MASTER

PHASE I

FINAL TECHNICAL REPORT

Feasibility Study of Residential and Commercial Heating Using Existing Water Supply Systems

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DISCLAIMER

SIX-MONTH SUMMARY REPORT

PHASE I

FEASIBILITY STUDY OF RESIDENTIAL AND COMMERCIAL DISTRICT HEATING USING EXISTING WATER SUPPLY SYSTEMS

ABSTRACT

A preliminary study of the feasibility of using existing drinking water supply systems to provide hot water for heating purposes to a typical 140-home subdivision has been undertaken. This preliminary study has centered on (i) types of municipal water system designs and effects of system design upon heating use; (ii) methods of using low-to-moderate temperature water for heating purposes and possible institutional barriers; (iii) identification and description of a typical residential community suitable for hot water heating; (iv) evaluation of thermal losses in the uninsulated main distribution system from the main pumping station having assumed geothermal heating to the subdivision; (v) evaluation of thermal losses in the uninsulated street mains in the subdivision; (vi) estimation of size and cost of the pumping station main heat exchanger to supply geothermal energy to the drinking water; (vii) sizing of individual house heat exchangers; (viii) pumping and power requirements to supply the increased water flow rate through the subdivision street water distribution lines; and (ix) pumping and piping requirements to provide heating water flow from the street lines to a typical residence. From the results obtained thus far, it would appear that the use of existing uninsulated water supply systems will be advantageous in many communities having nearby low-to-moderate temperature geothermal resources.

INTRODUCTION

District heating with geothermal energy has been very slow in developing in the United States. A major factor hindering its development is the high cost of providing hot water distribution throughout a community, city or subdivision. A unique approach to distribution of the hot water within a prospective using community is to utilize the existing potable water supply distribution system. In simplest form, such a scheme involves:

1. Supply of geothermal fluid via an insulated pipeline (new) to a central heat exchanger at a potable water pumping main station or substation.
2. Supply of thermal energy from the geothermal fluid to the potable water supply via a central heat exchanger with the potable water at a higher pressure than the geothermal fluid.
3. Distribution of heated potable water throughout the using community through the existing distribution mains with flow as needed to the individual thermal user's heat exchanger and return, via supplemental pumping, to the street distribution line.
4. Workable systems only in communities with significant water useage rates to maintain a sufficiently high water temperature in the main distribution lines -- a large commercial, industrial or agricultural use near the end of the existing water supply piping system would be ideal.

The Phase I investigation was intended to provide a preliminary examination of a number of factors of importance to the proposed district or community heating scheme using the existing water supply including

- energy loss rate from uninsulated piping
- additional pumping energy
- approximate cost of the main heat exchanger at the water treatment/pumping station, and
- approximate cost of installed equipment at using points.

Specific conditions to be considered included a range of temperatures from 90°F - 140°F with heat pumps to be used at the lower temperatures for both heating and air conditioning. Basic questions to be answered in the Phase I work were:

- Are typical community water systems so designed as to be able to be used when filled with a heated water stream, and are these capable of recirculation with reasonable modifications to minimize the ΔT througout the system?
- What would be needed at domestic or commercial sites to appropriately extract heat?
- Generically, what are the institutional barriers?
- What are the generalized economics of a distribution and heating system of this type?

The study thus far has produced some of the answers to these questions; it has included all factors and questions originally intended to be studied, but some of these have not been pursued to the extent that may be desired. In particular, institutional barriers have been treated in a cursory way, and the generalized economic study is not yet complete. Nonetheless, all results of the study to date are very encouraging. These results are discussed in the following sections.

I. TYPICAL WATER SYSTEMS

City water system layouts are of two general types -- a single main line design or a dual loop design. These are depicted in Figure 1. As a general rule a city having a population of less than 30,000 will have a single main system; a city with a population of more than 30,000 will have a dual-loop type main system.

In general, the dual loop system is more readily adaptable to district heating because of the occurence of communities (subdivisions) or commercial centers along one of the cross-connecting sub-mains. This allows supply from one side of the loop with downstream flow through the opposite side. Often a subdivision will have a complete internal loop attached to one of the cross-connecting sub-mains.

The single main system will have fewer complete loops. In general, it will necessitate addition of a downstream line, such as the dotted one in Figure (1.a). Either system, however, can usually be adapted for district heating due to the common occurance of public rights-of-way, alleys, etc., with minimal or no land acquisition expense.

The materials commonly used in older public water systems are cast iron and concrete. Either of these is insensitive to the modest temperatures of interest in this study. Some of the newer systems utilize polyvinyl-chloride piping which is structurally sensitive to temperature and probably not suitable for use with water above 100°F, but a large percentage of the systems installed in western communities during the past 10-15 years use abestos-cement pipe which should present no problems.

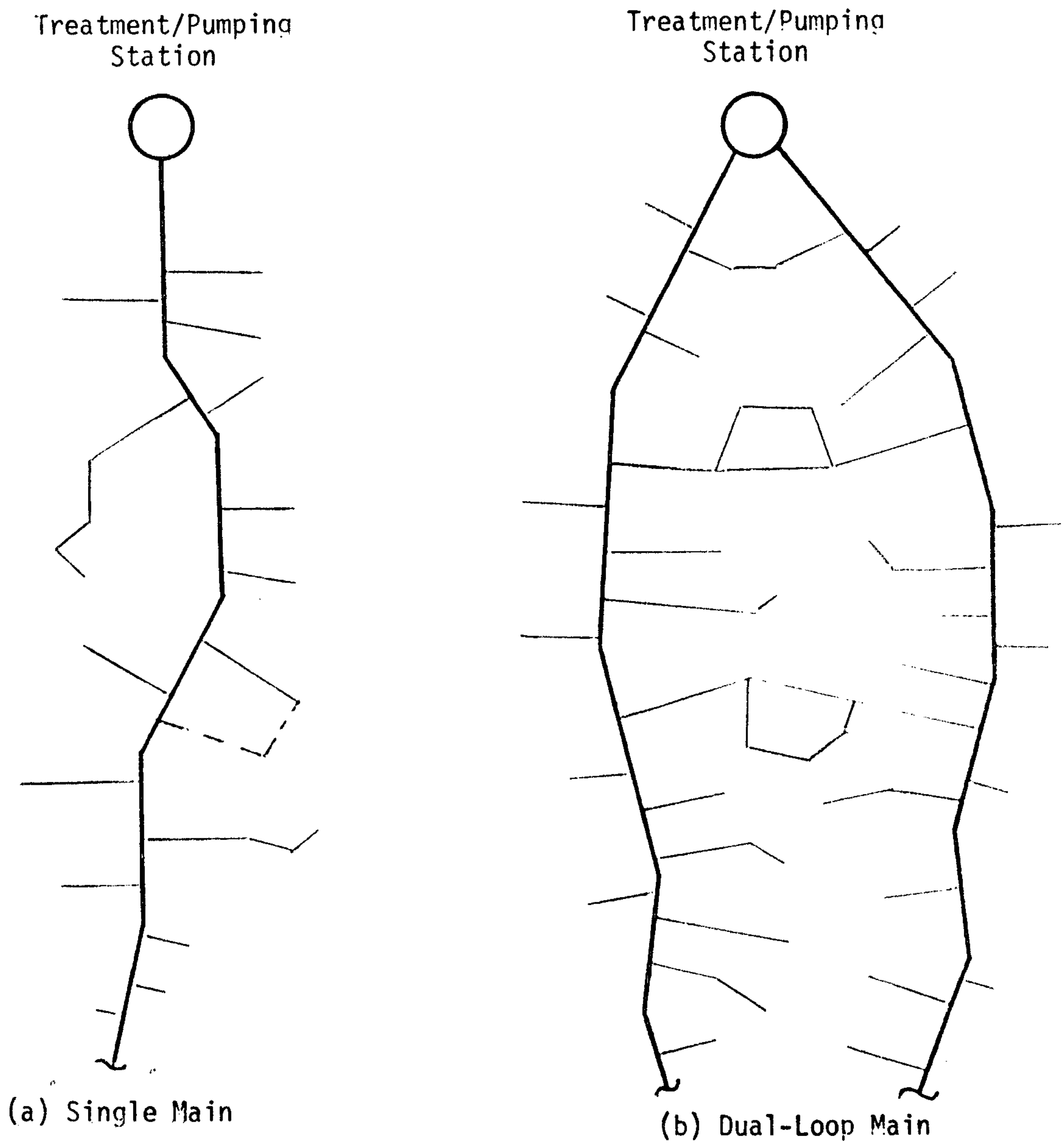


Figure 1. Typical City Water Distribution Main Line Designs.

II. METHODS OF USE

The hot water resource in the temperature range of this study is suitable for either (i) space heating by direct heat exchange with simple water-to-air heat exchangers or (ii) space heating by means of heat pumps; the choice is primarily a function of the temperature of the fluid at the using point. The relatively low temperature effectively precludes air conditioning applications, since the lower limit of practical generator temperatures for any of the absorption cycle refrigeration/air conditioning devices currently available is considerably above 150°F.

Direct Heat Exchange Space Heating

From a practical viewpoint, forced hot air heating systems should have a minimum lower temperature limit of about 100°F, and this dictates an approximate lower limit of about 115°F on the inlet fluid temperature for a water-to-air heat exchanger for direct space heating. Consequently, in view of the 140-150°F resource maximum temperature, we have considered water side temperature drops of 15°F, 25°F and 35°F only for direct space heating. These temperature changes control the mass flow rate required for a given residential thermal load, and this in turn is a major variable in determination of the pumping power and temperature degradation in the distribution lines. Most of the pressure loss, pump power and thermal loss parametric studies of this report were undertaken for direct heat exchange space heating.

Heat Pump Space Heating

An efficient water source system heat pump (current off-the-shelf model) would provide an economical approach to using warm water in the 50°F to 100°F temperature range. An approximate plot of the coefficient of performance of a unit of this type is given as Figure 2; the approximate nature arises due to extrapolation to higher source temperatures than furnished

