUNIT.LT.C) MONITE=-24*TMUNIT
  IF(TMUNIT.LT.C) TMUNIT=-1./TMUNIT
  WRITE(6,*) TMUNIT
  CUMM=0.
  LDTYPES=0
  NTMACH=0
  READ(2,*) NMCHTP
  WRITE(4,*) NMCHTP
  WRITE(6,*) NMCHTP
C - - - - -

```

C LOOP THROUGH LDYPES FOR EACH MCHTYPE

```

DO 120 MCH=1,NMCHTP
  READ(2,*) NMACH(MCH),NLT
  WRITE(4,*) NMACH(MCH),NLT
  WRITE(6,*) "MCHTYPE,NBRMACHINES,NBRLDYPES=",MCH,NMACH(MCH),NLT
  NTMACH=NTMACH+NMACH(MCH)
  DO 110 I=1,NLT
    LDTPS=LDTPS+1
    MCHTYP(LDTPS)=MCH
    READ(2,*) TYPRBL(LDTPS),CYCLE(LDTPS),NPHS
    WRITE(4,*) TYPRBL(LDTPS),CYCLE(LDTPS),NPHS
    CUMM=CUMM+TYPRBL(LDTPS)
    WRITE(6,*) TYPRBL(LDTPS),CYCLE(LDTPS),NPHS
    READ(2,*) ((ICYLE(J,K,LDTPS),J=1,3),K=1,NPHS)
    WRITE(4,90) ((ICYLE(J,K,LDTPS),J=1,3),K=1,NPHS)
    WRITE(6,90) ((ICYLE(J,K,LDTPS),J=1,3),K=1,NPHS)
  90 FORMAT(2I3,I2,(I5,I3,I2))

```

C CHECK VALIDITY OF PHASE PARAMETERS

```

DO 109 K=1,NPHS
  IF(ICYLE(3,K,LDTPS).LT.1.OR.ICYLE(3,K,LDTPS).GT.3)WRITE(6,*)
  + "INVALID FILL TEMPERATURE CODE (.NE.1,2,3)...LDTP,PHASE="
  + ,LDTPS,K
  DO 109 J=1,2
    IF(ICYLE(J,K,LDTPS).LT.2-J)WRITE(6,*)
    + "INVALID PHASE DURATION (.LT.0)...FILL/DRAIN,LDTP,PHASE="
    + ,J,LDTPS,K

```

109 CONTINUE

```

  JCYCLE(2,LDTPS)=MCH
  110 JCYCLE(1,LDTPS)=NPHS
  120 ILDTYP(MCH)=LDTPS
END LDTYPE,MCHTYP LOGPS
(NOTE: ILDTYP(NTMACH)=LDTPS= MAX ENTRIES IN CYCLE ACTIVE STACK)

```

C UNFORTUNATELY, NMACH(NMCHTP+1) GETS REFERENCED IN CBUGQ, SO SET

```

  NMACH(NMCHTP+1)=0
  WRITE(6,*) "NTMACH=",NTMACH
  WRITE(6,*) " NMACH=", ( NMACH(I),I=1,NMCHTP)
  WRITE(6,*) " ILDTYP=", (ILDTYP(I),I=1,NMCHTP)
  WRITE(6,*) " MCHTYP=", (MCHTYP(I),I=1,LDTPS)

```

C WRITE(6,*) "CUMM=",CUMM," CUMMTYPRBLITIES AS FOLLOWS:"

```

  TYPRBL(1)=TYPRBL(1)/CUMM
  IF(LDTPS.LE.1) GOTO 150
  DO 140 I=2,LDTPS
  140 TYPRBL(I)=TYPRBL(I-1)+TYPRBL(I)/CUMM
  150 CONTINUE
  WRITE(6,*)(TYPRBL(I),I=1,LDTPS)

```

C READ/COMPUTE DEMAND PARAMETERS

```

  READ(2,*) AVGTBL
  WRITE(4,*) AVGTBL
C READ HOURLY=ARRIVAL-RATE=FACTOR PROFILE. FIRST COLUMN IS WEEKDAY PROFILE;
C SECOND IS WEEKEND PROFILE. FUNCTION TBL INTERPOLATES BTWN HRLY VALUES.
  DO 210 IH=1,24
    READ(2,*) DTBLF(IH,1),DTBLF(IH,2)
  210 WRITE(4,*) DTBLF(IH,1),DTBLF(IH,2)
  CC 220 J=1,2
  220 DTBLF(25,J)=DTBLF(1,J)
  CCULD INTEGRATE DTBLF OVER (MOPEN,MCLOSE) AND THEN NORMALIZE

```

C READ DAILY ARRIVAL RATE FACTOR PROFILE (TAKEN TO BE CONSTANT OVER EACH DAY)

```

  READ(2,*) (WTBLF(IW),IW=1,7)
  WRITE(4,*) (WTBLF(IW),IW=1,7)

```

C READ/COMPUTE RUN PARAMETERS

```

  READ(2,*) NDAYS,IDGWK
  WRITE(4,*) NDAYS,IDGWK
  IDGWK=IDGWK-1
  WRITE(6,*) "AVG TM BTWN LOADS=",AVGTBL,"NDAYS TO SIMULATE=",NDAYS
  RETURN
END

```

A-17

```

PROGRAM HXSIM(INPUT=1018,OUTPUT=1018,TAPE7=1018,
+TAPE3=1018,TAPE5=INPUT,TAPE6=OUTPUT
+ ,TAPE4=1018)
COMMON/PUAPAR/ NDAYS
COMMON/SYSPAR1/TMAIN,TSET,TAMB,MOPEN,MCLCSE
COMMON/EVENTQ/ MAXQ,IQ,NQEV,Q(3,60)
10 CALL INIT
   DO 20 ID=1,NDAYS
     CALL DAYSIM
20 CONTINUE
   GOTO 10
END

C * * * * *
C SUBROUTINE DAYSIM
C REPAIRS TO FILL, CRT, QXT CALCS, N BECOMES NODES, ETC...3-23-82
C REWRITE DRAIN, CHANGE MEASR ONCE AGAIN. 3-30-82
COMMON/TIME/ MTIME,NLTIME,IDQWK
COMMON/SYSPAR1/TMAIN,TSET,TAMB,MOPEN,MCLOSE
COMMON/MEASR/ QAT,QST,QWT,QXST,QATT,QSTT,QWTT,CXSTT
CALL NEWDAY(TSAVG,TWAVG)
WRITE(6,*)"NEWDAY...TSAVG,TWAVG=",TSAVG,TWAVG
10 CONTINUE
   MSTEP=MTIME
   READ(4) MTIME,DTIME,FTHMAS,DTHMAS,TWW
   READ(4,90) MTIME,DTIME,FTHMAS,DTHMAS,TWW
90 FORMAT(1X,I5,F20.15,3F10.5)

C A SACRIFICIAL DUMMY RECORD HAS BEEN PROVIDED AT END OF EACH DAY
C IF(MTIME.LT.MSTEP) GOTO 100
C
C IF(FTHMAS.GT.0.) CALL FILL(FTHMAS,TMAIN,TSAVG)
C IF(DTHMAS.GT.0.) CALL DRAIN(DTHMAS,TWW,TWAVG)
C WRITE(3,*)"DAYSIM..TSAVG,TWAVG,DTIME=",TSAVG,TWAVG,DTIME
C CALL RELAX(DTIME,TSAVG,TWAVG)
C WRITE(3,*)"AFTER RELAX..TSAVG,TWAVG=",TSAVG,TWAVG
C IF(MTIME.LT.MCLOSET) GOTO 10
C IF(DTIME.GT.0.) GOTO 10

C AFTER CLOSING TIME(ACTUALLY,TIME OF LAST LD START) INHIBIT NEW LD STARTS...
C ...JUST LOOP UNTIL QUEUE IS EXHAUSTED.
C
100 DEMAND=QAT+QST
   IAT=QAT
   IST=QST
   IWT=QST
   IXST=QXST
   F1=QST/DEMAND
   F2=QXST/DEMAND
   WRITE(6,93) IAT,IST,IWT,IXST,F1,F2
   WRITE(3,93) IAT,IST,IWT,IXST,F1,F2
93 FORMAT(" MEASR..A,S,W,XS,F1,F2=",4I9,2F9.5)
   RETURN
   END

C * * * * *
C SUBROUTINE NEWDAY(TSAVG,TWAVG)
COMMON/MEASR/ QAT,QST,QWT,QXST,QATT,QSTT,QWTT,CXSTT
COMMON/TNODES/TS(101),TW(101)
COMMON/SYSPAR1/TMAIN,TSET,TAMB,MOPEN,MCLOSE
COMMON/HXPAR1/NODES,CSN,CWN,FS,FW,RWALL,ANGOE,XL,CHNLS,CHNLW,SUMCN
COMMON/TIME/ MTIME,NLTIME,IDQWK

C ZERO ACCUMULATORS FOR AUXILIARY (QAT) AND RECOVERED (QRT) HEAT
QAT=0.
QST=0.
QWT=0.
QXST=0.

C
C TSAVG=0.
C TWAVG=0.

C S E T I N I T I A L SUPPLY & WASTE WATER N O D E TEMPERATURES
DO 10 I=1,NODES
   TSAVG=TSAVG+TS(I)
   TWAVG=TWAVG+TW(I)
10 TSAVG=TSAVG/NODES
   TWAVG=TWAVG/NODES

C
C
C FETCH OR GENERATE TIME & TYPE OF FIRST LOAD
   MTIME=MCPEN
   RETURN
   END

```

```

C * * * * *
SUBROUTINE FILL (THMASS,TSUPPLY,TSAVG)
COMMON/MEASR/ QAT,QST,QWT,QXST,QATT,QSTT,QWTT,QXSTT
COMMON/TNODES/TS(101),TW(101)
COMMON/SYSPAR1/TMAIN,TSET,TAMB,MOPEN,MCLGSE
COMMON/HXPAR1/NODES,CSN,CWN,FS,FW,RWALL,ANODE,XL,CHNLS,CHNLW,SUMCN

```

```

C THRU=THMASS/CSN $NTHRU=THRU
C WRITE(6,*)"TRHU,NTHRU",THRU,NTHRU
FSW=THRU-NTHRU $FSNMW=1.-FSW
NM=NODES-NTHRU-1 $NM1=NODES-1 $NP1=NODES+1 $NTHRUP=NTHRU+1
C WRITE(6,*)"F,IMF,NM1,NP1,NTP",FSW,FSNMW,NM1,NP1,NTHRUP
C QS=TSAVG

```

```

C TSAVG=0. $IF(NM1.EQ.0) GOTO 30
DO 20 I=1,NM1
K=NP1-I
IF(NTHRUP-K) 11,12,13
11 TS(K)=FSW*TS(K-NTHRUP)+FSNMW*TS(K-NTHRU)
GOTO 20
12 TS(K)=FSW*TSUPPLY+FSNMW*TS(K-NTHRU)
WRITE(6,*)"K,TSK,TAV",K,TS(K),TSAVG
GOTO 20
13 TS(K)=TSUPPLY
WRITE(6,*)"K,TSK,TAV",K,TS(K),TSAVG
20 TSAVG=TSAVG+TS(K)
30 IF(1.LT.NTHRUP) GOTO 50
C THAT IS, IF(NTHRU.GT.0) GOTO 50
TS(1)=FSW*TSUPPLY+FSNMW*TS(1)
GOTO 60
50 TS(1)=TSUPPLY
60 TSAVG=(TSAVG+TS(1))/NODES
QS=(QS-TSAVG)*CSN*NODES
QST=QST+QS
QA=(TSET-TSUPPLY)*THMASS-QS
QAT=QAT+QA

```

```

C WRITE(3,*)"FILL..THMASS,FSW,NTHRU,QA,QS=",THMASS,FSW,NTHRU
C * QA,QS
C RETURN
C END

```

```

C * * * * *
SUBROUTINE DRAIN(THMASS,TDRAIN,TWAVG)
COMMON/SYSPAR1/TMAIN,TSET,TAMB,MOPEN,MCLGSE
COMMON/TNODES/TS(101),TW(101)
COMMON/HXPAR1/NODES,CSN,CWN,FS,FW,RWALL,ANODE,XL,CHNLS,CHNLW,SUMCN
COMMON/MEASR/ QAT,QST,QWT,QXST,QATT,QSTT,QWTT,QXSTT
C DEFINE THE MIXING FUNCTION
AMIX(T1,T2) = FCWW*T1 + FCWNM*T2

```

```

C IF(TDRAIN.LE.TMAIN) RETURN

```

```

C EXPRESS QTY OF WATER DRAINED IN TERMS OF THERMASS OF NODE (INTEGER+PART)
FTHRU=THMASS/CWN $NTHRU=FTHRU $FCWW=FTHRU-NTHRU $FCWNM=1.-FCWW

```

```

C QW=TWAVG $TWAVG=0.
C WRITE(3,*)"DRAIN..THMASS,FCWW,NTHRU=",THMASS,FCWW,NTHRU
DO 50 I=1,NODES
IF(TW(I).GE.TDRAIN) GOTO 50

```

```

C SET POINTER IC (LOWER OF TWO) NODES THAT MIX TO PRODUCE NEW TW(I)
NPTR=I+NTHRU
IF(NPTR-NODES) 10,20,30

```

```

10 TW(I)=AMIX(AMIN1(TW(NPTR+1),TDRAIN),AMIN1(TW(NPTR),TDRAIN))
GOTO 50
20 TW(I)=AMIX(TDRAIN,AMIN1(TW(NPTR),TDRAIN))
GOTO 50

```

```

C HIGHEST NTHRU NODES WHICH ARE COOLER THAN TDRAIN FILLED A TMPRTUR TDRAIN
30 TW(I)=TDRAIN
GOTO 50

```

```

50 TWAVG=TWAVG+TW(I)
TWAVG=TWAVG/NODES
QW=(TWAVG-QW)*CWN*NODES
QWT=QWT+QW

```

```

C WRITE(3,*)"DRAIN..THMASS,FCWW,NTHRU,QW=",THMASS,FCWW,NTHRU,QW
C RETURN
C END

```

```

C *****
C SUBROUTINE RELAX(OTIME, TSAVG, TWAvg)
C ONE U-VALUE USED FOR VERTICAL EXTENT OF HEAT EXCHANGER WALL. INITIAL
C TEMPERATURE DIFFERENCE COMPUTED BETWEEN EACH PAIR (SUPPLY, WASTE WATER
C NODES) OF NODES; FINAL NODE TEMPERATURES AT END OF RELAXATION PERIOD
C THEN COMPUTED. HEAT TRANSFERRED ACCUMULATED (BASED ON CHANGE IN AVERAGE
C SUPPLY WATER NODE TEMPERATURE) IN QXST.
COMMON/FREECV/YSNUSL, YWNUSL, SK, WK, XITNS, XITNW, SRHC, WRHO
COMMON/TNODES/TS(101), TW(101)
COMMON/SYSPAR1/TMAIN, TSET, TAMB, MOPEN, MCLOSE
COMMON/MEASR/QAT, QST, QWT, QXST, QATT, QSTT, QWTT, QXSTT
COMMON/HXPAR1/NODES, CSN, CWN, FS, FW, RWALL, ANODE, XL, CHNLS, CHNLW, SUMCN
C FS=CSN/(CSN+CWN) $ FW=CWN/(CSN+CWN)
C DTEMP=TWAvg-TSAvg
C CALL RFREE2(TSAvg, TWAvg)
C LAX=ANODE/(RFREE(1)+RFREE(2)+RWALL)
C UAX=ANODE/(RFREE(1,WS,DTEMP)+RFREE(1,WW,DTEMP)+RWALL)
C LATER EVALUATE AVG BODY FORCES BY CALL TO RFREE3, THEN COMPUTE SRFREE
C AND WRFREE FOR EACH NODE
C SHOULD DEFINE A CRITICAL VALUE OF E, EC(.LT.0), SUCH THAT E.LT.EC WE
C SET DTIME=DTIME+EC*SUMCN AND SET E=EC. (BETTER DEFINE DTEMP, ETEMP TO
C CLARIFY WHAT IS GOING ON.)
E=-UAX*DTIME/SUMCN
X=1.
IF(E.GT.-200) X=1.-EXP(E)
WRITE(3,91) (TW(I), I=1, NODES)
WRITE(3,91) (TS(I), I=1, NODES)
5 QXS=TSavg
TSAvg=0. $ TWAvg=0.
WRITE(6,*) "RX...UA, DTM, DTP, X, TSV, TWV=", UAX, DTIME, DTEMP, X, QXS
WRITE(3,*) "RX...UA, DTM, DTP, X, TSV, TWV=", UAX, DTIME, DTEMP, X, QXS
DO 10 I=1, NODES
DDTMP1=(TW(I)-TS(I))*X
WRITE(3,*) "DDT, X=", DDTMPI, X
TW(I)=TW(I)-DDTMP1*FS
TS(I)=TS(I)+DDTMP1*FW
TSAvg=TSAvg+TS(I) $ TWAvg=TWAvg+TW(I)
10 CONTINUE
TSAvg=TSAvg/NODES $ TWAvg=TWAvg/NODES
WRITE(3,91) " TW ", TWAvg, (TW(I), I=1, NODES)
WRITE(3,91) " TS ", TSAvg, (TS(I), I=1, NODES)
QXST=QXST+(TSAvg-QXS)*CSN*NODES
RETURN
C *****
C VERTICAL CONDUCTION SHOULD BE TREATED IN NEXT GENERATION MODEL. TWO WAYS
C 1. FULL MATPIX SOLUTION; 5-BANDED MATRIX MUST BE DEFINED AND INVERTED AT
C EACH PHASE STEP;
C 2. ALTERNATE HORIZONTAL AND VERTICAL HEAT TRANSFERS SEPARATELY, USE FIXED,
C SMALLR TIME STEP DURING RELAXATION; SOLVE HZ, THEN VT TRANSFER AT EACH TIME
C TIME STEP; ASSUME VT TRANSFER IS PURE CONDUCTION; W/FIXED K, CAN INVERT
C MATRIX JUST ONCE; (IS MATRIX SAME FOR BOTH SIDES OF HEAT EXCHANGER??).
C 91 FORMAT(1X,A4,(1X,9F7.2/))
C END
C *****
C FUNCTION RFREE(ISIDE)
C RFREE RETURNS R-VALUE OF FREE CONVECTION BOUNDARY LAYER ON SUPPLY (ISIDE=1)
C OR WASTE (ISIDE=2) WATER SIDE OF HEAT EXCHANGER JACKET.
COMMON/FREECV/YSNUSL, YWNUSL, CNDVTY(2), XITNS, XITNW, SRHC, WRHO
COMMON/HXPAR1/NODES, CSN, CWN, FS, FW, RWALL, ANODE, XL, CHNL(2), SUMCN
IF(XL.GT.XITNS) GOTO 10
Y=.0925*(XL/XITNS)**(.25)
C FOR INTERIOR VESSEL WALL SHOULD USE FORM (FROM ALTAS PAPER):
Y=.0925*.01155*XL/XITNS
GOTO 20
Y=.129-.0365*XITNS/XL
Y=(.0925*XITNS+.129*(XL-XITNS))/XL (DIMENSIONLESS)
C FOR INTERIOR VESSEL WALL SHOULD USE FORM (FROM ALTAS PAPER):
Y=.129-.048*XITNS/XL = (.0809*XITNS + .129*(XL-XITNS))/XL
20 RFREE=AMIN1(YSNUSL/Y, CHNL(ISIDE)/4.)/CNDVTY(ISIDE)
WRITE(3,*) "XITNS, XL, Y, YSNUSL=", XITNS, XL, Y, YSNUSL
RETURN
C END

```

```

C *****
C SUBROUTINE RFREE2(TWAVG, TSAVG)
C COMMON/FREECV/YSNUSL, YWNUSL, SK, WK, XITNS, XITNW, SRHC, WRHC
C E CONST VISC*THDIFSVTY PRODUCT FOR NOW
C DATA LAPRCD, SK, WK/.25E-10, .353, .353/
C UNITS: BTU, LBM, FT, HR, DEGF
C
C ASSUME FILM-BULK DENSITY DIFFERENCE IS EQUALLY DIVIDED BETWEEN
C SUPPLY AND WASTE WATER SIDES OF HEAT EXCHANGER:
C SRHO=ABS(DELRHO(TWAVG, TSAVG)/2.)
C SRHO=AMAX1(SRHO, .1E-20)
C YSNUSL=(UAPRCD/SRHO)**.33333333
C XITNS=1000.*YSNUSL
C RETURN
C END
C *****
C FUNCTION DELRHO(T1, T2)
C RETURNS DENSITY DIFFERENCE IN LBM/FT3
C FOR T1, T2 GIVEN IN DEGF.
C X1=39.1-.05555556*T1
C X2=39.1-.05555556*T2
C DELRHO=-0.68745*(X1-X2)
C +14.604*(SORT(X1)-SORT(X2))
C +7.9058*(X1**-.33333333-X2**-.33333333)
C WRITE(3,*)"DELRHO=", DELRHO
C RETURN
C END
C *****
C SUBROUTINE INIT
C LOGICAL PASS1
C DIMENSION MIDLIM(6), DIMLIM(6)
C COMMON/SYSPAR1/TMAIN, TSET, TAMB, MOPEN, MCLOSE
C COMMON/SYSPAR2/LDTYPS, TMUNIT, JCYCLE(2,20), CYCLE(20), ICYCLE(3,5,20)
C COMMON/SYSPAR3/AVGTBL, TYPRBL(20)
C COMMON/TBLFCN/ DTBLF(25,2), WTBLF(7)
C COMMON/HXPAR1/NODES, CSN, CWN, FS, FW, RWALL, ANODE, XL, CHNLS, CHNLW, SUMCN
C COMMON/TNODES/TS(101), TW(101)
C COMMON/RUNPAR/ NDAYS
C COMMON/ENDCYC/ NTMACH, NTBUSY, MBUSY(6), LDTYPE(30), MENDTM(30)
C COMMON/MACHIN/ NMCHTP, NMACH(6), ILDTYP(6), MCHTYP(20)
C DATA MIDLIM/1C1, 6C, 3C, 5, 6, 20/
C DATA DIMLIM/4HNODE, 4H QUE, 4HSTAK, 4H PHS, 4HMCTP, 4HLDTP/
C DATA PASS1/.TRUE./
C
C READ/COMPUTE PLANT PARAMETERS
C
C IF(PASS1) REWIND 4
C REWIND 4
C TAMB SHOULD BE READ FROM TAPE7
C READ(4,*) TMAIN, TSET, TAMB, MOPEN, MCLOSE
C IF(EOF(4).EQ.C) GOTD 51
C WRITE(6,*) "EOF ON TAPE4"
C STOP
C 51 CONTINUE
C IF(PASS1)
C +WRITE(3,*) TMAIN, TSET, TAMB, MOPEN, MCLOSE
C IF(PASS1)
C +WRITE(6,*) TMAIN, TSET, TAMB, MOPEN, MCLOSE
C
C READ/COMPUTE WASHING MACHINE PARAMETERS
C
C READ(4,*) TMUNIT
C IF(PASS1)
C +WRITE(3,*) TMUNIT
C IF(TMUNIT.LT.0.) TMUNIT=-1./TMUNIT
C IF(PASS1)
C +WRITE(6,*) TMUNIT
C CUMM=0.
C LDTYPS=0
C NTMACH=0
C READ(4,*) NMCHTP
C IF(PASS1)
C +WRITE(3,*) NMCHTP
C IF(PASS1)
C +WRITE(6,*) NMCHTP

```