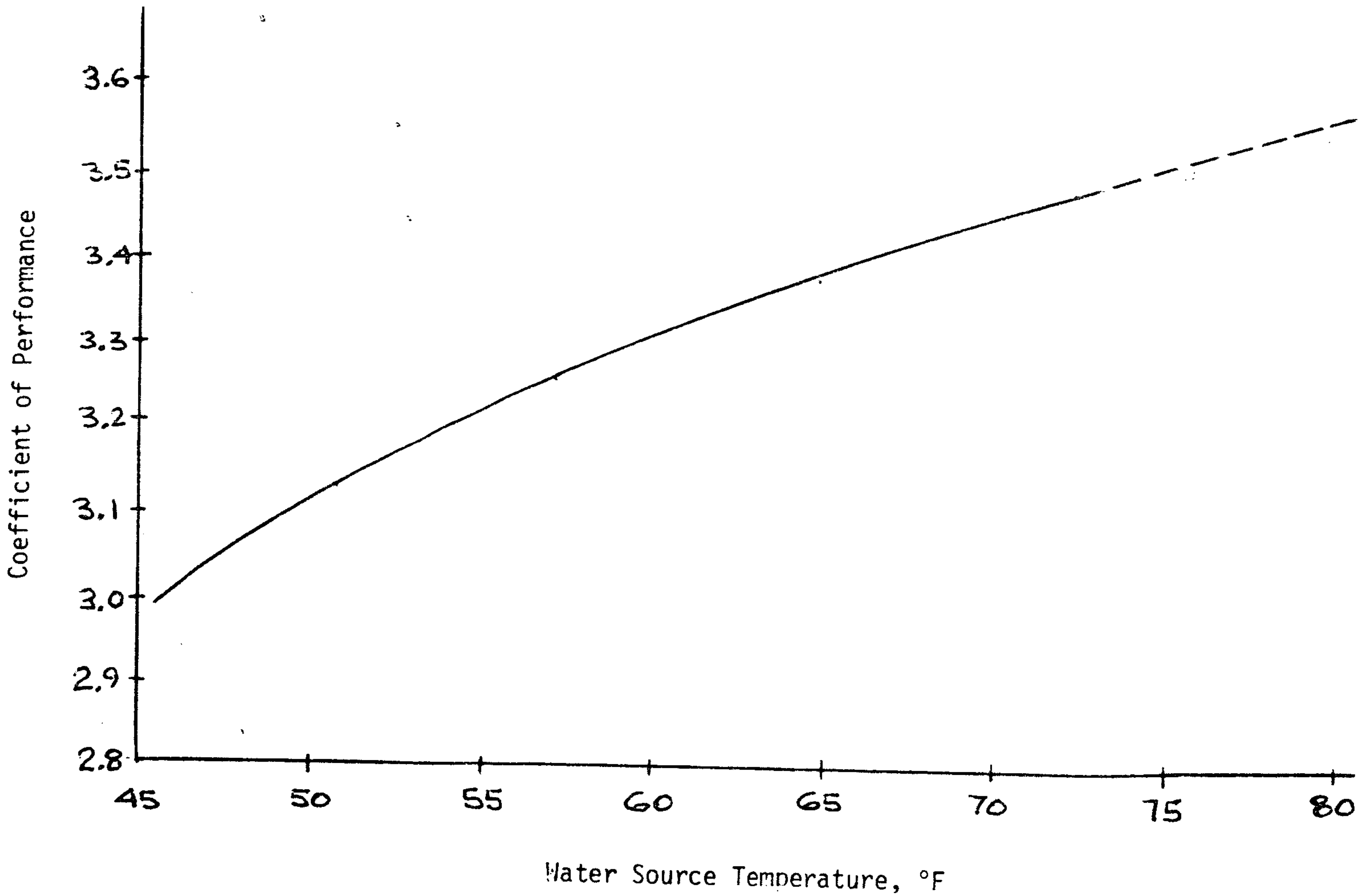


Figure 2. Coefficient of Performance of an Efficient Water Source Heat Pump as a function of Supply Temperature.

by the manufacturer. At 70°F water supply energy temperature, the C.O.P. is approximately 3.45; at 80°F supply the C.O.P. is approximately 3.55.

Contrasting this with a highly efficient air-to-air heat pump system (currently available model) the 17°F ambient air temperature C.O.P. is approximately 2.0, and this increases to about 2.6 at an ambient air temperature of 30°F. Clearly, these ambient temperatures are high for winter weather in much of the nation having geothermal energy; they are merely cited as examples of what is considered to be very efficient and economical heating in much of the country with air-to-air heat pumps.

Institutional Barriers

The only barrier that we have considered is that concerning the return of thermally spent water to the potable water distribution system. We have found no restrictions on this by either the state or local agencies in South Carolina, other than the requirement not to violate the contamination/impurity levels established by the U.S. Environmental Protection Agency for drinking water. A copy of the EPA requirements is included in the appendix.

For the direct heat exchange space heating application, a two pipe system would be used to supply the hot water from the street main to the heat exchanger(s) in the residence and then pump the used fluid back into the street main. In this case, a magnetic drive pump would be used and the entire system would be sealed with no reason for contamination after the system is put into operation. Any subsequent failure would simply result in water loss with no contamination of the potable water system.

There are possible contamination problems associated with the heat pump space heating approach. Use of a single wall potable water-to-refrigerant fluid heat exchanger would not be an acceptable approach, since a leak could result in oil and freon contamination of the water supply system. This problem has not been resolved, but we are formulating a conceptual design utilizing a double-jacket water-to-water-to-freon heat exchanger. In this approach, heat exchanger failure on either side would contaminate an isolated fluid only. Such designs are currently being implemented in solar hot water systems using silicone oil to heat the potable water supply, and these designs are being granted plumbing code approvals.

III. TYPICAL RESIDENTIAL COMMUNITY

To "fix" a residential area for study, we chose a real residential subdivision of 140 homes with its existing water supply piping. This community is known as Idlewild, and it is a part of the Greenville, S.C., metropolitan area, but we think that the street piping size, lengths and interconnections, as well as the water flow requirements, are typical of small residential areas. This community is served by a dual-loop main city water system, and there is a rather large industrial user of water (Michelin Tire factory) just downstream resulting in a water through-flow six-to-seven times as large as the consumptive residential flow. The piping layout is given in Figure 3.

The Ashmore Branch road water line is a 16-inch cast iron line connecting the two main dual-loop lines (not shown) of the Greenville water system. Part of the Idlewild community lies on each side of the Ashmore Branch road 16-inch water line; the major part is on the south side of this line.

A large percentage of the street distribution lines are 2-inch diameter; this raises the obvious question concerning the cost of pumping sufficient water for house heating through these small lines along relatively long streets. Note that street lengths in this community are on the order of 1000 ft.

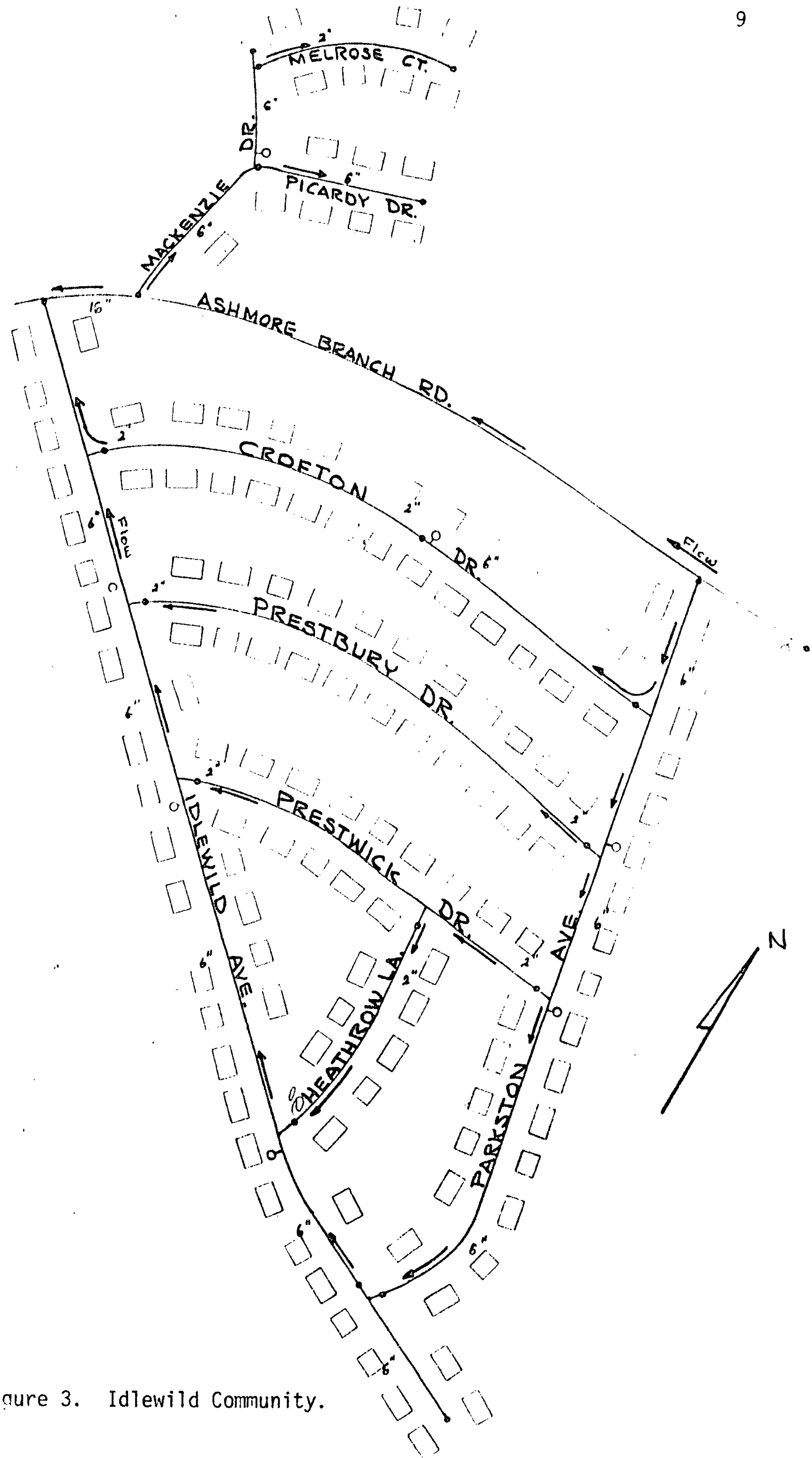


Figure 3. Idlewild Community.

An equally obvious concern is the thermal loss or temperature degradation of the water in these buried, uninsulated pipes. These problems are considered in detail in sections V and VIII of this report.

Before undertaking any of the engineering analyses related to supply of heating fluid, the thermal load must be established. For a typical 1800 sq. ft. residence in this community, the 0°F, 15 mph wind design heat load calculated using standard ASHRAE procedures is approximately 28,000 Btu/hr. Based on actual water use records, the average daily consumption of the 140-home community is 34,405 gallons per day. This represents an average household consumptive flow of 246 gallons per day. Assuming a maximum water temperature change of 35°F in a house heat exchanger, this flow rate would supply only about 10% of the design heating load; more precisely about 72,000 Btu/day. Fortunately, the through-flow in the 16-inch sub-main along Ashmore Branch road is rather steady due to the nature of the downstream industrial use, and this total flow averages 215,000 gallons per day. Dividing this among the 140 houses in the residential sector yields 1536 gallons per house-day. This would provide approximately 18,680 Btu/hr-house, or 2/3 of the design heating requirement.

While we have not analyzed the average daily cyclic heating requirement nor have we examined the cyclic or hourly industrial water flow rates, this overall energy balance consideration appears to be suitable for feasibility analyses. For much of the heating season, the hot water supply would be adequate for heating; part of the load in the coldest weather would be supplied by a topping system using fossil fuel or electric heat.

IV. THERMAL LOSSES BETWEEN PUMPING STATION AND RESIDENTIAL SITE

The Idlewild community is located approximately 16.68 miles from the main water treatment/pumping station. It is between the two main lines of a dual-loop system; one of these lines begins with 30-inch diameter pipe while the other begins with 42-inch diameter pipe. The 30-inch initial diameter main line normally supplies this community through the interconnecting 16-inch line along Ashmore Branch road.

The supply system consists of

(1) 4.4 mi of 30-inch pipe,
(2) 1.0 mi of 24-inch pipe,
(3) 3.61 mi of 20-inch pipe,
(4) 6.17 mi of 12-inch pipe, and
(5) 1.5 mi of 16-inch pipe

in the order listed. All piping is cast iron. The 16-inch pipe downstream of the 12-inch pipe seems illogical, but this resulted from growth of the system.

The temperature degradation occuring in this 16.68 mile distance depends heavily upon the flow rate which is varying and certainly not precisely known at any point in the system. The total Greenville water supply system has an average flow of 23 x 10^6 gallons per day. Prorating this between the two main loop lines, we assigned a flow of 3.32 x 10^6 lbm/hr to the 30-inch (initial size) loop main. For simplicity, we modeled the useage upstream of Idlewild by a linear decrease with distance expression. Since Idlewild is near the end of this loop and has a total flow of 2.16 x 10^{-5} lbm/hr, the linear expression for the flow at any distance L from the treatment/pumping station is

$$\dot{m} \simeq (3.32 \times 10^6 - 35L)\ lb_m/hr \qquad (1)$$

where L is in feet.

Using equation (1) to determine the mid-point mass flow rate in each size pipe, the temperature drop over each section was calculated by the steady-state conduction shape factor method. The heat transfer rate from a given length L of pipe buried in homogeneous soil having a constant pipe surface temperature T_1 and soil surface temperature T_2 is

$$q = SkL(T_1 - T_2) \tag{2}$$

where: k = soil thermal conductivity

L = Pipe length

S = conduction shape factor

The soil conductivity is dependent upon both soil type and moisture content. Values of k reported in the literature[1] range from 0.3 to 1.0 Btu/hr-ft-°F. For the present calculation, we used a value of 0.62 Btu/hr-ft-°F, a value reported for earth with 42% moisture -- very wet!