```

C CHECK MACHINE TYPE LIMIT
M=5
IF(NMCHTP.GE.MIDLIM(M)) GOTO 900
C LOOP THROUGH LD TYPES FOR EACH MCHTYPE
DO 120 MCH=1,NMCHTP
  READ(4,*) NMACH(MCH),NLT
  IF(PASS1)
    +WRITE(3,*) NMACH(MCH),NLT
  IF(PASS1)
    +WRITE(6,*) "MCHTYPE,NBRMACHINES,NBRLDTYPES=",MCH,NMACH(MCH),NLT
  NTMACH=NTMACH+NMACH(MCH)
  DO 110 I=1,NLT
    LDTPS=LDTPS+1
    MCHTYP(LDTPS)=MCH
    READ(4,*) TYPRBL(LDTPS),CYCLE(LDTPS),NPHS
    IF(PASS1)
      +WRITE(3,*) TYPRBL(LDTPS),CYCLE(LDTPS),NPHS
    IF(PASS1)
      +WRITE(6,*) TYPRBL(LDTPS),CYCLE(LDTPS),NPHS
C CHECK PHASES PER CYCLE LIMIT
M=4
IF(NPHS.GT.MIDLIM(M)) GOTO 900
CUMM=CUMM+TYPRBL(LDTPS)
READ(4,*) ((ICYCLE(J,K,LDTPS),J=1,3),K=1,NPHS)
IF(PASS1)
  +WRITE(3,*) ((ICYCLE(J,K,LDTPS),J=1,3),K=1,NPHS)
IF(PASS1)
  +WRITE(6,*) ((ICYCLE(J,K,LDTPS),J=1,3),K=1,NPHS)
C CHECK VALIDITY OF PHASE PARAMETERS
NDUR=0
DO 109 K=1,NPHS
  IF(ICYCLE(3,K,LDTPS).LT.1.OR.ICYCLE(3,K,LDTPS).GT.3) WRITE(6,*)
  + "INVALID FILL TEMPERATURE CODE (.NE.1,2,3)...LDTYP,PHASE="
  + ,LDTPS,K
  DO 109 J=1,3
    IF(ICYCLE(J,K,LDTPS).LT.2-J) WRITE(6,*)
    + "INVALID PHASE DURATION (.LT.0)...FILL/DRAIN,LDTYP,PHASE="
    + ,J,LDTPS,K
    NDUR=NDUR+ICYCLE(J,K,LDTPS)
109 CONTINUE
C CHECK QUEUE LIMIT
M=2
IF(NDUR.GE.MIDLIM(M)) GOTO 900
JCYCLE(2,LDTPS)=MCH
110 JCYCLE(1,LDTPS)=NPHS
120 ILDTYP(MCH)=LDTPS
C END LDTYPE, MCHTYP LOOPS
C (NOTE: ILDTYP(NTMACH)=LDTPS= MAX ENTRIES IN CYCLE ACTIVE STACK)
C CHECK STACK LIMIT
M=3
IF(NTMACH.GE.MIDLIM(M)) GOTO 900
C UNFORTUNATELY, NMACH(NMCHTP+1) GETS REFERENCED IN CBUGQ, SO SET
  NMACH(NMCHTP+1)=0
  IF(PASS1)
    +WRITE(6,*) "NTMACH,LDTPS=",NTMACH,LDTPS
C CHECK CYCLE TYPE LIMIT
M=6
IF(LDTPS.GT.MIDLIM(M)) GOTO 900
IF(PASS1)
  +WRITE(6,*) "NMACH=", (NMACH(I),I=1,NMCHTP)
IF(PASS1)
  +WRITE(6,*) "ILDTYP=", (ILDTYP(I),I=1,NMCHTP)
IF(PASS1)
  +WRITE(6,*) "MCHTYP=", (MCHTYP(I),I=1,LDTPS)
C
  IF(PASS1)
    +WRITE(6,*) "CUMM=",CUMM," CUMMTYPRBLITIES AS FOLLOWS:"
    TYPRBL(1)=TYPRBL(1)/CUMM
    IF(LDTPS.LE.1) GOTO 150

```

```

DO 140 I=2,LDTPS
140  TYPBRL(I)=TYPBRL(I-1)+TYPBRL(I)/CUMM
60  CONTINUE
    IF(PASS1)
      +WRITE(6,*) (TYPBRL(I),I=1,LDTPS)
C READ/COMPUTE DEMAND PARAMETERS
C READ(4,*) AVGTBL
    IF(PASS1)
      +WRITE(3,*) AVGTBL
C READ HOURLY ARRIVAL RATE FACTOR PROFILE. FIRST COLUMN IS WEEKDAY PROFILE;
C SECOND IS WEEKEND PROFILE. FUNCTION TBL INTERPOLATES BTWN HRLY VALUES.
    DO 210 IH=1,24
      READ(4,*) DTBLF(IH,1),DTBLF(IH,2)
      IF(PASS1)
        +WRITE(3,*) DTBLF(IH,1),DTBLF(IH,2)
210  CONTINUE
    DO 220 J=1,2
      220  DTBLF(25,J)=DTBLF(1,J)
C COULD INTEGRATE DTBLF OVER (MOPEN,MCLOSE) AND THEN NORMALIZE
C READ DAILY ARRIVAL RATE FACTOR PROFILE (TAKEN TO BE CONSTANT OVER EACH DAY)
    READ(4,*) (WTBLF(IW),IW=1,7)
    IF(PASS1)
      +WRITE(3,*) (WTBLF(IW),IW=1,7)
C READ/COMPUTE RUN PARAMETERS
    READ(4,*) NDAYS,IDOWK
    IF(PASS1)
      +WRITE(3,*) NDAYS,IDOWK
      IDOWK=IDOWK-1
      IF(PASS1)
        +WRITE(6,*) "AVG TM BTWN LOADS=",AVGTBL,"NDAYS TO SIMULATE=",NDAYS
C UNITS: BTU,LBM,FT,HR,DEGE
C READ/COMPUTE HX PARAMETERS
    IF(PASS1) REWIND 7
20  READ(7,*) NODES,CSN,CWN,RWALL,ANODE,XL,CHNLS,CHNLW
    IF(EOF(7).EQ.C) GOTO 21
    WRITE(6,*) "EOF ON TAPE 7"
    STOP
21  CONTINUE
    WRITE(6,*) NODES,CSN,CWN,RWALL,ANODE,XL,CHNLS,CHNLW
    WRITE(3,*) NODES,CSN,CWN,RWALL,ANODE,XL,CHNLS,CHNLW
    IF(NODES.LE.C) STOP
C CHECK NODE LIMIT
    M=1
    IF(NODES.GE.MIDLIM(M)) GOTO 900
    ANODE=ANODE/NODES
    CSN=CSN/NODES
    CWN=CWN/NODES
    SUMCN=CSN+CWN
    FS=CSN/(SUMCN)
    FW=CWN/(SUMCN)
    READ(7,*) TSAVG,TWAVG
    DO 40 I=1,NODES
      TS(I)=TSAVG
      TW(I)=TWAVG
40  READ(7,*) NDAYS7,IDOWK7
      NDAYS=M*INC(NDAYS,NDAYS7)
      IF(IDOWK7.NE.IDOWK) WRITE(6,*) "IDOWK7.NE.IDOWK",IDOWK7,IDOWK
      IF(IDOWK7.NE.IDOWK) WRITE(3,*) "IDOWK7.NE.IDOWK",IDOWK7,IDOWK
      PASS1=.FALSE.
    RETURN
C DIAGNOSTIC RESPONSE TO INPUT THAT WOULD RESULT IN ARRAY OVERFLOW
90  WRITE(6,990) DIMLIM(M),MIDLIM(M)
    WRITE(3,990) DIMLIM(M),MIDLIM(M)
990  FORMAT(" ****FATAL ERROR. ILLEGAL INPUT WILL CAUSE ",A4,
      + " OVERFLOW...LIMIT=",I3)
    STOP
END

```


SIMULATION OF THE INFINITE NTU HEAT EXCHANGER

A slightly modified version of HXSIM, called IXSIM, is used to simulate the performance of a hypothetical "infinite NTU" heat exchanger. In such a heat exchanger all paired nodes (i.e. each waste water node and its corresponding supply water node) come to equilibrium during the relaxation period no matter how short that period is. Subroutine RELAX and its subordinate subroutines RFREE, RFREE2, and DELRHO, simulate this aspect of the heat exchanger's behavior. The dimensionless variable, X , in RELAX, quantifies the progress towards equilibrium attained during the relaxation period; $X = 0$ indicates no progress (no change in node temperatures, as would occur if $DTIME = 0$ or $ANODE = 0$), and $X = 1$ indicates complete achievement of equilibrium (temperatures of waste and corresponding supply water node become equal, as would occur if either $DTIME$ or $ANODE$ were infinite). The infinite NTU model is therefore implemented by simply replacing the statement $DDTMPI = (TW(I) - TS(I)) * X$ in subroutine RELAX with $DDTMPI = (TW(I) - TS(I))$, which, in effect, sets $X = 1$. The statements that calculate X for the finite NTU heat exchanger are bypassed. This eliminates references to subroutines RFREE, RFREE2 and DELRHO. Thus the modified program for simulating the infinite NTU heat exchanger, program IXSIM, runs much faster than the finite NTU program, HXSIM. The version of subroutine RELAX used in program IXSIM follows.

ALTERNATIVE VERSION OF RELAX THAT DISTINGUISHES IXSIM FROM HXSIM

```

C *****
C SUBROUTINE RELAX(DTIME, TSAVG, TWAVG)
C ONE U=VALUE USED FOR VERTICAL EXTENT OF HEAT EXCHANGER WALL. INITIAL
C TEMPERATURE DIFFERENCE COMPUTED BETWEEN EACH PAIR (SUPPLY, WASTE WATER
C SIDES) OF NODES; FINAL NODE TEMPERATURES AT END OF RELAXATION PERIOD
C WHEN COMPUTED. HEAT TRANSFERRED ACCUMULATED (BASED ON CHANGE IN AVERAGE
C SUPPLY WATER NODE TEMPERATURE) IN QXST.
COMMON/FREECV/YSNUSL, YWNUSL, SK, WK, XITNS, XITNW, SRHC, WRHO
COMMON/TNODES/TS(101), TW(101)
COMMON/SYSPAR1/TMAIN, TSET, TAMB, MOPEN, MCLOSE
COMMON/MEASR/QAT, QST, QWT, QXST, QATT, QSTT, QWTT, QXSTT
COMMON/HXPAR1/NODES, CSN, CWN, FS, FW, RWALL, ANODE, XL, CHNLS, CHNLW, SUMCN
GOTO 5

C FS=CSN/(CSN+CWN) $ FW=CWN/(CSN+CWN)
C DTEMP=TWAVG-TSAVG
C CALL RFREE2(TSAVG, TWAVG)
C UAX=ANODE/(RFREE(1)+RFREE(2)+RWALL)
C UAX=ANODE/(RFREE(H, WS, DTEMP)+RFREE(L, WW, DTEMP)+RWALL)
C LATER EVALUATE AVG BODY FORCES BY CALL TO RFREE3, THEN COMPUTE SRFREE
C AND WRFREE FOR EACH NODE
C
C SHOULD DEFINE A CRITICAL VALUE OF E, EC(.LT.0), SUCH THAT E.LT.EC WE
C SET DTIME=DTIME+EC*SUMCN AND SET E=EC. (BETTER DEFINE DTEMP, ETEMP TC
C CLARIFY WHAT IS GOING ON.)
E=UAX*DTIME/SUMCN
X=1.
IF(E.GT.=200) X=1.-EXP(E)
WRITE(3,91) (TW(I), I=1, NODES)
WRITE(3,91) (TS(I), I=1, NODES)
5 QXS=TSAVG
TSAVG=0. $ TWAVG=0.
WRITE(6,*)"RX...UA,DTM,DTP,X,TSV,TWV=",UAX,DTIME,DTEMP,X,QXS
WRITE(3,*)"RX...UA,DTM,DTP,X,TSV,TWV=",UAX,DTIME,DTEMP,X,QXS
DO 10 I=1,NODES
COTMPI=(TW(I)-TS(I))*X
DDTMP1=(TW(I)-TS(I))
WRITE(3,*)"COT,X=",COTMPI,X
TW(I)=TW(I)-DDTMP1*FS
TS(I)=TS(I)+DDTMP1*FW
TSAVG=TSAVG+TS(I) $ TWAVG=TWAVG+TW(I)
10 CONTINUE
TSAVG=TSAVG/NODES $ TWAVG=TWAVG/NODES
WRITE(3,91) "TW",TWAVG,(TW(I), I=1, NODES)
WRITE(3,91) "TS",TSAVG,(TS(I), I=1, NODES)
QXST=QXST+(TSAVG-QXS)*CSN*NODES
RETURN