The analytical result for the shape factor S for an infinitely long cylinder with its centerline a uniform distance Z below the surface of the soil and having a radius r is

$$\frac{S}{L} = \frac{2\pi}{\cosh^{-1}(Z/r)} \tag{3}$$

as given in most heat transfer texts.[2]

An energy balance on the fluid flowing through the pipe is

$$\dot{m} C_p (\Delta T)_{fluid} = q = SkL(T_1 - T_2) \tag{4}$$

or

$$(\Delta T)_{fluid} = \frac{SkL(T_1 - T_2)}{\dot{m} C_p} \tag{5}$$

[1] See, for example, Y. Rzheysky and G. Novik, "The Physics of Rocks," p. 143, which reports values of approx. 0.4 to 0.75 Btu/hr-ft-°F.

[2] See, for example, "Elements of Transport Phenomena" by L.E. Sissom and D.R. Pitts, McGraw Hill, 1972, p. 137.

Using this to obtain a crude approximation over each pipe section, we obtain for a 3-ft buried depth with T_1 equal to 135°F and T_2 equal to 0°F, the following results:

Pipe Section	Length	Diameter	$(\Delta T)_{fluid}$
1	23,280 ft	30"	2.00°F
2	5,280 ft	24"	0.83°F
3	19,040 ft	20"	2.89°F
4	32,568 ft	12"	5.35°F
5	7,920 ft	16"	4.74°F
Totals	88,440 ft		15.82°F

While this is a rough approximation, the use of the very high k value assures us that this an upper limit of the thermal degradation that would occur. A more probable value of k is about 0.5 Btu/hr-ft-°F.

Assuming a geothermal resource of 150°F at the pumping station indicates that 135°F water would be available for heating in the residential community 16.68 miles away. This is, of course, a rather extreme case. Most communities having nearby geothermal resources are much smaller than the city we were considering, and the thermal degradation in the main distribution line should be considerably less than in the case presented. Note in particular the approximate 5°F temperature loss associated with the final 1.5 miles of 16-inch pipe; this would be more nearly the situation for a small town.

V. THERMAL LOSSES IN UNINSULATED SUBDIVISION STREET DISTRIBUTION LINES

As in the main system distribution line, the losses in the uninsulated subdivision street water lines depends upon pipe depth, pipe diameter, soil conductivity, pipe surface (water) temperature and soil surface temperature. In addition, we have employed a 4X factor to the flow required for heating the houses along a street so that the thermal degradation due to extraction of energy for house heating will not overly reduce the temperature along a given street; otherwise the end houses along the street would be unable to effectively use the fluid. In addition to the preceding factors affecting the conductive heat transfer rate, the

- total flow rate,
- length of street, and
- distance between houses

all are factors influencing the fluid temperature decay by conduction in the uninsulated lines.

Since many of the streets have 2-inch distribution lines, study focuses on this size. Clearly a parametric study is necessary. We chose to consider

- 3 supply water temperatures -- 135°F, 125°F, and 115°F,
- house lot widths from 80 to 140 ft,
- a street length of 1000 ft (not critical to parametric study -- causes some end effect),
- a single distribution line depth of 3 ft,
- thermal conductivities from 0.3 to 1.0 Btu/hr-ft-°F,
- a soil surface temperature of 0°F,
- a heating load of 18000 Btu/hr-house,
- a 4X water flow factor

A parametric study of the water temperature decay due to steady-state, conductive heat transfer was carried out using the shape factor approach of equation (5) and assuming the pipe surface temperature to be the same as the water supply temperature. The results are given in Figures 4, 5, and 6. These depict the temperature loss due to conduction to the soil per 1000 ft of street length as a function of house spacing and soil thermal conductivity with all other parameters fixed. Of course, as the supply water temperature diminishes, the mass flow rate increases to provide the same heating to each house, and the thermal degradation diminishes. Also, the closer the house spacing, the lower the temperature conductive decay.

VI. SIZE AND COST OF MAIN HEAT EXCHANGER

An estimate of the size and cost of the main heat exchanger to supply thermal energy to the drinking water at the water treatment/pumping station is a necessary element for a cost study of the total system. Sizing of this unit is dependent upon

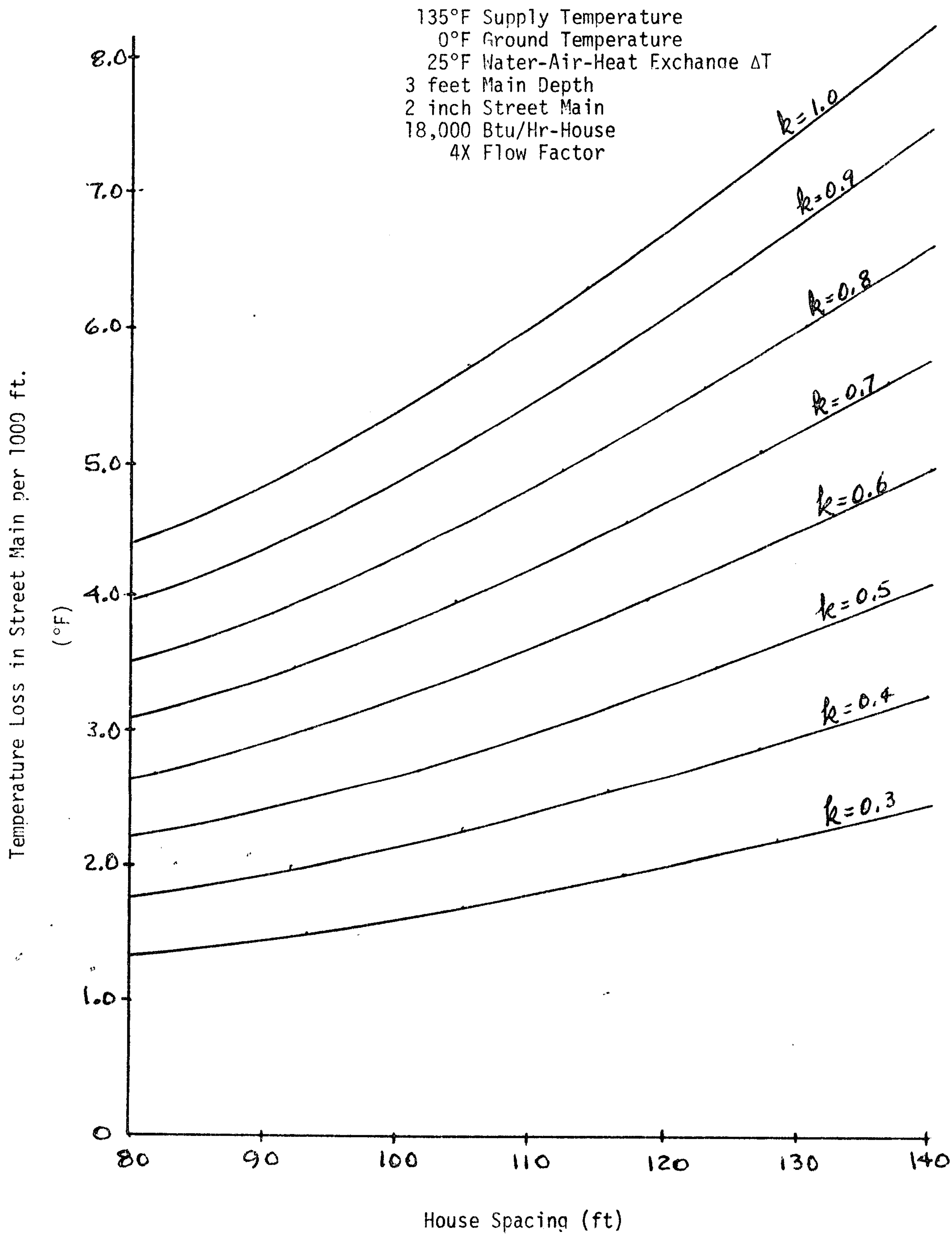


Figure 4. Temperature Loss in Uninsulated Street Water Lines For 135°F Supply Water Temperature.

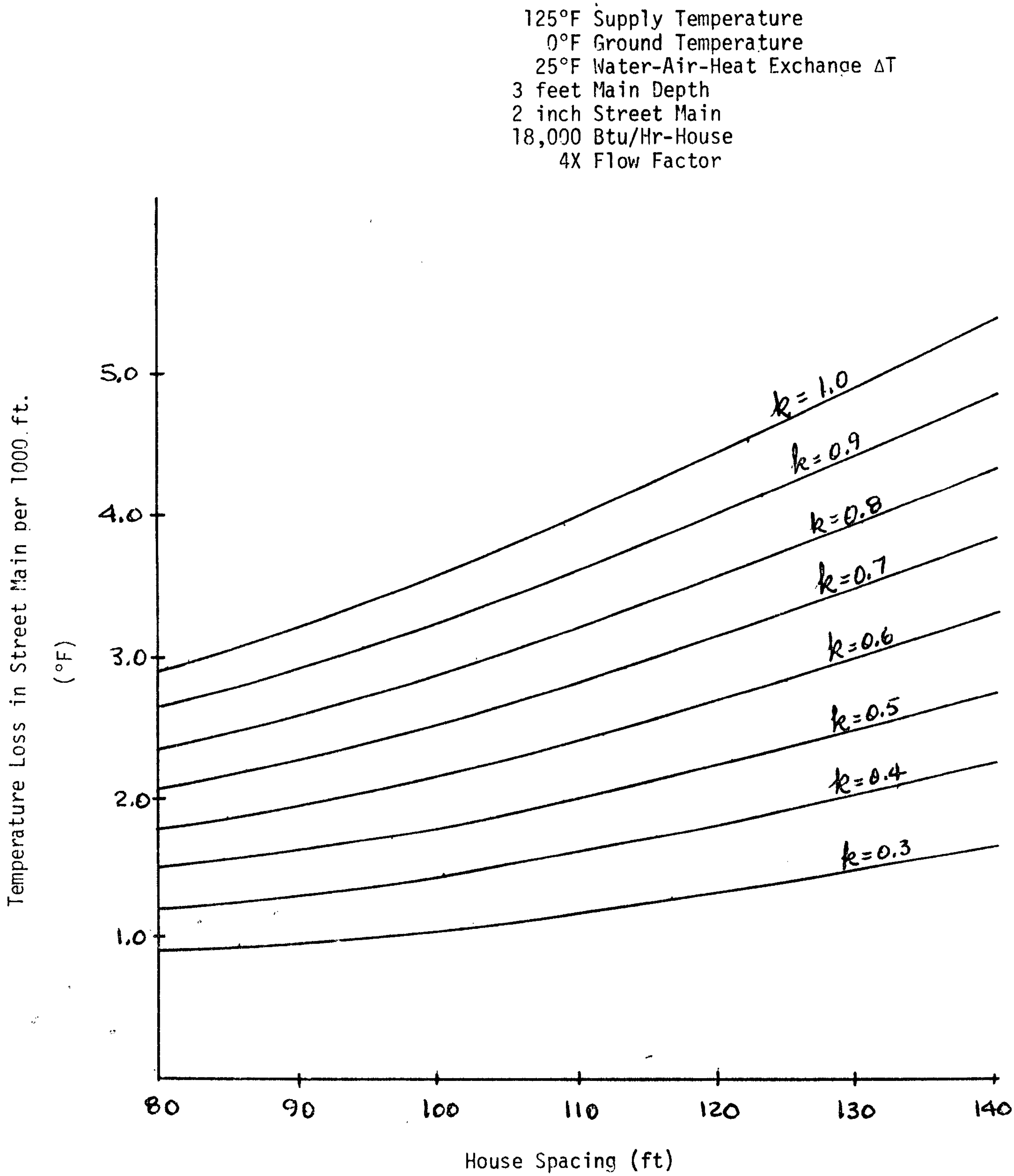


Figure 5. Temperature Loss in Uninsulated Street Water Lines For 125°F Supply Water Temperature.