C *****
C VERTICAL CONDUCTION SHOULD BE TREATED IN NEXT GENERATION MODEL. TWO WAYS
C 1. FULL MATRIX SOLUTION; 2-BANDED MATRIX MUST BE DEFINED AND INVERTED AT
C EACH PHASE STEP;
C 2. ALTERNATE HORIZONTAL AND VERTICAL HEAT TRANSFERS SEPARATELY, USE FIXED,
C SMALLER TIME STEP DURING RELAXATION; SOLVE HZ, THEN VT TRANSFER AT EACH TIME
C TIME STEP; ASSUME VT TRANSFER IS PURE CONDUCTION; w/FIXED K, CAN INVERT
C MATRIX JUST ONCE; (IS MATRIX SAME FOR BOTH SIDES OF HEAT EXCHANGER??).
C
C 91 FORMAT(1X,A4,(1X,9F7.2/))
C END
C *****
C FUNCTION RFREE(ISIDE)
C RFREE RETURNS R=VALUE OF FREE CONVECTION BOUNDARY LAYER CN SUPPLY (ISIDE=1)
C OR WASTE (ISIDE=2) WATER SIDE OF HEAT EXCHANGER JACKET.
COMMON/FREECV/YSNUSL, YWNUSL, CNDVTY(2), XITNS, XITNW, SRHC, WRHO
COMMON/HXPAR1/NODES, CSN, CWN, FS, FW, RWALL, ANODE, XL, CHNL(2), SUMCN
IF(XL.GT.XITNS) GOTO 10
Y=.0925*(XL/XITNS)**(.25)
C FOR INTERIOR VESSEL WALL SHOULD USE FORM (FROM ALTAS PAPER):
Y=.0925-.01155*XL/XITNS
GOTO 20
10 Y=.129-.0365*XITNS/XL
Y=(.0925*XITNS+.129*(XL-XITNS))/XL (DIMENSIONLESS)
C FOR INTERIOR VESSEL WALL SHOULD USE FORM (FROM ALTAS PAPER):
Y=.129-.046*XITNS/XL = (.0609*XITNS + .129*(XL-XITNS))/XL
20 RFREE=AMIN1(YSNUSL/Y, CHNL(ISIDE)/4.)/CNDVTY(ISIDE)
WRITE(3,*)"XITNS,XL,Y,YSNUSL=",XITNS,XL,Y,YSNUSL
RETURN
C END

```

APPENDIX B
COMPUTER SIMULATION RESULTS

Table B.1 Infinite NTU, 20 node model, flat hot-warm-cold driving series, AVGTBL = 12 minutes

		C_s/C_w					
		1.00	.90	.75	.50	.25	.125
$C_s + C_w$ (Btu/F)	200	.314	.320	.328	.304	.229	.152
	400	.478	.480	.480	.461	.387	.258
	800	.596	.597	.597	.588	.529	.420
	1600	.660	.660	.660	.656	.627	.556
	3200	.677	.677	.677	.676	.669	.640

TABLE OF HEAT RECOVERY FRACTIONS (f)

$$f = \frac{\text{RECOVERED HEAT}}{\text{RECOVERED} + \text{AUXILIARY HEAT}}$$

Table B.2 Infinite NTU, 20 node model, flat warm-cold driving series, AVGTBL = 12 minutes

		C_s/C_w					
		1.00	.90	.75	.50	.25	.125
$C_s + C_w$ (Btu/F)	200	.323	.322	.316	.284	.199	.119
	400	.570	.565	.552	.497	.363	.226
	800	.741	.738	.731	.702	.596	.402
	1600	.862	.861	.867	.839	.764	.635
	3200	.928	.928	.924	.913	.871	.788

HEAT RECOVERY FRACTIONS (f)

Table B.3 Simulation results for 20 node, finite NTU model,
with warm-cold load, profile, AVGTBL = 24 minutes,
R value of wall = .01

		C_s/C_w			
		1.0	.5	.25	
AREA (ft ²)	50	.2746	.2523	.1996	$C_s + C_w = 800$ Btu/F
	100	.3797	.3526	.2870	
	200	.4928	.4614	.3836	
	400	.5983	.5631	.4740	

		C_s/C_w			
		1.0	.5	.25	
AREA (ft ²)	50	.3073	.2845	.2250	$C_s + C_w = 1600$ Btu/F
	100	.4230	.3962	.3345	
	200	.5416	.5134	.4348	
	400	.6514	.6241	.5443	

Tables B.4, B.5 Heat recovery fraction (f) results for
finite NTU, 20 node, warm-cold fill, profile,
AVGTBL = 12 minutes

	R_{wall}	$C_s + C_w$	AREA (ft ²)	$C_s/C_w = 1.0$	$C_s/C_w = .5$	$C_s/C_w = .25$
TABLE B.04	.013 (PVC)	800	50	.2618	.2402	.1891
			100	.3656	.3389	.2747
			200	.4787	.4477	.3711
			400	.5867	.5518	.4636
		1600	50	.2931	.2708	.2131
			100	.4079	.3815	.3108
			200	.5276	.4992	.4208
			400	.6396	.6119	.5315
	.000021 (Lopper)	800	50	.3298	.3052	.2467
			100	.4384	.4096	.3394
			200	.5476	.5151	.4334
			400	.6407	.6044	.5122
		1600	50	.3678	.3431	.2777
			100	.4835	.4563	.3822
			200	.5957	.5686	.4912
			400	.6951	.6694	.5925

Table B.6

RESULTS for: ☒ profile ☒ W, C fills ☐ AVGTBL = 6 ☐ time:
 Driving function: ☐ flat ☐ H, W, C fills ☒ AVGTBL = 12 ☐ CLR 4H

$R_{wall} = .013$ Grad/F-ft² (2" PVC, Schedule 40) file: 7H

N	C _s +C _w (ft ² /F)	A (ft ²)	C _s /C _w =			
			1.00	.80	.50	.25
1	800	50	.2262		.2129	.1759
		100	.3006		.2878	.2482
		200	.3736		.3631	.3270
		400	.4368		.4294	.3996
		800				
	1600	50	.2337		.2205	.1843
		100	.3028		.2902	.2539
		200	.3657		.3557	.3245
		400	.4180		.4113	.3887
		800				
	3	50	.2599		.2397	.1978
		100	.3615		.3379	.2720
		200	.4708		.4452	.3663
		400	.5738		.5484	.4565
		800				
	1600	50	.2824		.2640	.2069
		100	.3877		.3681	.2930
		200	.4943		.4766	.3984
		400	.5915		.5785	.4925
		800				
7	800	50	.2617		.2398	.1988
		100	.3655		.3382	.2741
		200	.4785		.4463	.3700
		400	.5862		.5495	.4619
		800				
	1600	50	.2911		.2703	.2126
		100	.4040		.3804	.3092
		200	.5210		.4975	.4190
		400	.6298		.6093	.5291
		800				
20	800	50	.2618		.2402	.1891
		100	.3656		.3389	.2742
		200	.4787		.4477	.3711
		400	.5867		.5518	.4636
		800				
	1600	50	.2931		.2708	.2131
		100	.4079		.3815	.3109
		200	.5276		.4992	.4208
		400	.6396		.6119	.5315
		800				

Table B.7

RESULTS for ☒ profile ☒ W, C fills ☐ AVGTBL = 6
 Driving Function: ☒ flat ☐ H, W, C fills ☒ AVGTBL = 12 minutes Name: CLR44

$R_{WALL} = .000021$ Bink/F. ft² ($\frac{3}{4}$ " ϕ Copper.) file 7I

N	G _s +G _w (Btu/°F)	A (ft ²)	c _s /c _w =			
			1.00	.80	.50	.25
1	800	50	.2738		.2612	.2241
		100	.3460		.3352	.3000
		200	.4120		.4042	.3744
		400	.4646		.4594	.4358
		800				
	1600	50	.2769		.2645	.2292
		100	.3405		.3299	.2981
		200	.3954		.3877	.3633
		400	.4396		.4350	.4191
		800				
3	800	50	.3215		.3044	.2444
		100	.4218		.4079	.3352
		200	.5364		.5122	.4267
		400	.6244		.6008	.5036
		800				
	1600	50	.3505		.3317	.2668
		100	.4544		.4365	.3626
		200	.5527		.5346	.4607
		400	.6382		.6292	.5502
		800				
7	800	50	.3297		.3046	.2462
		100	.4382		.4084	.3384
		200	.5472		.5131	.4319
		400	.6400		.6014	.5102
		800				
	1600	50	.3644		.3422	.2768
		100	.4778		.4548	.3806
		200	.5821		.5662	.4889
		400	.6835		.6661	.5896
		800				
20	800	50	.3298		.3052	.2467
		100	.4384		.4096	.3394
		200	.5476		.5151	.4334
		400	.6407		.6044	.5122
		800				
	1600	50	.3678		.3431	.2777
		100	.4835		.4563	.3822
		200	.5957		.5686	.4912
		400	.6951		.6694	.5925
		800				

Table B.8

RESULTS for Driving Function: ☒ profile ☒ W,C A'lls ☒ AVG TBL = 6 ☒ Name: 2R4H
☐ flat ☐ H,W,C A'lls ☒ AVG TBL = 12 minutes

$R_{wall} = .11123$ Blad/O.F. $1/8"$ (1" stainless tube) file 7J

N	G+G _w (F ₀ /F)	A(F ₀)	C _w /C ₀ =			
			1.00	.80	.50	.25
1	800	50	.2729		.2602	.2231
		100	.3451		.3343	.2990
		200	.4113		.4035	.3735
		400	.4641		.4588	.4351
		800				
	1600	50	.2761		.2636	.2283
		100	.3398		.3291	.2972
		200	.3948		.3872	.3626
		400	.4392		.4346	.4186
		800				
3	800	50	.3251		.3021	.2432
		100	.4304		.4065	.3339
		200	.5352		.5109	.4257
		400	.6235		.5992	.5027
		800				
	1600	50	.3491		.3303	.2656
		100	.4531		.4352	.3613
		200	.5516		.5375	.4595
		400	.6373		.6283	.5493
		800				
7	800	50	.3283		.3033	.2450
		100	.4367		.4070	.3371
		200	.5459		.5118	.4307
		400	.6391		.6005	.5093
		800				
	1600	50	.3629		.3407	.2754
		100	.4764		.4534	.3792
		200	.5858		.5649	.4876
		400	.6825		.6651	.5884
		800				
20	800	50	.3284		.3039	.2455
		100	.4369		.4082	.3381
		200	.5463		.5178	.4322
		400	.6397		.6035	.5113
		800				
	1600	50	.3662		.3516	.2764
		100	.4820		.4548	.3807
		200	.5944		.5672	.4898
		400	.6941		.6693	.5914
		800				

Table B.9

RESULTS for ☒ profile ☒ W, C A11c ☒ AVGTBL = 6 ☐ Name:
 Driving Function: ☐ flat ☐ H, W, C A11c ☐ AVGTBL = 12 ☒ CURVE

$R_{wall} = .013$ Blah/°F-ft² (2" ϕ PVC Schedule 40) files: 7K, 7K6

N	($C_1 + C_2$) (°F)	A (ft ²)	$C_1/C_2 = 1.00$	.80	.50	.25
1	800	50				
		100	.2410		.2270	.1891
		200	.3189		.3050	.2652
		400	.3893		.3767	.3397
		800	.4430		.4323	.4008
	1600	50				
		100	.2415		.2286	.1928
		200	.3121		.3006	.2666
		400	.3742		.3655	.3382
		800	.4226		.4170	.3980
	3200	50				
		100	.2459		.2332	.1975
		200	.3128		.3015	.2677
		400	.3696		.3608	.3338
		800	.4128		.4069	.3880
3	800	50				
		100	.2789		.2581	.2037
		200	.3876		.3630	.2930
		400	.4936		.4668	.3875
		800	.5800		.5528	.4596
	1600	50				
		100	.2951		.2759	.2178
		200	.4064		.3868	.3161
		400	.5153		.4985	.4205
		800	.6080		.5955	.5145
	3200	50				
		100	.3033		.2842	.2320
		200	.4095		.3903	.3343
		400	.5093		.4930	.4423
		800	.5923		.5809	.5407
7	800	50				
		100	.2812		.2582	.2050
		200	.3925		.3633	.2956
		400	.5026		.4674	.3877
		800	.5933		.5538	.4653
	1600	50				
		100	.3048		.2832	.2243
		200	.4247		.4010	.3296
		400	.5444		.5215	.4436
		800	.6480		.6272	.5477
	3200	50				
		100	.3247		.3011	.2401
		200	.4480		.4223	.3511
		400	.5682		.5442	.4715
		800	.6714		.6517	.5837

Table B.10

RESULTS for ☒ profile ☒ W, C Alls ☒ AVG TBL = 6 ☐ Name: CR/E
 Driving Function: ☒ flat ☐ H, W, C fills ☐ AVG TBL = 12
 minutes

$R_{wall} = .000021$ Brak./F. 10^2 ($3/4$ " Copper) files: 7L, 7LB

N	$C_s + C_w$ (10^3 / F)	A (ft ²)	$C_1/C_0 = 1.00$	.80	.50	.25
1	800	50				
		100	.2905		.2770	.2429
		200	.3696		.3500	.3174
		400	.4217		.4108	.3797
		800	.4644		.4551	.4289
	1600	50				
		100	.2852		.2736	.2482
		200	.3631		.3394	.3110
		400	.4013		.3948	.3740
		800	.4406		.4366	.4232
	3200	50				
		100	.2867		.2752	.2526
		200	.3435		.3361	.3076
		400	.3926		.3859	.3648
		800	.4280		.4236	.4097
3	800	50				
		100	.3498		.3270	.2636
		200	.4557		.4303	.3535
		400	.5483		.5220	.4340
		800	.6178		.5916	.4960
	1600	50				
		100	.3662		.3473	.2821
		200	.4736		.4566	.3829
		400	.5705		.5572	.4792
		800	.6473		.6379	.5587
	3200	50				
		100	.3707		.3514	.2992
		200	.4694		.4527	.4019
		400	.5563		.5436	.5013
		800	.6360		.6176	.5862
7	800	50				
		100	.3538		.3273	.2659
		200	.4632		.4309	.3573
		400	.5681		.5228	.4705
		800	.6333		.5928	.5025
	1600	50				
		100	.3818		.3594	.2935
		200	.4993		.4768	.4032
		400	.6068		.5860	.5096
		800	.6931		.6745	.5931
	3200	50				
		100	.4032		.3786	.3123
		200	.5214		.4973	.4271
		400	.6278		.6072	.5500
		800	.7155		.6929	.6321

Table B.11

RESULTS for ☒ profile ☐ W, C fills ☐ AVG TBL = 6 ☐ Name:
 Driving function: ☒ flat ☒ H, W, C fills ☒ AVG TBL = 12 ☐ CLR 4B
 $R_{wall} = 0.013$ Blk./F. ft² (2" ϕ PVC) file: 7P

N	$C_s + C_w$ (Blk./F)	A (ft ²)	$C_s/C_w = 1.00$	.80	.50	.25
1	800	50	.2099	.2079	.1985	.1719
		100	.2541	.2526	.2445	.2199
		200	.2983	.2976	.2921	.2704
		400	.3379	.3376	.3342	.3178
		800				
	1600	50	.2096	.2079	.1988	.1730
		100	.2572	.2502	.2428	.2191
		200	.2910	.2904	.2851	.2654
		400	.3246	.3245	.3213	.3081
		800				
3	800	50	.2655	.2674	.2579	.2091
		100	.3379	.3418	.3361	.2757
		200	.4117	.4138	.4141	.3478
		400	.4763	.4843	.4852	.4147
		800				
	1600	50	.2870	.2800	.2663	.2247
		100	.3558	.3540	.3408	.2946
		200	.4255	.4243	.4137	.3687
		400	.4849	.4847	.4782	.4395
		800				
7	800	50	.2729	.2710	.2582	.2136
		100	.3501	.3484	.3346	.2821
		200	.4275	.4267	.4142	.3563
		400	.4956	.4957	.4862	.4259
		800				
	1600	50	.3012	.3006	.2871	.2389
		100	.3851	.3853	.3728	.3173
		200	.4650	.4664	.4571	.4008
		400	.5330	.5358	.5306	.4795
		800				

Table B.12

RESULTS for ☒ profile ☒ W,C fills ☒ AVGTBL=6 ☒ Name:
 Driving Function: ☒ flat ☒ H,W,C fills ☒ AVGTBL=12 ☒ QRLB

$R_{WALL} = .000021$ Blk/°F-ft² (3/4" copper) file: 70

N	$\frac{C_1 + C_2}{C_1 C_2}$	A (ft ²)	$C_1/C_2 = 1.00$	.80	.50	.25
1	8φφ	50	.2391	.2375	.2295	.2053
		100	.2820	.2810	.2749	.2533
		200	.3230	.3225	.3188	.3021
		400	.3556	.3553	.3532	.3411
		800				
	16φφ	50	.2360	.2348	.2271	.2037
		100	.2754	.2742	.2697	.2493
		200	.3104	.3102	.3067	.2918
		400	.3390	.3390	.3368	.3280
		800				
3	8φφ	50	.3160	.3195	.3121	.2574
		100	.3881	.3936	.3892	.3268
		200	.4548	.4624	.4626	.3950
		400	.5070	.5157	.5193	.4488
		800				
	16φφ	50	.3324	.3305	.3174	.2739
		100	.4074	.4000	.3890	.3449
		200	.4624	.4626	.4548	.4154
		400	.5123	.5127	.5088	.4764
		800				
7	8φφ	50	.3271	.3281	.3124	.2638
		100	.4029	.4018	.3896	.3351
		200	.4732	.4731	.4634	.4056
		400	.5283	.5285	.5207	.4616
		800				
	16φφ	50	.3591	.3592	.3468	.2949
		100	.4382	.4393	.4297	.3750
		200	.5084	.5108	.5048	.4537
		400	.5644	.5677	.5656	.5215
		800				

APPENDIX C
MATERIAL SPECIFICATIONS

Material Selection

* KYNAR, Reg. Trade Mark — Pennwalt Corporation
 • Reg. Trade Mark — E.I. duPont de Nemours

Industrial Thermoplastics and Elastomers

PVC

(Polyvinyl Chloride) Type 1, Grade 1 PVC is the most frequently specified of all thermoplastic materials. It has been used successfully for over 30 years in such areas as chemical processing, industrial plating, chilled water distribution, deionized water lines, chemical drainage, and irrigation systems. PVC is characterized by high physical properties and resistance to corrosion and chemical attack by acids, alkalis, salt solutions and many other chemicals. It is attacked, however, by polar solvents such as ketones, some chlorinated hydrocarbons and aromatics. Of the various types and grades of PVC used in plastic piping systems, Type 1, Grade 1, PVC (Cell Classification 12454-B) conforming to ASTM D-1784 is superior with respect to the above properties. The maximum service temperature of PVC is 140°F. With a design stress of 2000 PSI, PVC has the highest long term hydrostatic strength at 73°F of any of the major thermoplastics being used for piping systems. PVC is joined by solvent cementing, threading or flanging.

CPVC

(Chlorinated Polyvinyl Chloride) Type 4 Grade 1 CPVC (Cell Classification 23477-BK) conforming to ASTM D-1784 has physical properties at 73°F similar to those of PVC, and its chemical resistance is similar to or generally better than that of PVC. CPVC, with a design stress of 2000 PSI and maximum service temperature of 210°F has, over a period of about 15 years, proven to be an excellent material for hot corrosive liquids, hot and cold water distribution and similar applications above the temperature range of PVC. CPVC is joined by solvent cementing, threading or flanging.

Polypropylene

(PP) Type 1 Polypropylene is a polyolefin which is lightweight and generally high in chemical resistance. Although Type 1 polypropylene conforming to ASTM D-2146 is slightly lower in physical properties compared to PVC, it is chemically resistant to organic solvents as well as acids and alkalis. Generally, polypropylene should not be used in contact with strong oxidizing acids, chlorinated hydrocarbons and aromatics. With a design stress of 1000 PSI at 73°F, polypropylene has gained wide acceptance in the petroleum industry where its resistance to sulfur-bearing compounds is particularly useful in salt water disposal lines, crude oil piping and low pressure gas gathering systems. Polypropylene has also proved to be an excellent material for laboratory and industrial drainage where mixtures of acids, bases and solvents are involved. Polypropylene is joined by the thermo-seal fusion process, threading or flanging.

PVDF*

(Polyvinylidene Fluoride) is a strong, tough and abrasive resistant fluorocarbon material. It resists distortion and retains most of its strength to 280°F. It is chemically resistant to most acids, bases, and organic solvents and is ideally suited for handling wet or dry chlorine, bromine and other halogens. No other solid thermoplastic piping components can approach the combination of strength, chemical resistance and working temperatures of PVDF. PVDF is joined by the thermo-seal fusion process, threading or flanging.

Viton®

(Fluorocarbon) Viton is inherently compatible with a broad spectrum of chemicals. Because of this extensive chemical compatibility which spans considerable concentration and temperature ranges, Viton has gained wide acceptance as a material of construction for butterfly valve O-rings and seats. Viton can be used in most applications involving mineral acids, salt solutions, chlorinated hydrocarbons and petroleum oils.

EPDM (EPT)

EPDM is a terpolymer elastomer made from ethylene — propylene diene monomer. EPDM has good abrasion and tear resistance and offers excellent chemical resistance to a variety of acids and alkalines. It is susceptible to attack by oils and is not recommended for applications involving petroleum oils, strong acids, or strong alkalines.

Physical Properties of Thermoplastic Piping Materials

	ASTM Test Methods	Properties	MATERIAL			
			PVC TYPE I	CPVC TYPE IV	PVDF	POLYPROPYLENE
General	D-792	Specific Gravity	1.38	1.55	1.76	.905
	D-570	Water Absorption % 24 Hrs. @ 73°F	.05	.05	.04	.02
	D-638	Tensile Strength psi @ 73°F	7,940	8,400	6,000	5,000
Mechanical	D-638	Modulus of Elasticity in Tension psi @ 73°F x 10 ³	4.2	4.2	2.1	1.7
	D-790	Flexural Strength psi	14,500	15,600	9,700	7,000
	D-256	Izod Impact Strength @ 73°F (Notched)	.65	3.0	3.8	1.3
	D-696	Coefficient of Thermal Expansion in/in°F x 10 ⁻⁵	3.0	3.8	7.9	5.0
Thermal	C-177	Thermal Conductivity BTR/HR/Sq. Ft./°F/in	1.2	.95	.79	1.2
	D-648	Heat Distortion Temp. °F @ 66 psi	NA	238	284	220
	D-648	Heat Distortion Temp. °F @ 264 psi	160	221	194	140
		Resistance to Heat °F at Continuous Drainage	140	210	280	180
	D-2863	Limiting Oxygen Index (%)	43	60	44	17
Flammability	E-84	Flame Spread	15-20	15	0	NA
		Underwriters Lab Rating (Sub. 94)	94V-O	94V-O	94V-O	94HB

Pressure Ratings

Pipe and Fittings

Temperature Correction Factors

NOMINAL PIPE SIZE	MAXIMUM OPERATING PRESSURE (PSI) AT 75°F					
	SCHEDULE 40 CPVC & PVC	SCHEDULE 80 CPVC & PVC		SCHEDULE 80 POLYPROPYLENE	SCHEDULE 80 PVDF	
	Socket End	Socket End	Threaded End	Thermo-Seal Joint**	Thermo-Seal Joint	Threaded End
1/2	600	850	420	410	580	290
3/4	480	690	340	330	470	235
1	450	630	320	310	430	215
1 1/4	370	520	260	—	—	—
1 1/2	330	471	240	230	320	160
2	280	400	200	200	275	135
2 1/2	300	425	210***	185	—	—
3	260	375	190***	185	—	—
4	220	324	160***	160	—	—
6	180	280	N.R.	N.R.	—	—
8	160	250	N.R.	—	—	—
10	140	230	N.R.	—	—	—
12	130	230	N.R.	—	—	—

*Based on water service, for more severe service, an additional correction factor may be required.

**Threaded Polypropylene not recommended for pressure service.

***For threaded and backwelded joints.

N.R. — Not Recommended.

Plastic Piping is not recommended for air or compressed gas service.

Operating Temperature (°F)	FACTORS			
	PVC	CPVC	PP	PVDF
70	1.00	1.00	1.00	1.00
80	0.90	0.96	0.97	0.95
90	0.75	0.92	0.91	0.87
100	0.62	0.85	0.85	0.80
110	0.50	0.77	0.80	0.75
115	0.45	0.74	0.77	0.71
120	0.40	0.70	0.75	0.68
125	0.35	0.66	0.71	0.66
130	0.30	0.62	0.68	0.62
140	0.22	0.55	0.65	0.58
150	N.R.	0.47	0.57	0.52
160	N.R.	0.40	0.50	0.49
170	N.R.	0.32	0.26	0.45
180	N.R.	0.25	—	0.42
200	N.R.	0.18	N.R.	0.36
210	N.R.	0.15	N.R.	0.33
240	N.R.	N.R.	N.R.	0.25
280	N.R.	N.R.	N.R.	0.18

N.R.—Not Recommended.

*Recommended for continuous drainage pressure only.

Valves

As with all other thermoplastic piping components, the maximum non-shock operating pressure is a function of temperature.

To determine the maximum operating pressure for

temperatures above 73° F, refer to the graph below. For 10" butterfly valves at elevated temperatures divide the ratings shown by two.

Flanged Systems

Maximum pressure for any flanged system is 150 psi. At elevated temperatures the pressure capability of a flanged system must be derated as follows:

MAXIMUM OPERATING PRESSURE (PSI)

Operating Temp. F.	PVC***	CPVC**	PP***	PVDF
100	150	150	150	150
110	135	145	140	150
120	110	135	130	150
130	75	125	118	150
140	50	110	105	150
150	N.R.	100	93	140
160	N.R.	90	80	133
170	N.R.	80	70	125
180	N.R.	70	50	115
190	N.R.	60	N.R.	106
200	N.R.	50	N.R.	97
210	N.R.	40	N.R.	90
240	N.R.	N.R.	N.R.	60
280	N.R.	N.R.	N.R.	25

NR = Not Recommended

**PVC and CPVC flanges sizes 2 1/2-3 and 4 inch threaded must be back welded for the above pressure capability to be applicable.

***Threaded PP flanges size 1/2 through 4 inch as well as the 6" back weld socket flange are not recommended for pressure applications (drainage only).

PROPERTY	UNIT	ASTM	TYPE I PVC	TYPE II PVC	TYPE IV CPVC
		TEST NO.	1050	4050	4125
MECHANICAL					
Specific Gravity		D-792	1.47	1.38	1.53
Tensile Strength	PSI	D-638	7,000	5,800	M7700 T6500
Elongation					
Ultimate	%	D-638	132	122	37M
Yield	%	D-638	3	4	
Modulus of Elasticity	PSI in Tension	D-638	4.2x10 ⁵	3.1x10 ⁵	
Flexural Strength	PSI	D-790	12,500	10,800	M14,600 T13,750
Flexural Modulus	PSI x 10 ⁵	D-790	4.5	3.7	M4.74 T4.50
Impact Strength (Notched Izod)	Ft. Lbs. / in. Notch	D-256	0.8	15	1.65
Hardness, Rockwell	"R"	D-785	115	111	118
Hardness, Shore	"D"	D-2240	82	80	82
Compression Strength	PSI	D-695	10,830	8,020	11,400
Shear Strength	PSI	D-732	9,240	6,510	9,220
THERMAL					
Thermal Expansion	In/In/°C	D-696	2.95x10 ⁻⁵	3.72x10 ⁻⁵	7.97x10 ⁻⁵
Heat Distortion	°F @ 264 psi	D-648	168	160	212
Thermal Conductivity	BTU / HR / FT ² / °F / In.	C-177	0.72	0.74	0.641
Specific Heat	CAL / GM / °C	C-351	0.20	0.21	0.220
ELECTRICAL					
Dielectric Strength	Volts / Mil.	D-149	552	335	
Dielectric Constant	60 Hertz	D-150	2.91	2.85	
Dissipation Factor	60 Hertz	D-150	.018	.016	
Loss Index	60 Hertz	D-150	.051	.045	
Volume Resistivity	(Ohms/cm) x 10 ¹⁴	D-257	1.48	1.80	
OTHER					
Water Absorption 24 hrs.	%	D-570	.032	.056	.035
Flammability	1 / 8" and over	*	94V-0	94V-0	94V-0
Thermoformability			Poor	Fair	Fair
Weld gas recommended			Air	Air	Nitrogen

NA — Not Applicable

* Flammability — Ratings for PVC in accordance with UL94V
Ratings for Olefins in accordance with ASTM D-635

Physical Properties - - Typical Values

PVC			POLYETHYLENES		POLY- PROPYLENE	PROPERTY
Clear 4300 Type II	Extruded 1065 Type I, Grade 2	Extruded 4065 Type II, PVC	Low Density 5000	High Density 5200	5500	
						MECHANICAL
1.29	1.39	1.34	0.923	0.952	0.905	Specific Gravity
5,600	7,800	6,040	1,350	3,020	4,080	Tensile Strength
	90	140	363	1,250	585	Elongation
	4	4	15	16	19	Ultimate
3.0×10^5	4.2×10^5		$.34 \times 10^5$	$.17 \times 10^6$	$.18 \times 10^6$	Yield
	13,190	9,540	NA	NA	6,420	Modulus of Elasticity
	4.7	3.37	0.24	1.11	1.90	Flexural Strength
10 — 15	1.0	15	NA	3 - 7	.5 - 1.0	Flexural Modulus
	118	110	NA	67	102	Impact Strength (Notched Izod)
72	86	82	58	72	76	Hardness, Rockwell
7,400	10,323	7,430	4,030	4,570	6,720	Hardness, Shore
6,060	8,180	5,740	2,440	3,380	5,710	Compression Strength
						Shear Strength
						THERMAL
7.37×10^3	6.24×10^{-5}	7.51×10^{-5}	1.61×10^{-4}	1.25×10^{-4}	9.58×10^{-5}	Thermal Expansion
150	158	151	NA	151	210 * *	Heat Distortion
0.73	0.80	0.86	1.50	1.53	0.83	Thermal Conductivity
0.29	0.273	0.295	0.67	0.52	0.46	Specific Heat
						ELECTRICAL
350	363	293	488	486	503	Dielectric Strength
3.93	3.72	4.01	2.63	2.65	2.65	Dielectric Constant
0.005	0.010	0.008	0.006	0.006	0.005	Dissipation Factor
0.020	0.037	0.032	0.016	0.016	0.013	Loss Index
5.26	14.1	5.26	8.48	8.37	8.45	Volume Resistivity
						OTHER
0.15	0.15	0.15	< 0.04	< 0.03	< 0.03	Water Absorption 24 hrs.
94V-0	94V-0	94V-0	1 in. / min	1 in. / min.	1 in. / min.	Flammability
Fair	Poor	Good	Good	Good	Good	Thermoformability
Air	Air	Air	Air	Air	Nitrogen	Weld gas recommended

* * Heat Distortion Rating is at 66 psi. Rating drops to about 140 degrees F at 264 psi.

Robintech PVC Pressure Pipe

- Pressure Rated (SDR)
- Schedule 40
- Schedule 80

Robintech manufactures PVC Pressure Pipe to meet the following standards:

ASTM D 1784 PVC Compounds

ASTM D 1785 PVC Plastic Pipe, Schedules 40 & 80

ASTM D 2241 PVC Plastic Pipe, (SDR)

ASTM D 2672 Bell-End PVC Pipe

NSF Standard No. 14

SCS 432-D High Pressure Underground Plastic Irrigation Pipelines

Key Definitions:

O.D. — Average Outside Diameter