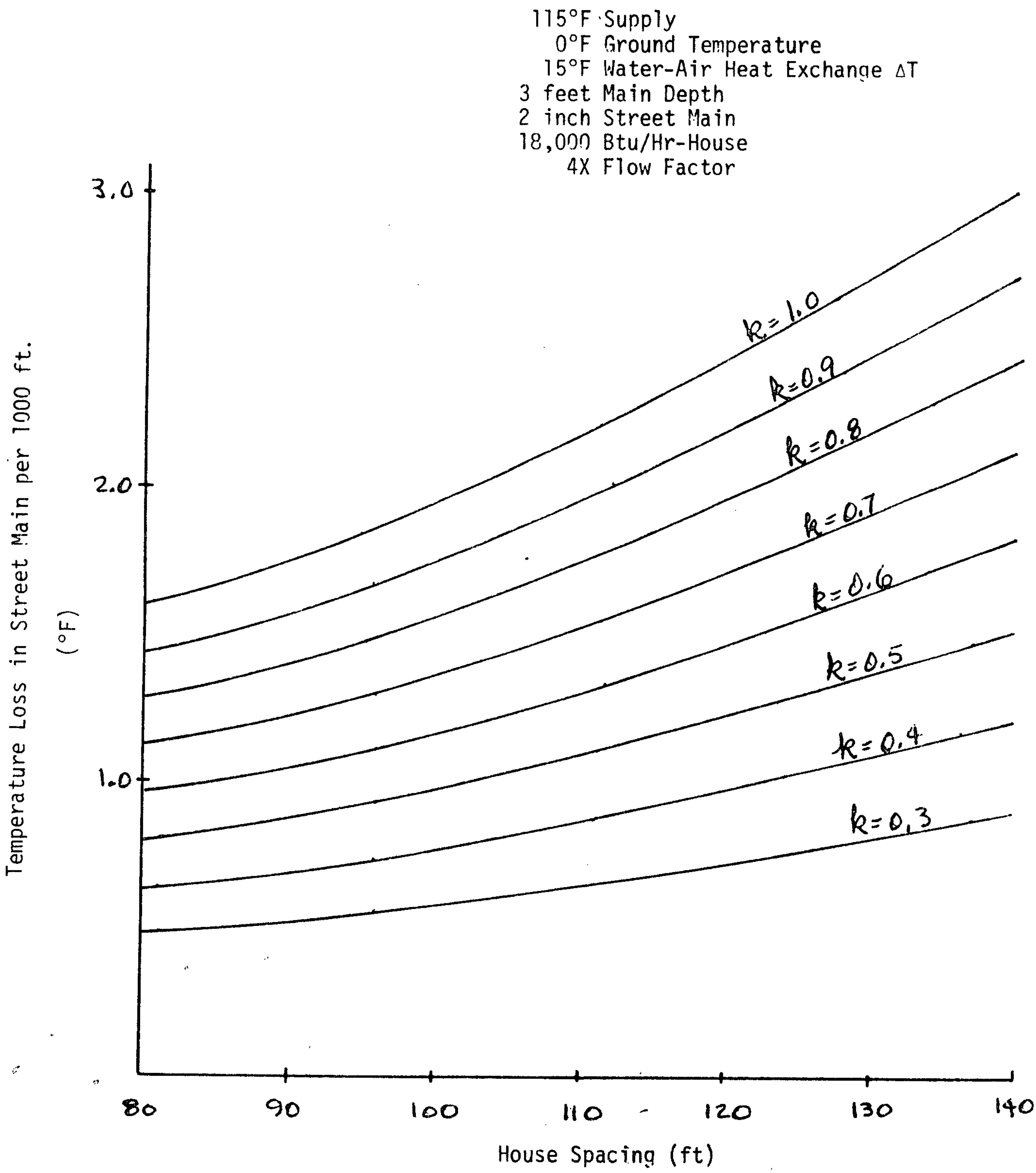


Figure 6. Temperature Loss in Uninsulated Street Water Lines For 115°F Supply Water Temperature.

- configuration-type of heat exchanger
- hot-side geothermal fluid temperature
- cold-side supply temperature, and
- cold-side exit temperature (heating fluid for residences).

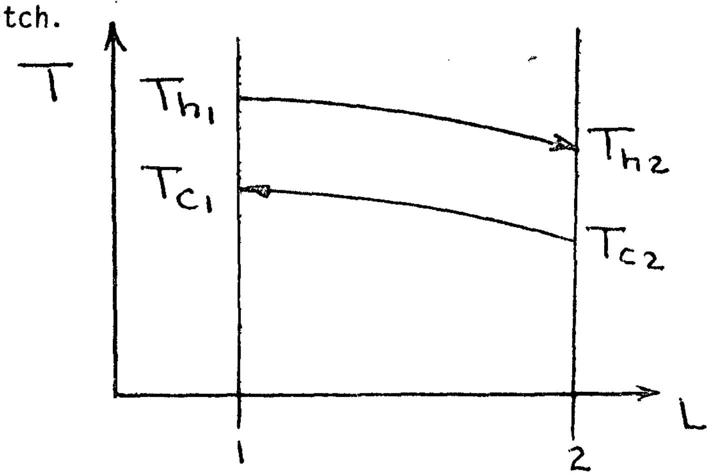
All of the fluid temperatures are dependant upon local conditions, but to determine ranges of main heat exchanger size and cost, a parametric study was carried out. The basic design was stipulated to be a shell-and-tube unit, which is generally considered to be the most practical large heat exchanger design. The ranges of parameters studied were:

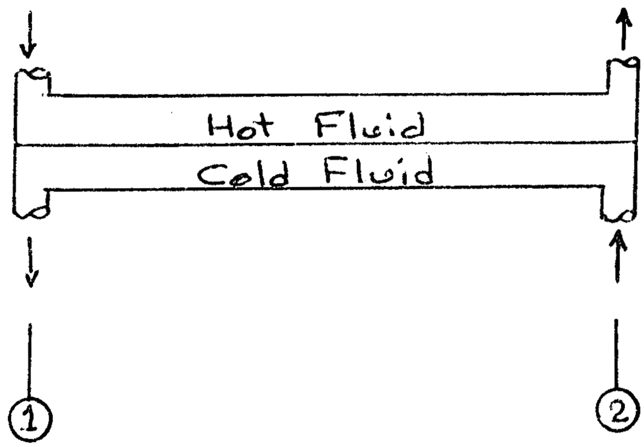
Geothermal fluid supply temparature	--	160 - 200°F
Cold side fluid discharge temperature	--	115 - 135°F
Cold side fluid supply temperature	--	45°F
Overall heat transfer coefficient (Btu/hr-ft^2-°F)	--	110 - 170
Shell side fluid passes	--	one and two

The heat exchanger size calculation begins with the calculation of the Log Mean Temperature Difference defined by

$$LMTD = (\Delta T)_{lm} = \frac{(T_{h2} - T_{c2}) - (T_{h1} - T_{c1})}{\ln[(T_{h2} - T_{c2})/(T_{h1} - T_{c1})]} \quad (6)$$

where the temperatures are as defined for a counterflow heat exchanger in the following sketch.





To proceed, we assumed

1. $T_{c_1} = 135°F$, $125°F$ or $115°F$
2. $T_{c_2} = 45°F$
3. $(T_{h_2} - T_{c_1}) = 10°F$

Then from charts in standard heat transfer texts, F values for one- and two- shell pass shell-and- tube heat exchangers can be obtained with the specified temperatures, and then the total heat transfer is given by

$$q = UAF(\Delta T)_{lm} \tag{7}$$

The total q to be supplied by the heat exchanger is the product of total flow rate, specific heat, and temperature rise provided by the heat exchanger,

$$q = \dot{m}C_p\ (T_{c_1} - T_{c_2}) \tag{8}$$

The mass flow rate is obtained from the thermal load and fluid ΔT at the residences, $(\Delta T)_{house}$, viz

$$\dot{m}C_p(\Delta T)_{house} = 4(140)(18,667)\frac{Btu}{hr} \tag{9}$$

where we have included the 4X flow factor. For a specified set of conditions including fluid ΔT at the residences, all temperatures at the main heat exchanger and a value of U, the combination of equations (7), (8) and (9) permit calculation of the heat exchanger area A.

Parametric results are given in Figures 7, 8, 9, 10, 11, and 12. These show the main heat exchanger size for a 140 house residential heating system under the earlier assumptions of (i) 2/3 of the design heating load of 28,000 Btu/hr-house supplied by the hot water and (ii) a 4X flow factor used to prevent excessive temperature decay along a given street.

Cost estimates have been obtained for carbon steel heat exchangers using the usual power-law scaling factor applied to reported costs of units 2 to 5 times the size needed inthe present application. In 1979 dollars, the range of main heat exchanger cost on a per home basis for the 140 home community is from $320 to $760. This cost is more dependent upon the geothermal fluid supply temperature than any other variable.

An interesting feature of these results is that the heat exchanger size and cost increase with decreasing potable water discharge temperature. This is because the total thermal residential load remains constant, hence for a given U, which also fixes velocities, the total heat exchange area increases with decreasing exit cold side (potable water) temperature.

This parametric study should be extended to include (i) 140° and 150°F geothermal fluid supply temperatures and (ii) higher potable water discharge temperatures, where possible, from the main heat exchanger.

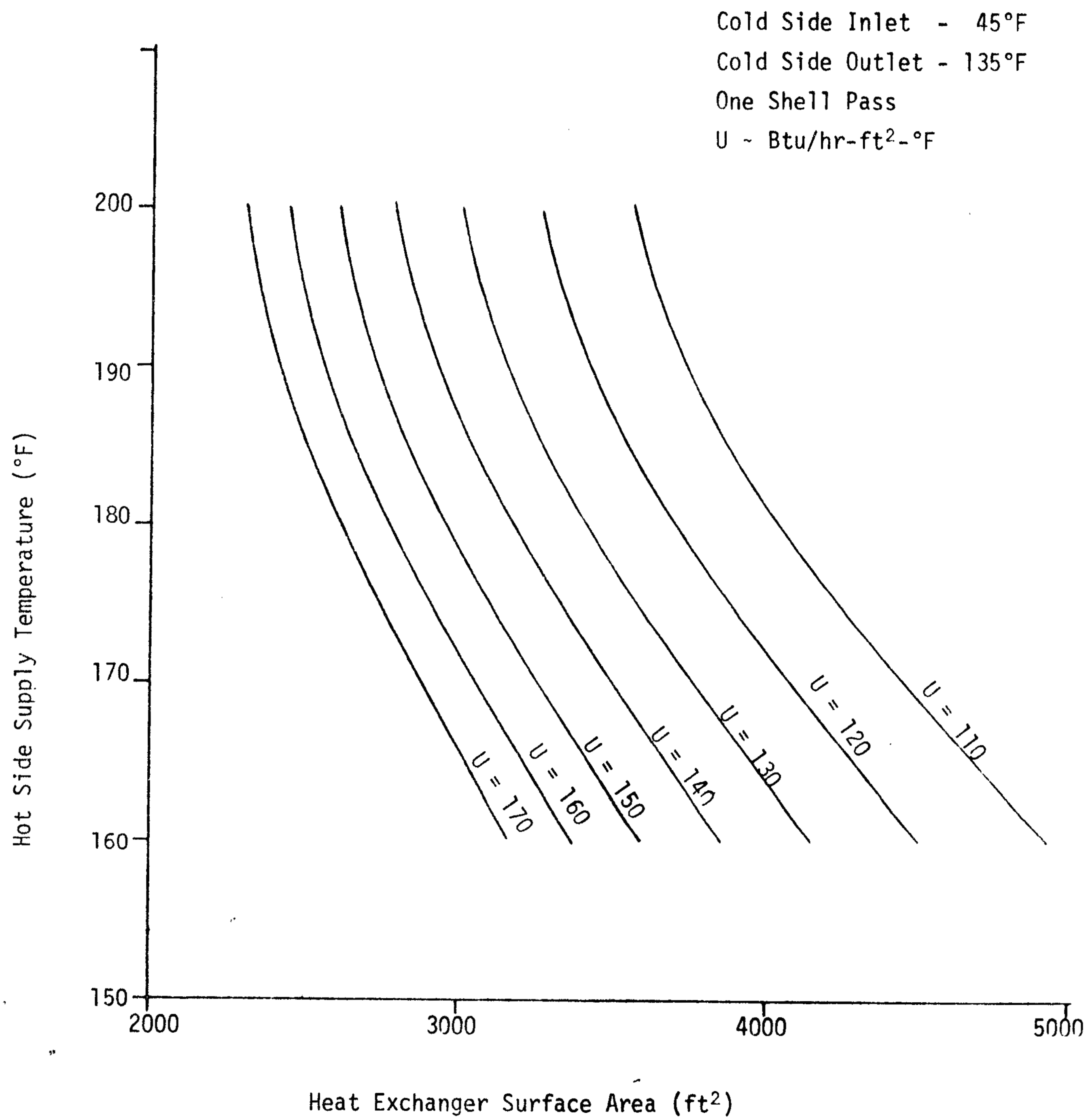


Figure 7. Main Heat Exchanger Size as a Function of Geothermal Fluid Supply Temperature and U Value, 135°F Cold Side Outlet, One Shell Side Pass.

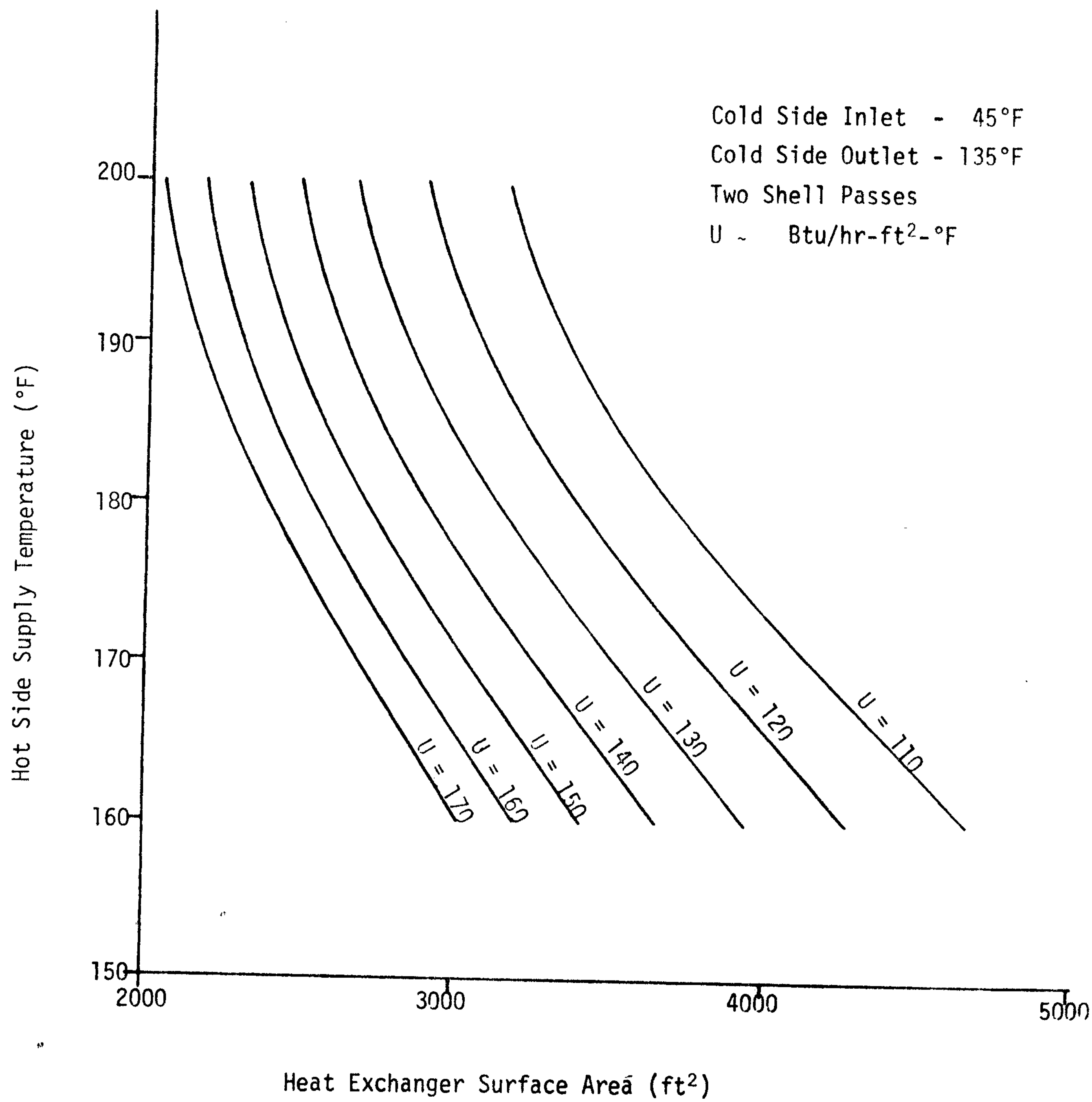


Figure 8. Main Heat Exchanger Size as a Function of Geothermal Fluid Supply Temperature and U Value, 135°F Cold Side Outlet, Two Shell Side Passes.

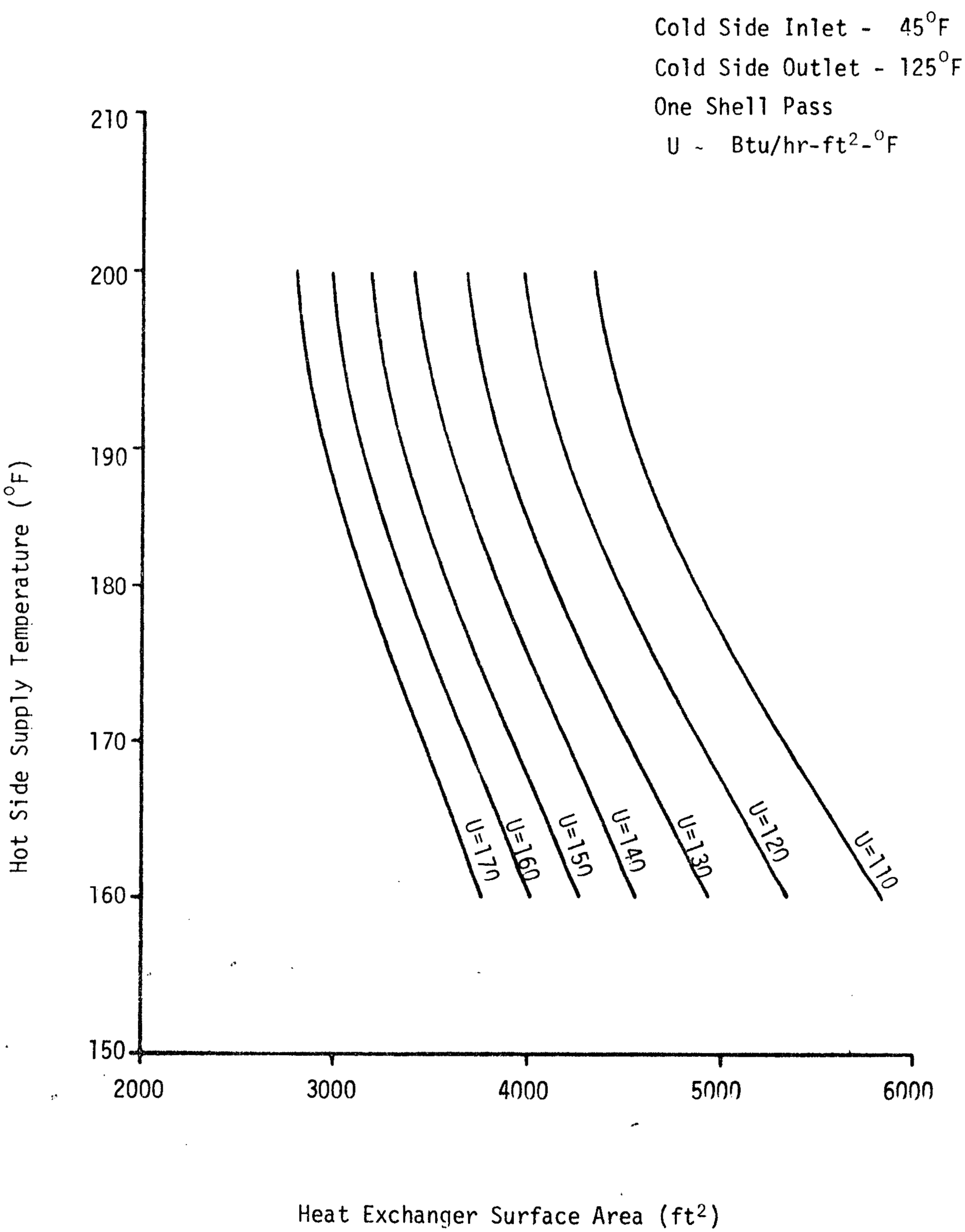


Figure 9. Main Heat Exchanger Size as a Function of Geothermal Fluid Supply Temperature and U Value, 125°F Cold Side Outlet, One Shell Side Pass.

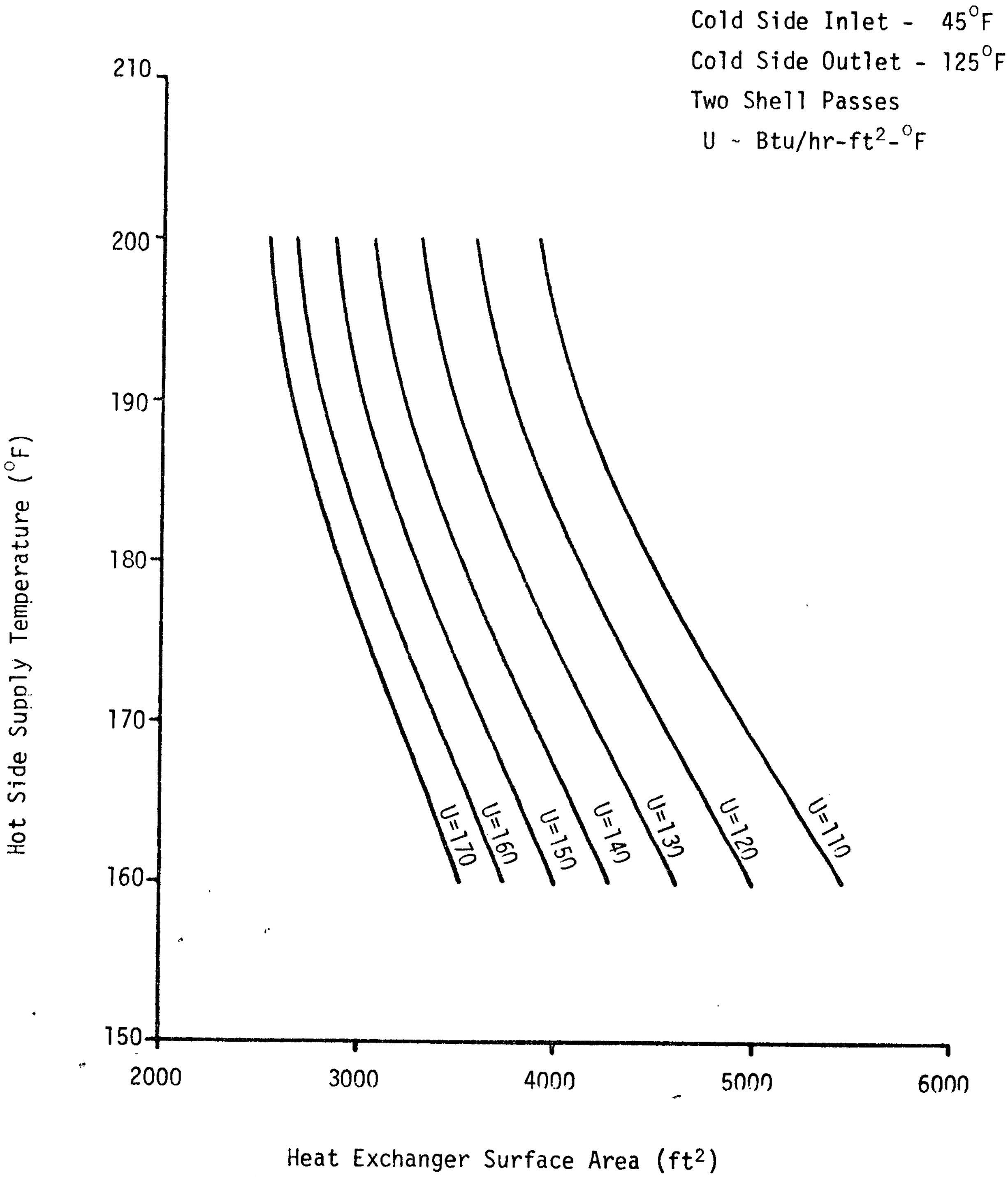


Figure 10. Main Heat Exchanger Size as a Function of Geothermal Fluid Supply Temperature and U Value, 125°F Cold Side Outlet, Two Shell Side Passes.