I.D. — Average Inside Diameter (The I.D. dimensions shown are the result of using the minimum wall thickness plus half of the wall tolerance.)

WA. — Minimum Wall Thickness

WT. — Weight Per Foot

PSI — Rated Working Pressure for water at 73.4° F.

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TWX (910) 839-5074 Pho. (817) 336-7323

DIMENSIONAL SPECIFICATIONS

SIZE IN INCHES	O.D. IN INCHES	KEY	100 PSI SDR 41	125 PSI SDR 32.5	168 PSI SDR 26	200 PSI SDR 21	250 PSI SDR 17	315 PSI SDR 13.5	SCHED 40	SCHED 80
1/2	.840	I.D.		.730				.696	.602	.528
		WA.		.045				.062	.109	.147
		WT.		.082				.106	.164	.205
		PSI							.600	.850
3/4	1.050	I.D.		.934		.910	.906	.874	.804	.722
		WA.		.048		.060	.062	.078	.113	.154
		WT.		.102		.131	.135	.162	.218	.278
		PSI							.480	.690
1	1.315	I.D.		1.191	1.175	1.169	1.141	1.101	1.029	.936
		WA.		.052	.060	.063	.077	.097	.133	.179
		WT.		.149	.167	.174	.208	.248	.321	.409
		PSI							.450	.630
1 1/4	1.680	I.D.		1.528	1.512	1.482	1.444	1.394	1.360	1.258
		WA.		.056	.064	.079	.098	.123	.140	.191
		WT.		.202	.225	.258	.321	.389	.434	.567
		PSI							.370	.520
1 1/2	1.900	I.D.		1.764	1.734	1.703	1.656	1.598	1.590	1.476
		WA.		.058	.073	.090	.112	.141	.145	.200
		WT.		.239	.289	.346	.416	.506	.518	.686
		PSI							.330	.470
2	2.375	I.D.		2.209	2.173	2.129	2.075	2.002	2.047	1.913
		WA.		.073	.091	.113	.140	.176	.154	.218
		WT.		.364	.440	.530	.639	.783	.695	.949
		PSI							.280	.400
2 1/2	2.875	I.D.		2.679	2.635	2.581	2.517	2.423	2.446	2.290
		WA.		.088	.110	.137	.169	.213	.203	.276
		WT.		.521	.633	.768	.924	1.147	1.096	1.448
		PSI							.300	.420
3	3.500	I.D.		3.310	3.264	3.210	3.146	3.063	2.951	3.042
		WA.		.085	.108	.135	.167	.206	.259	.216
		WT.		.620	.764	.932	1.127	1.376	1.699	1.435
		PSI							.260	.370
3 1/2	4.000	I.D.		3.784	3.734	3.672	3.597	3.502	3.372	3.521
		WA.		.098	.123	.154	.190	.235	.286	.226
		WT.		.805	.985	1.205	1.470	1.789	2.217	1.729
		PSI							.240	.350
4	4.500	I.D.		4.260	4.204	4.133	4.046	3.938	3.794	3.998
		WA.		.110	.138	.173	.214	.265	.333	.237
		WT.		1.007	1.234	1.521	1.858	2.271	2.804	2.043
		PSI							.220	.320
5	5.563	I.D.		5.271	5.200	5.109	5.001	4.870	4.690	5.016
		WA.		.136	.171	.214	.265	.327	.412	.258
		WT.		1.515	1.876	2.330	2.843	3.467	4.291	2.776
		PSI							.190	.290
6	6.625	I.D.		6.282	6.193	6.084	5.955	5.798	5.584	6.031
		WA.		.162	.204	.255	.316	.390	.491	.280
		WT.		2.126	2.652	3.299	4.036	4.926	6.092	3.600
		PSI							.180	.280
8	8.625	I.D.		8.180	8.063	7.921	7.754	7.548		7.942
		WA.		.210	.265	.332	.411	.508		.322
		WT.		3.589	4.491	5.578	6.825	8.349		5.427
		PSI								.160
10	10.750	I.D.		10.195	10.048	9.874	9.665	9.410		9.976
		WA.		.262	.331	.413	.512	.632		.365
		WT.		5.577	6.992	8.652	10.599	12.937		7.683
		PSI								.140
12	12.750	I.D.		12.091	11.919	11.711	11.463	11.160		11.889
		WA.		.311	.392	.490	.607	.750		.406
		WT.		7.851	9.829	12.183	14.913	18.206		10.171
		PSI								.130

*There is no NSF Standard for sizes 1/2" through 2 1/2" SDR 32.5, or 3" SDR 41.

WELD-ON

A GENERAL BULLETIN COVERING PLASTIC PIPE CEMENTS AND ACCESSORIES

NOTE: The information provided here is of a general nature and is meant to be used descriptively only. Users should obtain explicit detailed literature and instructions for any of our products before using them.

PVC SOLVENT CEMENTS

For use with PVC pipe class 12454 — B & C, 14333-D, per ASTM D 1784, D 1785, D 2241, D 2466, D 2467, D 2513, D 2665, D 2672, D 2729, D 2740, D 2855, D 3033, D 3034, F 512, CS 272, IAPMO IS 8, IS 9, IS 10, PS 21, PS 22, PS 27, UL 651, FHA UM 41, FHA UM 53, FHA 563, L-P 320a PVC DWV.

WELD-ON NUMBER	GENERAL DESCRIPTION	COMMENTS
705	A clear or gray thixotropic (slow flowing) medium bodied, fast curing, very high strength cement. For Types I & II PVC pipe in sizes through 6" interference fits only. For all schedules and classes except Schedule 80. IAPMO-UPC and NSF listed. CSA certified for DWV. Meets ASTM D 2564. (See P-70 Primer.)	For potable water pressure pipe, gas, conduit and (D.W.V.) drain, waste and vent. Flows more rapidly than 711, 717 but not as free flowing as 710. It has a longer open time than 710 and better gap-filling properties. Application temperature 40°F to 110°F.
710	A clear — light bodied — very fast set high strength cement. For types I & II PVC in sizes 1/8 to 2" interference fits only. For all pipe schedules and classes, except Schedule 80. IAPMO-UPC and NSF listed. Meets ASTM D 2564. (See P-70 Primer.)	For potable water pressure pipe, gas, conduit, drain pipe. In field use since 1958. Flows freely. Makes a thin layer. Very fast cure. Dries quickly. Application temperature 40°F to 110°F.
711	A gray, heavy bodied — fast set high strength cement. For types I & II PVC in sizes through 12". For all pipe schedules and classes, including schedule 80. Especially formulated for large sizes and heavy schedules. May also be used for smaller sizes. IAPMO-UPC and NSF listed. Meets ASTM D 2564. (See P-70 Primer.)	For potable water pressure pipe, gas, conduit, drain pipe, and (D.W.V.) drain, waste and vent. Provides a thicker layer of cement on the pipe than 705. Helps to fill gaps in the larger sizes and looser fits. Allows a longer time for assembly. Application temperature 40°F to 110°F.
717	Available in Gray — is a heavy bodied — medium set cement. For types I & II PVC in sizes through 12". For all pipe schedules and classes including schedule 80. Formulated to have a slower set than 711 to allow more time in assembly of large sizes. IAPMO-UPC and NSF listed. Meets ASTM D 2564. (See P-70 Primer.)	For potable water pressure pipe, gas, conduit, drain pipe, (D.W.V.) drain, waste and vent. Provides a thicker layer of cement on the pipe than 705. Helps to fill gaps in larger sizes and looser fits with its thickness. Allows a longer time for assembly. Application temperatures 40°F to 110°F.
719	Available in Gray & White — is an extra heavy bodied, highly thixotropic medium set cement. For Types I & II PVC in sizes 4" through 15". For all pipe schedules and classes including Sch. 80. Especially formulated for large sizes and heavy schedules that require very high gap filling properties. NSF listed. Meets ASTM D 2564.	For potable water pressure pipe, gas, conduit, drain pipe, sewer pipe and (D.W.V.) drain, waste and vent. Can be used in fabrication of large size PVC fittings. Ideal for saddles. Provides thicker layer and higher gap filling properties than 711 and 717. Allows slightly longer working time than 711. Application temperatures 40°F to 110°F.
795	For joining flexible PVC to itself or to rigid PVC. It is not recommended for rigid PVC to rigid PVC connections. It is a clear-medium bodied fast set cement. For use on low pressure systems. (See P-70 Primer.)	This cement has been used in many fields. Its prime use has been in the irrigation field in sprinkler systems. Because of the wide variation of hose materials, suitability of 795 for specific materials and applications should be verified by users. (Hose pull systems.) Provides a medium layer, with medium gap filling property. Forms a strong flexible joint. Application temperatures 40°F to 110°F.

CPVC SOLVENT CEMENTS

For use with CPVC pipe class 23447-B as per ASTM D 1784, F 438, F 439, F 441, F 442, F 443 and D 2846.

713	An orange — light bodied — fast set cement. For CPVC in sizes through 2" interference fits only. Copper tube size and pipe schedules, except Schedule 80. IAPMO-UPC and NSF listed. Meets ASTM F 493 and D 2846.	For potable water pressure pipe and industrial systems, cold or hot water (180°F maximum). Flows freely, moderate gap filling properties, fast cure. Application temperatures 40°F to 110°F. Store below 90°F.
714	An orange — heavy bodied — medium set cement. For sizes through 6". For all pipe schedules. NSF listed. Meets ASTM F 493 and D 2846.	For potable water pressure pipe and industrial systems, cold or hot water (180°F maximum). Provides a thicker layer on the pipe than 713. Fills the gaps in larger sizes and looser fits. Allows more time in assembly than 713. Application temperatures 40°F to 110°F. Store below 90°F.

ABS SOLVENT CEMENT

For use with ABS pipe Type I grades 1, 2, 3, Type II grade 1, Type IV grade 1 (Class 3-2-2, 5-2-2, 3-5-5, 4-4-5, 1-3-3) per ASTM D 1527, D 1788, D 2282, D 2489, D 2661, D 2750, D 2751; PS 18 (CS 218-59) PS 19 (CS 254-63) CS 270-65; FHA UM 43, FHA UM 49, FED SPEC. L-P 322a, IAPMO (WPOA TSC 6-61), PS 17-66, IS 5, IS 11.

771	A milky clear — medium bodied — fast set cement. For Types I & II ABS in sizes through 6". For all pipe schedules. IAPMO-UPC and NSF listed. Meets ASTM D 2235.	For drain pipe, gas, conduit and (D.W.V.) drain waste and vent. Chiefly used where the black color of 773 is undesirable. Fast cure. Makes a medium layer when applied. Application temperatures 40°F to 110°F.
773	A black — medium bodied — fast set cement. For Types I & II ABS in sizes through 6". For all pipe schedules. IAPMO-UPC and NSF listed. CSA certified for DWV. Meets ASTM D 2235.	For drain pipe, gas, conduit and (D.W.V.) drain, waste and vent. For pressure and non-pressure applications. Fast curing, makes a medium layer when applied. Chiefly used in D.W.V. installations. Application temperatures 40°F to 110°F.
793 ABS to PVC	This cement was developed for cementing ABS parts to PVC parts. It is not recommended for ABS to ABS or PVC to PVC. It is a clear — light bodied — very fast set cement. For interference fits in sizes through 2 1/2". For all pipe schedules and classes except Schedule 80.	For potable water, drain pipe conduit and (D.W.V.) drain waste and vent. It is a high strength cement. Makes a thin layer when applied that is fast setting. (Because of the wide variety of ABS and PVC materials being used, suitability of the cement should be verified for specific materials.) Application temperatures 40°F to 110°F.
794 ABS to PVC	A green colored, medium bodied, fast setting high-strength solvent cement for joining ABS to PVC in non-pressure transition joints only. It is formulated for interference type fits up to 6" in all schedules and classes except Schedule 80. IAPMO-UPC and NSF listed. Meets ASTM D 3138.	This cement has bonded ABS to PVC satisfactorily for many years. But because of wide variations in pipe and fitting materials, users should verify the suitability of 794 for specific materials and applications. Not recommended for joining PVC to PVC or ABS to ABS. Application temperatures 40°F to 110°F.

STYRENE RUBBER SOLVENT CEMENT

For use with Styrene rubber plastic pipe and fittings per CS 228-81 and Federal Spec. LP-001221. (GSA-FSS) ASTM D 2852.

WELD-ON NUMBER	GENERAL DESCRIPTION	COMMENTS
763	A milky — medium bodied — fast set cement for Styrene. Do not use this cement on other types of plastics. Application temperatures 40°F to 110°F. Meets ASTM D-3122.	For drain pipe, conduit, sewer and vacuum piping systems. It is a fast curing cement that makes a medium layer with medium gap filling properties.

PRIMERS

P-70 PVC	is a Primer. It is purple colored and very fast setting. Recommended for use with all PVC cements. For all sizes, types and schedules. Developed to soften and prepare plastic pipe and fittings for bonding. IAPMO-UPC and NSF listed.	The function of this primer is to penetrate and soften the bonding surfaces of pipe and fittings. Its high strength penetrates rapidly on the hard finish, high gloss products that many manufacturers are now producing. (Do not use as a thinner.) Application temperature 40°F to 110°F.