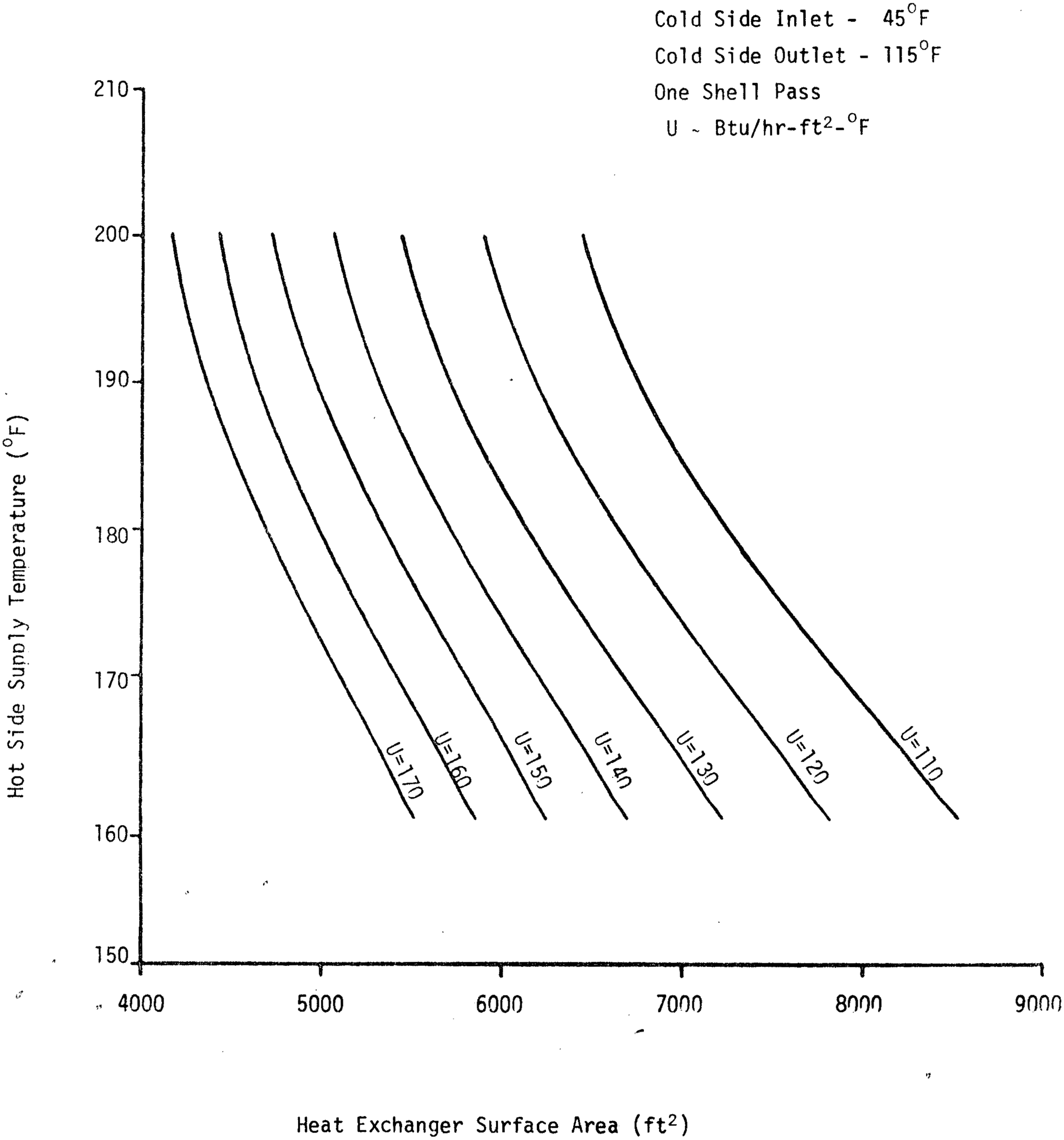


Figure 11. Main Heat Exchanger Size as a Function of Geothermal Fluid Supply Temperature and U Value, 115°F Cold Side Outlet, One Shell Side Pass.

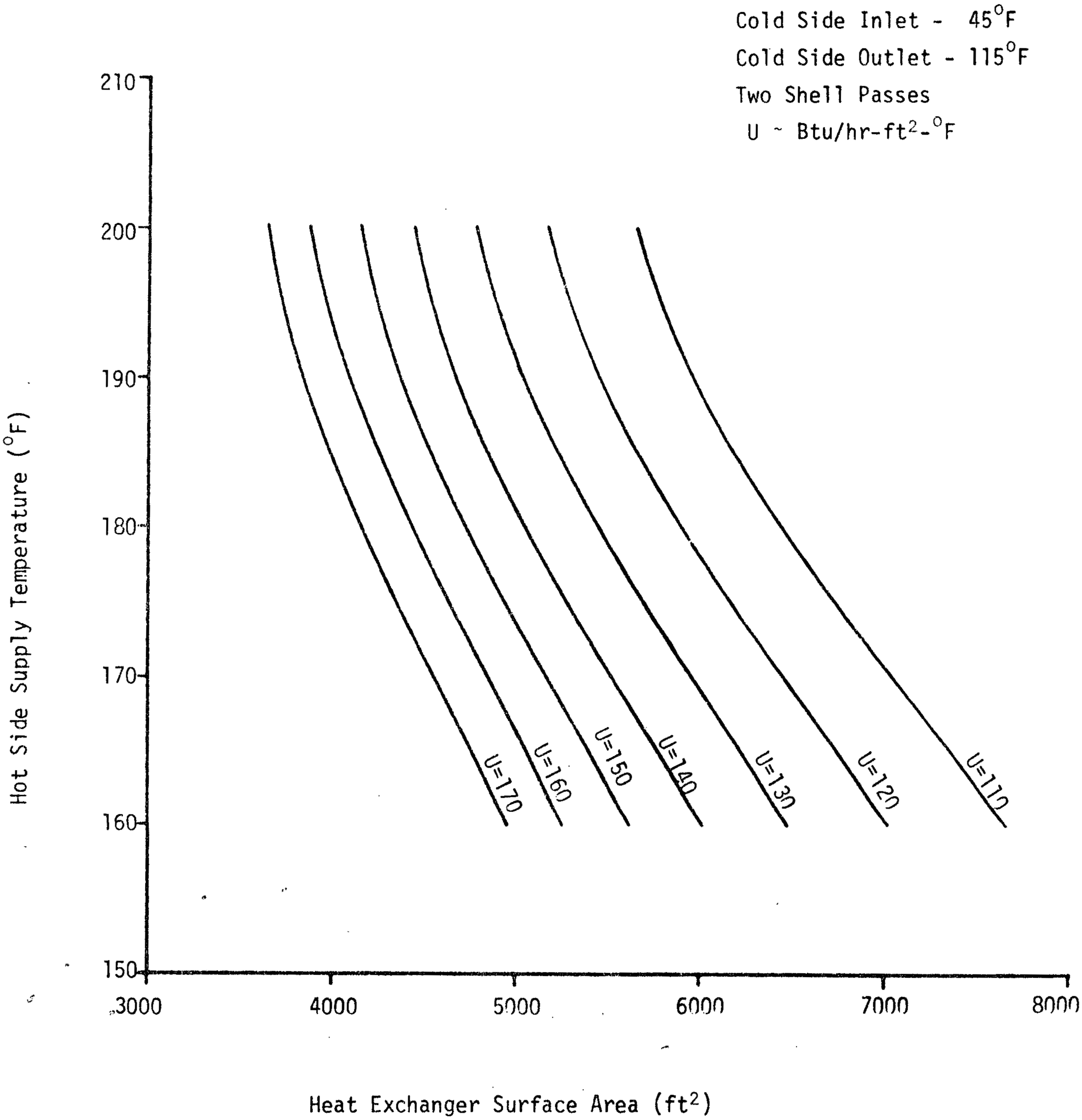


Figure 12. Main Heat Exchanger Size as a Function of Geothermal Fluid Supply Temperature and U Value, 115°F Cold Side Outlet, Two Shell Side Passes.

VII. SELECTION AND SIZING OF INDIVIDUAL HOUSE HEAT EXCHANGERS

For direct space heating, the simplest house system modification would result when the existing heating unit is a forced hot air, central heating system. In this case, the modification would consist of (i) the addition of a crossflow finned tube water-to-air heat exchanger in the existing furnace system return air plennum or other suitable ductwork location, and (ii) a modification to increase the blower speed and/or size to accommodate the increased flow resistance due to the additional heat exchanger.

Existing central systems including air conditioning may require a bootstrap duct and blower addition to provide the necessary flow rate for the added heat exchanger pressure drop. In general, older furnace systems having vee-belt drive blower systems can be modified by changing pulley sizes and drive motors -- newer furnace systems using integral blower/motor units can often be increased in motor speed electrically; otherwise these require a bootstrap added blower. In either case, the added water-to-air heat exchanger should be upstream of the furnace heating unit since operation of both with the arrangement reversed would result in significant furnace heat being supplied to the hot water, a very undesirable effect.

In the absence of a central forced air system, two other possible designs appear to be suitable. The first of these would be the installation of baseboard water-to-air free convective units of the type commonly used in hydroponic central hot water systems. The hot water supplied through the district heating system must not be mixed in any way with a central hot water house heating system, however. The baseboard heaters must be used only with the potable water. Also, the use of room-sized forced air wall convectors, such as being introduced by some of the solar heating equipment manufacturers, appears to be an attractive supplemental system concept.

To get an idea of system costs, we directed our attention primarily to the add-on water-to-air heat exchanger for existing forced air central systems. In every case, a new heat exchanger is required, and we conducted a study of the size of unit needed. The unit design considered was the conventional cross-flow, finned-tube type with the hot water in the tubes. The values of parameters used in the analyses were:

- thermal load ---------------------------------- 18,666 Btu/hr
- air exit temperatures ------------------- 110°F, 100°F, 90°F
- air inlet temperature --------------------------------- 65°F
- water inlet temperatures --------------- 135°F, 125°F, 115°F
- water exit temperature -------------------------------- 100°F
- overall U values (Btu/hr-ft^2-°F) ---------------- 5, 7.5, 10

Using the Log Mean Temperature Difference of equation (6) together with equation (7) and F-factors from the heat transfer literature for cross-flow, finned-tube heat exchangers with both fluids unmixed, the required surface area for the 18,667 Btu/hr thermal load can be calculated. A set of computations covering the above listed ranges of system parameters was carried out, and the results are summarized in Figures 13, 14, and 15.

VIII. STREET WATER LINE PUMPING REQUIREMENTS

The increased water flow rate through relatively small distribution lines within typical residential areas could result in a significant added pump power requirement. This increased water flow rate will be an inevitable consequence of using the water supply system for heating purposes. It should be noted that in the community considered in this study, the 4X flow factor is applied to a flow rate that has been increased by a factor of 6 above the consumptive flow -- the flow is then about 24 times the normal

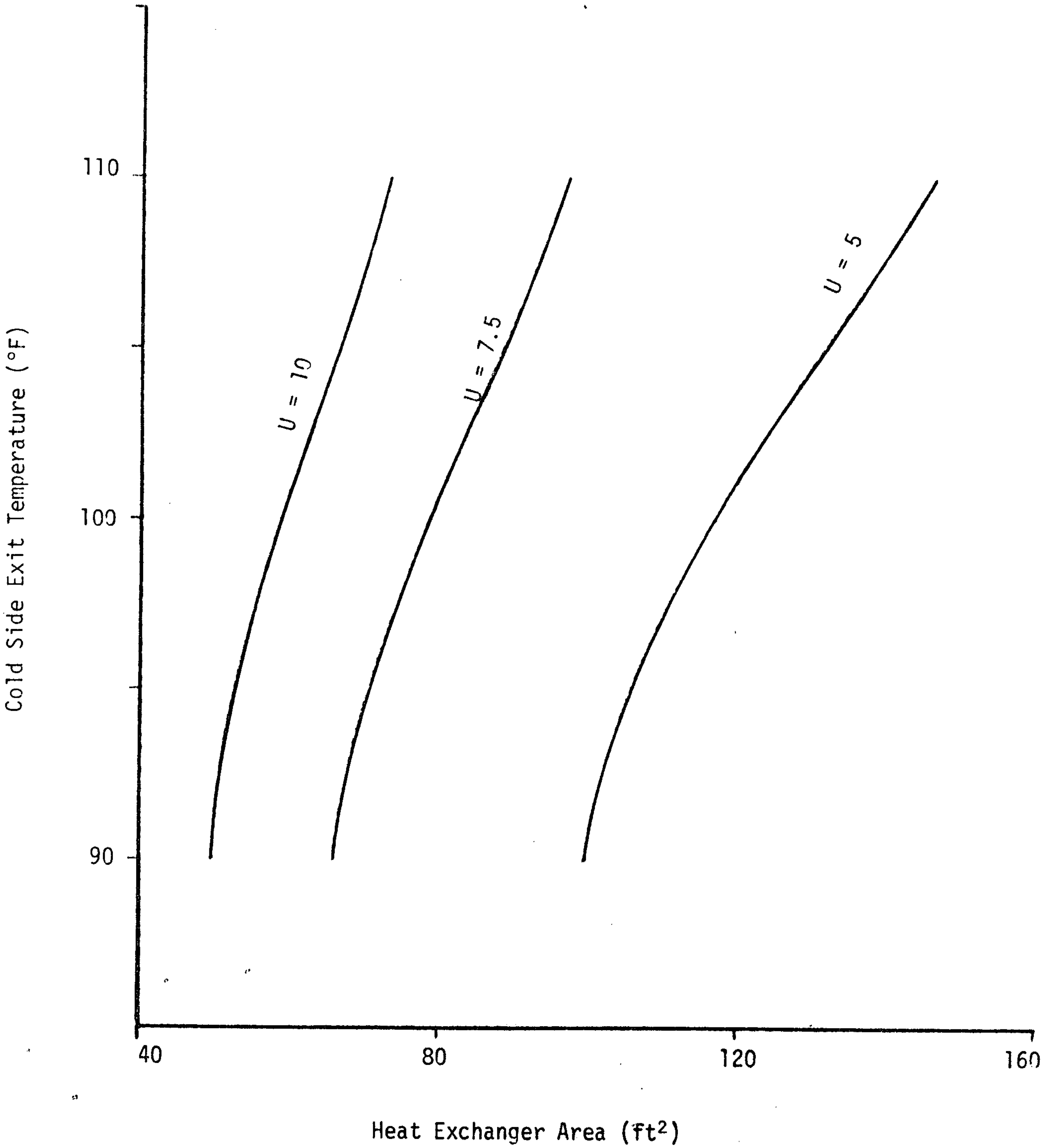


Figure 13. Heat Exchanger Area Versus Cold Side Exit Temperature for 135°F Supply Temperature.

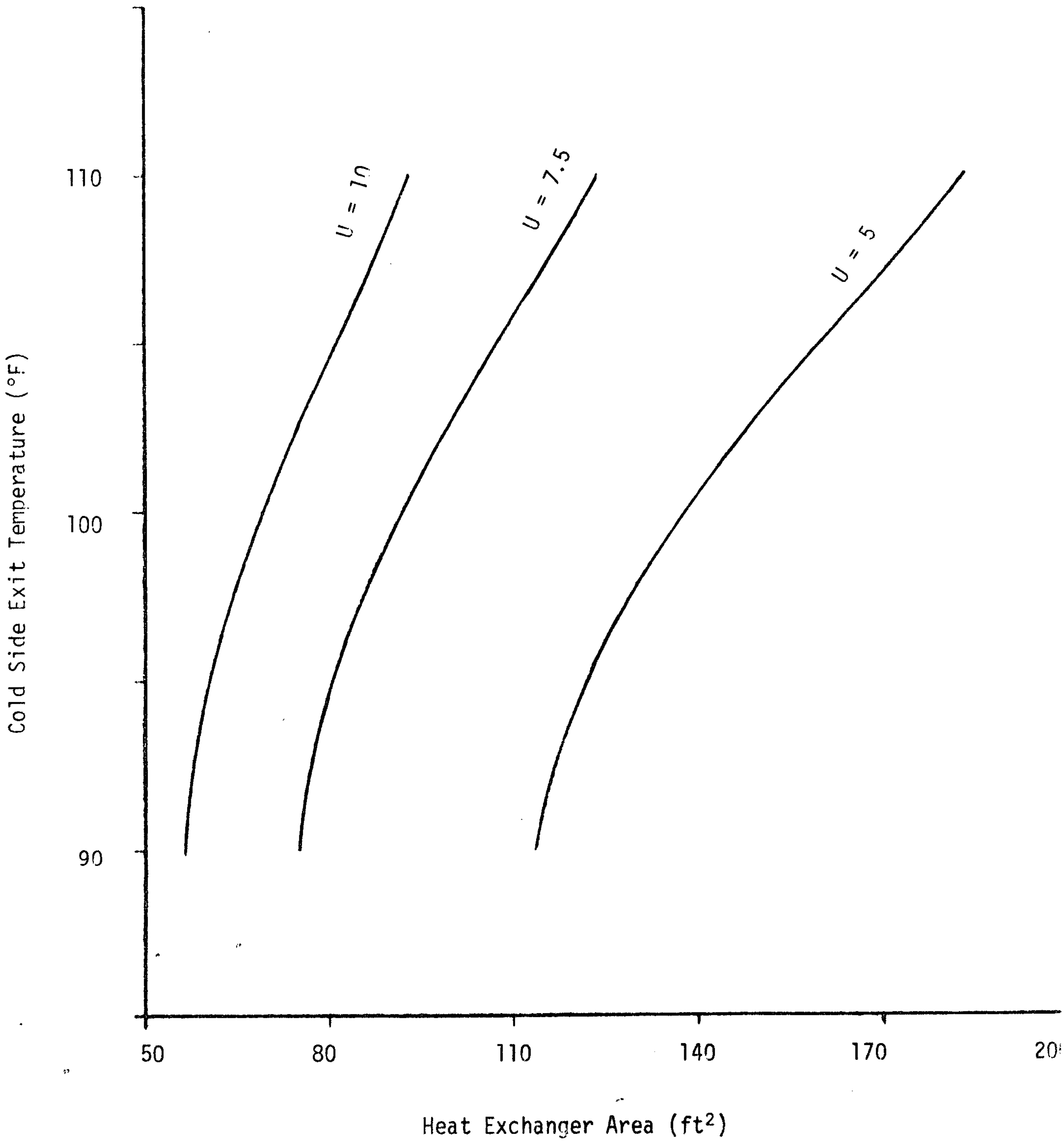


Figure 14. Heat Exchanger Area Versus Cold Side Exit Temperature for 125°F Supply Temperature.

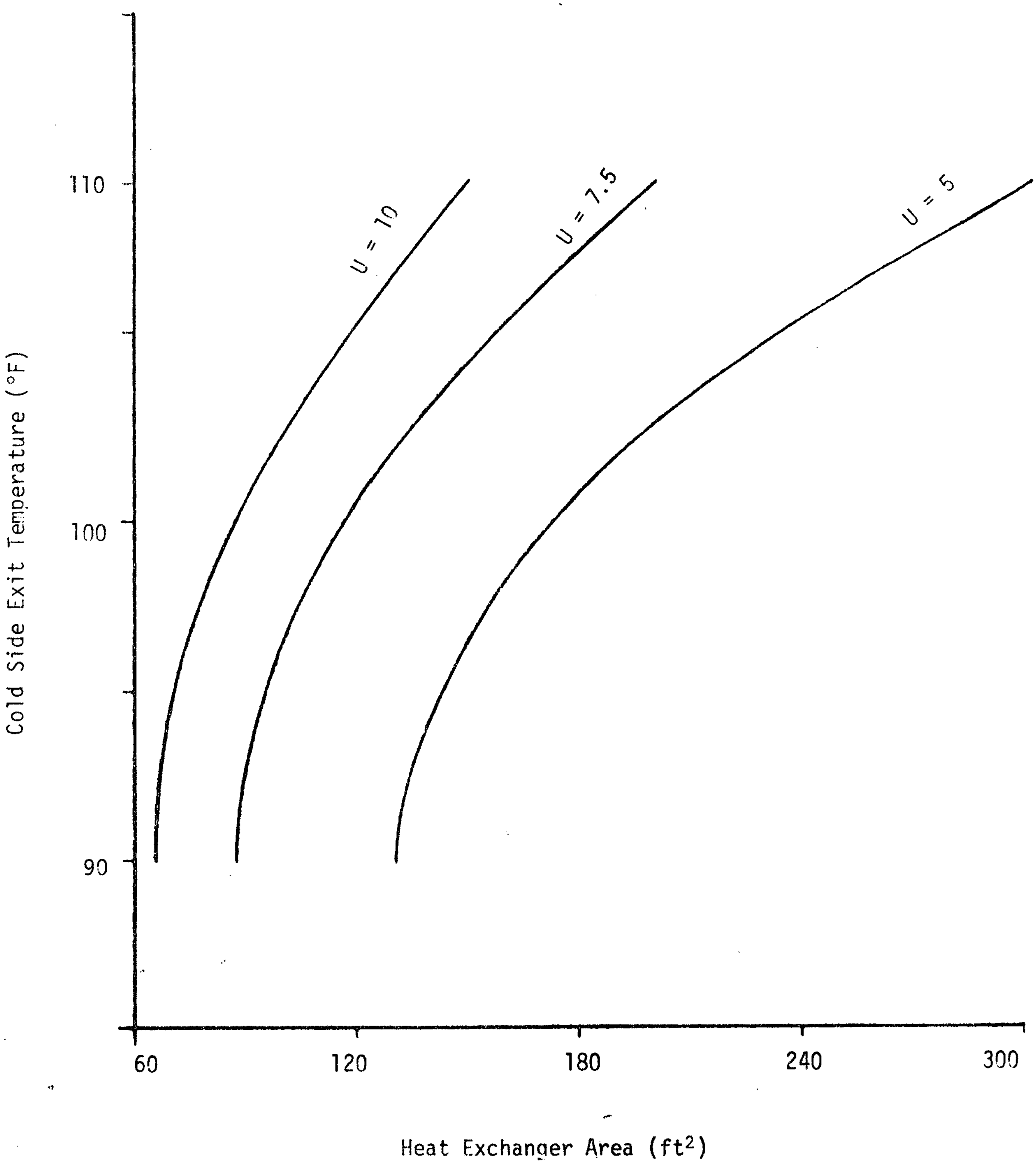


Figure 15. Heat Exchanger Area Versus Cold Side Exit Temperature for 115°F Supply Temperature.

residential flow rate! The preponderance of 2-inch lines in this community raises the question of increased pumping power.