P-72 CPVC	A clear, fast acting primer, recommended for use with CPVC Type IV grade 1 (Class 23447-B). IAPMO-UPC and NSF listed.	P-72 is developed especially to soften and prepare CPVC pipe and fitting surfaces for bonding.

GASKET LUBRICANT

787	A non-flammable, non-toxic vegetable soap. Paste-like. Easy to apply. Ask for specific information.	For use in assembling gasketed plastic pipe and fittings. Suitable for use in potable water systems. Does not support bacterial growth. Has no adverse effect on most rubber gaskets. Application temperatures 10°F to 120°F.

TWO-COMPONENT REACTIVE ADHESIVES

10	A two-part, white, fast curing, high gap filling, very high strength reactive adhesive. For joining rigid PVC pipe and fittings. May also be used for fabrication of fittings and for joining CPVC, ABS, styrene, concrete, clay pipe, etc. Ask for detailed information.	A unique characteristic of this adhesive is that both parts need not be mixed before application. The catalyst may be applied to one surface and resin to the other. Can also be used by mixing both parts. Pot life: 30-40 minutes. Provides high peel strength, impact resistance and water and weather resistant joints.
1950	A two-part, thixotropic, very high gap filling, high strength reactive adhesive. For joining rigid PVC pipe to other plastics and non-plastic piping materials. Ask for detailed information.	May also be used to bond PVC or ABS to concrete or clay pipe. Good for plastic to non-plastic transition joints. Pot life: 30-40 minutes approximately. Produces joints that are water and weather resistant.

QUALITY ASSURANCE

Every batch of our cement is checked to assure that consistent quality is maintained. An infra red absorption curve is recorded for each batch to ensure that our cement is properly formulated. Samples are taken from each batch and kept for a period of at least one year. A batch identification code is stamped on each can. For any questions regarding the contents of any container, please refer to this identification code.

EDUCATIONAL PROGRAMS/FIELD TRAINING

As part of our customer service, we provide a free educational/training program on solvent cementing plastic pipe. This audio visual program has been used by numerous groups such as installers, contractors, inspectors, government agencies, trade schools, etc. We strongly recommend that all who are concerned with the design and installation of plastic piping systems should see this program. Other printed literature on solvent cementing plastic pipe and fittings is also available.

IMPORTANT NOTE

Our Weld-On cements will consistently make strong, leak free joints if properly used with correct materials. Installers should verify for themselves that they can make satisfactory joints under varying conditions. Toward this end, it is highly desirable that they receive personal instruction from trained instructors or installers. Contact us or your supplier for additional information and assistance.

APPENDIX D
TIME-USE STUDY DATA

TIME-USE SURVEY BACKGROUND

A survey was undertaken of local area laundromats to determine the best laundromats to do a time-use study, survey the equipment and methods in use, and locate a possible test site for actual testing of the appliance. A total of 12 laundromats were studied listed in Table B.1. The information available was dependent on the accessibility of the owner, the extent to which they had the information and/or agreed to be interviewed. Laundromat owners were receptive to the idea of a waste water heat recovery system to save energy but none of them offered their laundromat to test the waste water heat recovery appliance. This necessitated using simulations to determine the performance of the appliance and steady state testing for verification of heat transfer coefficients.

TIME-USE STUDY

Time-use studies were made of two laundries. This was done over a couple of days. A sample time use data record is shown in Figure D.1. From this the following general observations were made.

There appeared to be some patterns in the number of customers over a given time period. Observations from one owner are: the busiest day is Sunday, followed by Saturday. Thursday is the least busy. Customers arrive in peaks throughout the day, e.g. middle of the mornings and afternoons. Customers generally start-up two, three

Table D.1 Laundromat survey

1. EAST-SIDE, 501 RIVERSIDE ,Owner Carol Butler, 493-9691. This location was not a "typical" one in that most of the work was by contract or dry cleaning. The hot water temperature setting was 190 degrees (F).
2. A AND B, 127 W. MULBERRY
The owner of this laundry tried a dryer heat exchanger but removed them due to lint problems. The best heat exchange (temperature rise of incoming cold water to boiler) had been eight degrees(F), usually less. There were 21 Speedqueens (\$.60), 8 Tumblette dryers, 4 Speedqueen dryers, 2 Philco commercial washers(\$1), and 2 Speedqueen commercial washers (\$1). The machines used 37 gallons per load, 1 wash and 2 cold rinses.
3. COINOMAT, 415 S. MASON, owner; Don and George Barrett, 482-3007,4342. Open 6am-10pm. Equipment includes: 4-Zamussi 35 pounders (\$1.25/load),24 Wascomat Juniors,w-73, (\$.60), 5-Speedqueens (\$.60), 3 older Wascomat juniors (\$.60), 1-triple loader Wascomat senior (\$1.00), 7-Heubsch dryers, 4-Cissell dryers(\$.10/6min) variable temperature. Fuel is 25% of the total cost. No room for appliance. Does not want the mess.
4. FABRIC-CARE CENTER, 105 E. MYRTLE, owner Clarence and Martha Jones. 5-commercial Speedqueens, 11-Maytags, 1 large Aldwash (extractor). Horizontal baseboard fin tubes were run over the dryers which preheated the incoming hot water.
5. X-L LAUNDROMAT, 1700 LAPORTE,owner Thomas Jenoway 482-5865. unattended with 12-w73 Wascomats juniors, 5-white Westinghouse commercial washers, and 2-heavy duty Westinghouses. 1-Bock centrifugal extractor (\$.10), 5-Heubsch Loadstar dryers(\$.10), 3-dryers(\$.10). Incoming cold water, 40 degrees(f). Had two preheaters for water. Solar air system with air-liquid exchanger, and waste heat recovery on dryers. Both preheaters hooked in parallel. Solar collectors (air) installed by Nature Power of Laporte. Three banks of collectors 24 feet by 30 inch. Owner thinks gas consumption dropped from 1800-1900 cubic feet per month to about 1400 to 1500. Payback of about five years. Interested in waste heat recovery. Thinking about electric pilots. Trough and pit type drain for waste water.

Table D.1 Laundromat survey (cont.)

6. ECON-O-WASH, 503 S SHIELDS, owner Mr. Wedum. southland corp no. 13171 32-Speedqueens (\$.50). 8-Heubsch dryers, 1-Wascomat senior. Owner not interested.
7. CAMPUS WEST, 1126 W ELIZABETH, owner Clarence and Martha Jones, 484-8335. 20-Maytags, cold water storage tank, 1-Milnor washer (44gal), 25min cycle time, 35lb triple washer, 10-dryers, 8-90Kbtu, 1-78Kbtu. He thought that the wash and rinse need to be diverted separately. Gas consumption included washers, dryers, room heating, and drycleaning machines. Uses about 115000 gallons of water per month. The waste water discharge was at the opposite end of the laundry (relative to the boiler).
8. PROSPECT PLAZA, 304 W PROSPECT, 52-Maytag commercial washers, unattended.
9. PROSPECTOR LAUNDROMAT, PROSPECTOR SHOPPING CENTER, 20-Maytags, no attendant, no customers at time of survey.
10. BW COINOP, 1733 S COLLEGE, Owners Betty and Warren White, 482-0925 (7pm or early morning). 16-Whirlpool commercial washers, model CDAL75010, "no real patterns, no business around dinner, busy different times on saturday", empty at 4pm during survey. Hot water set at 130 degrees (f), air conditioning.
11. STARK LAUNDROMAT, 138 N HOWES 30-Maytags (commercial), 2-Aedspin extractors, 18-commercial Maytag dryers.
12. VILLAGE WEST, W. PLUM, Colorado State University 21-Maytags (HA17CM), 35 cent wash, 25cent 15min dry. 25min cycle time on washers, maintained by poudre valley coin-op laundry, 493-4130.

Figure D.1 Time-use study data sample.

machines, and as many as 4-5 were noted. Most temperature settings were hot (120° F) or warm (80° F) about equally divided. Sometimes more than one person would arrive and start up as many as 8 machines at one time.

Conclusions for the computer model are: use random number generator with Poisson distribution for customer inflow (peaking at mid morning and mid afternoon), vary the number of machines started and temperature settings on the machines.

RESEARCH

The laundry business is very sensitive to fuel prices. Typical laundries pay 25 to 40 percent of the gross revenues in energy costs or utilities. Research first concentrated on the general market. The laundry trade magazines have many advertisements for energy saving products. An understanding of these products is important to "see the whole picture" of energy consumption in a laundry and how the waste heat recovery appliances would be applied. The most abundant energy conservation devices seem to center around dryers. Dryers consume typically 75,000 to 100,000 btus per hour in operation. The most typical device is a waste heat recovery device that fits on the vent hoods for the dryers. These are usually set-up to preheat the incoming water to the boiler. The biggest problem noted was lint build-up (causing constant maintenance). Automatic pilots are used, since the gas for pilots was costing one

owner 3 to 5 dollars per month. Set temperatures have been lowered on boilers (usually down to 120° to 130° F).

Boilers are usually in a typical loop configuration. The boiler heats a tank (about 280 gallons) to the thermostat temperature (120-130° F). The city water inlet temperature in the survey area is about 40 to 50° F. The warm cycle on the machines typically runs about 80° - 90° F mixed at the washing machines.

Most laundries have the waste for the washing machines empty at floor level. Usually machines are placed back-to-back in rows with the utilities run between them as shown in Figure D.2. A channel is provided in the center which accepts the outgoing waste from the machines. This presents a potential problem for waste heat recovery since the present design option for the appliance relies on gravity flow. This would necessitate placing the exchanger below the waste water level (usually ground level).

Possible solutions to this problem are to put the appliance at the waste discharge level and pump the waste water to drain level if required, or to raise the discharge level of the washing machine output to above machine level, e.g. ceiling. The last option depends on the water head of the pump in the washers. Some locations had the waste drain and boiler at opposite ends of the building. This last problem is easy to fix in the design phase but would have to be remedied in retrofit cases or tolerate a lot of piping.

1-2 SWISSWASH
 3, 5, 7, 9, 10
 12, 13, 14, 15, 17, 19, 20 } WASCOMAT JR.
 25, 26, 27
 4, 6, 8, 11, 16, 18 SPEED QUEEN
 21-24 ZANUSSI
 28-31 —

Figure D.2 Schematic of typical laundromat.

Machine cycles varied among the different machines. An example is shown in the data in Appendix E. This is for a Maytag commercial washer, a typical machine in use.

APPENDIX E
WASHER SPECIFICATIONS

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