Because of the large differences in flow rate through the various lines and because of the decreasing nature of the flow along a given street due to consumptive residential flow, manual computations of the pumping power are very tedious. Consequently, a computer program in Fortran IV has been prepared to carry out these calculations. The computer program flow chart is given as Figure 16, and a complete listing of the program is included in the appendix.

Briefly stated this program reads the following information:

1. density ρ and viscosity μ of the water
2. relative roughness e of the pipe
3. amount of water required per house for heating, QH
4. amount of water required per house for consumption, QD
5. number of houses in sub-division, NH
6. number of houses on a section of pipe, H, and pipe section length and diameter, L and D
7. a program control number which indicates the equation used to calculate water flow rate

The community is divided into sections that consist of lengths of pipe between houses.

The information of 6 and 7 above is "read-in" for each section of pipe. Depending upon the control number read, the program will use a specified equation to calculate the water flow rate in that pipe section. The water consumed by the house at the end of the section is subtracted to obtain the flow rate for the next pipe section.

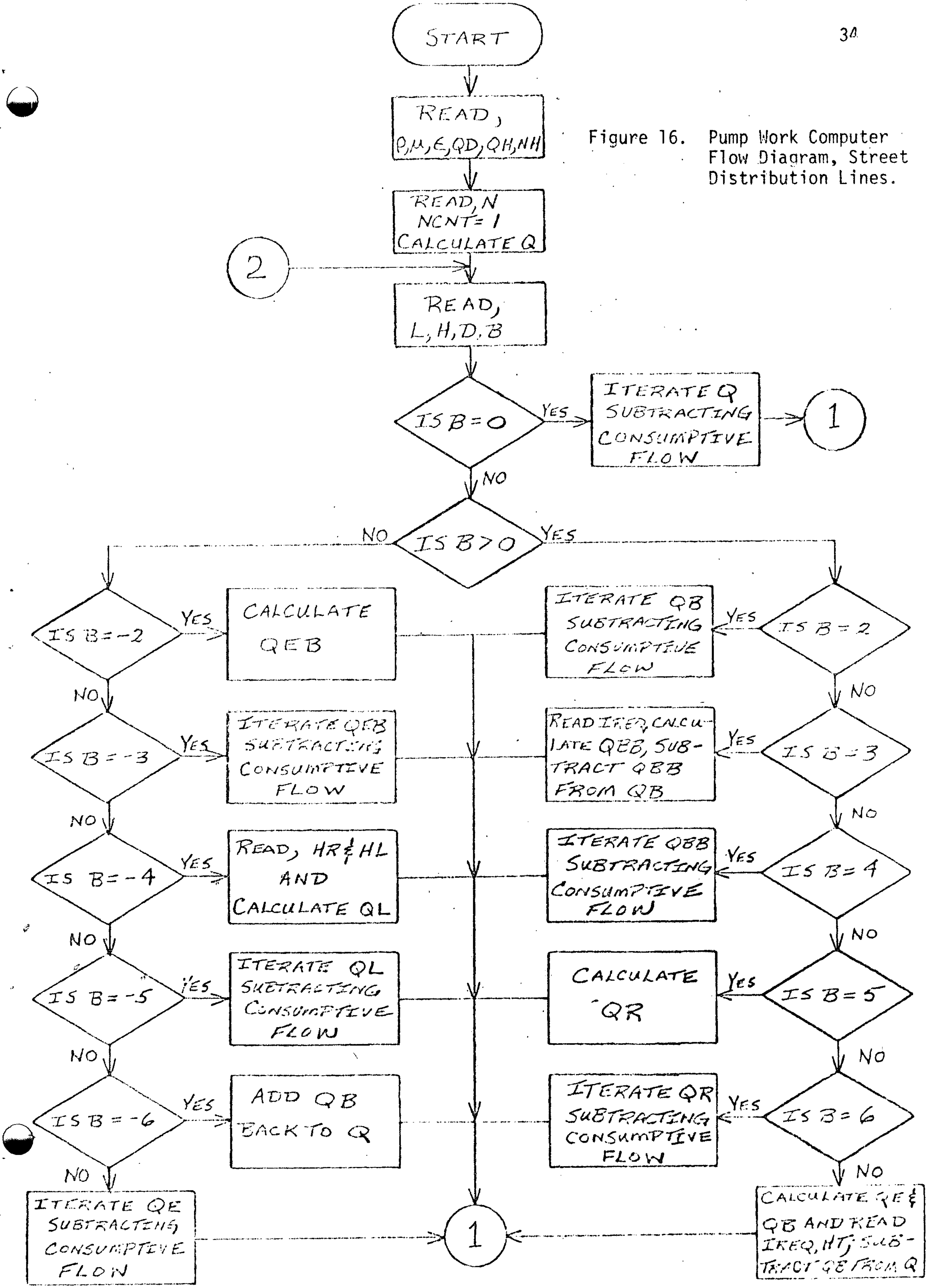


Figure 16. Pump Work Computer Flow Diagram, Street Distribution Lines.

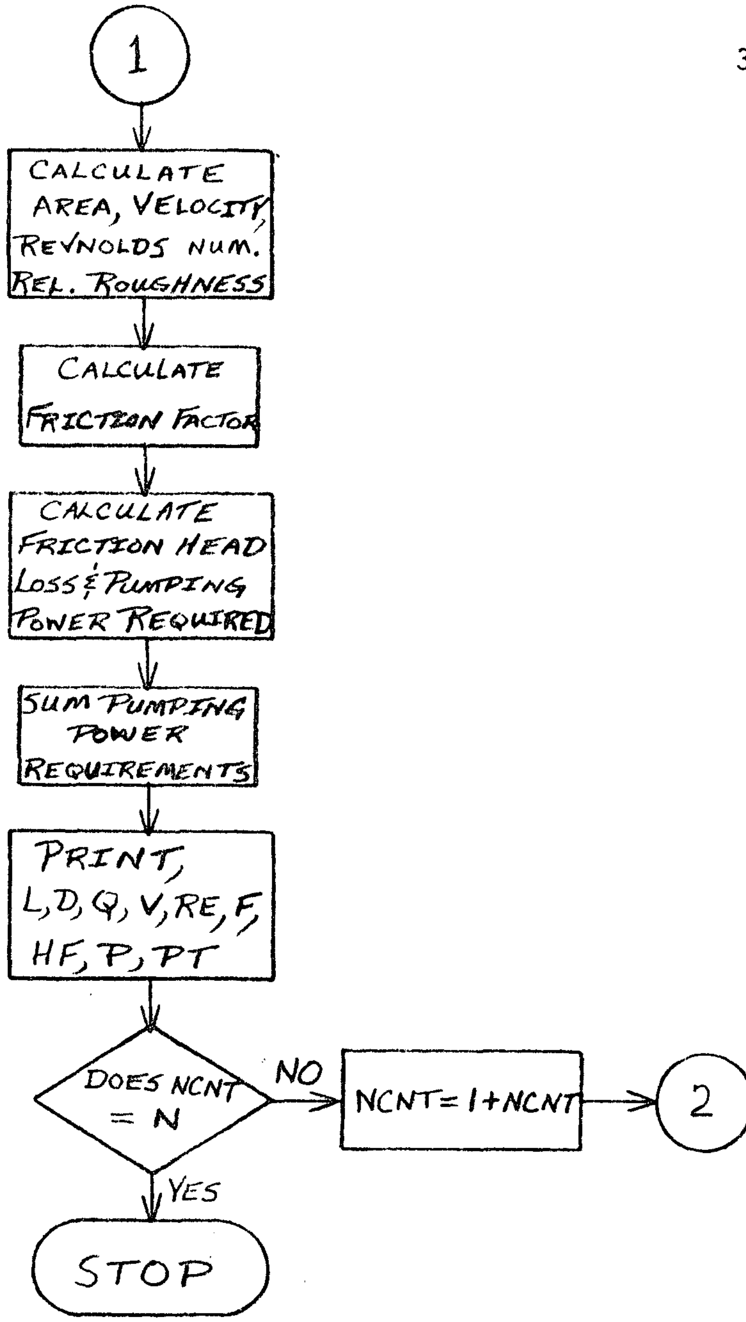


Figure 16. (Continued) Pump Work Computer Flow Diagram, Street Distribution Lines.

Once the flow rate for a length of pipe has been calculated, a subroutine is called. The subroutine calculates first the Reynolds Number and relative roughness and then the friction factor. The Darcy-Weisbach equation is used to find the friction head loss in feet. The pumping power required for the section is found from the friction head loss. These pumping power requirements for each section are summed to obtain the horsepower required to pump the water through the sub-division.

This computer program was applied to the entire community of Idlewild. The conditions used were:

- The water temperature was 135°F
- The thermal load was 18,667 Btu/hr-house
- The consumptive water flow rate was 246 gallons per day per house
- The water flow rate used was four times that required to satisfy selected thermal load (2/3 of design heating load); this would prevent excessive thermal degradation as the fluid moves the length of each street.
- The total community flow rate less that for the 15 houses serviced by the line along Mackenzie Dr. (under the same 4X factor) is used at the beginning section along Parkston Ave. (down to Crofton Dr.). Appropriate reductions in this flow were applied at each street branch take-off. The reverse procedure was followed along Idlewild Ave., with the final section between Crofton Dr. and Ashmore Branch Rd. carrying the total community flow rate, less that for the 15 houses serviced by the line along Mackenzie Dr.

- A comparable procedure was used for the fifteen houses serviced by the line along Mackenzie Dr. A return line (6-inch) was assumed to be installed connecting Melrose Ct. and Picardy Br. supply lines back to the 16-inch line along Ashmore Branch Rd.

The total pumping power for the community under these conditions is 32.3 hp. Since the total community contains 140 houses, the supply/return pumping in the street mains requires approximately 0.231 hp per house, on the average. Assuming a 60 percent pumping efficiency, this pumping power is the equivalent of 5.25% of the thermal energy supplied to each house.

IX. PUMPING AND POWER REQUIREMENTS FOR THE ADDED FLOW SYSTEM FOR A TYPICAL RESIDENCE

To provide a thermal water flow and return at each residence, the conceptual design that we selected consists of separate supply and return lines from the street water main to the residential heat exchanger. To provide the required flow rate and to re-inject the thermally used water into the street main, a magnetic drive, sealed pump would be installed inside each house, just downstream of the heat exchanger.

The configuration that we considered is for a house set-back of 75 feet from the street main and approximately 25 feet of supply line from the house foundation to the heat exchanger use point. The piping system includes

- 200 linear feet of tubing
- one pump
- one gate value (in supply line)
- one check valve (in return line)

- 5 90° bends
- the heat exchanger which has a variable equivalent length of 1.43 ft, 2.61 ft or 6.68 ft depending upon the flow rate
- one throttling flow control valve which is the equivalent of a gate valve.

The system is depicted schematically in Figure 17.

The resulting head loss in the piping was calculated for the three hot supply temperatures, 135°F, 125°F, 115°F, and for copper tubing sizes of 3/4", 1/2", 3/8", and 1/4". The head loss in the tubing and fittings was added to the heat exchanger head loss to obtain the overall friction head loss in the house supply system. These losses were used in conjunction with the flow rates to obtain the power required to pump the water back into the main distribution system. A table of these values is shown below:

Table 1. Pumping Power for House Piping. Multiply Values by 10^{-3} to Obtain Horsepower.

Pipe Size	3/4"	1/2"	3/8"	1/4"
35° ΔT	0.662	1.9	5.2	18.0
25° ΔT	1.71	4.7	13.5	44.8
15° ΔT	7.18	19.6	55.4	190.0

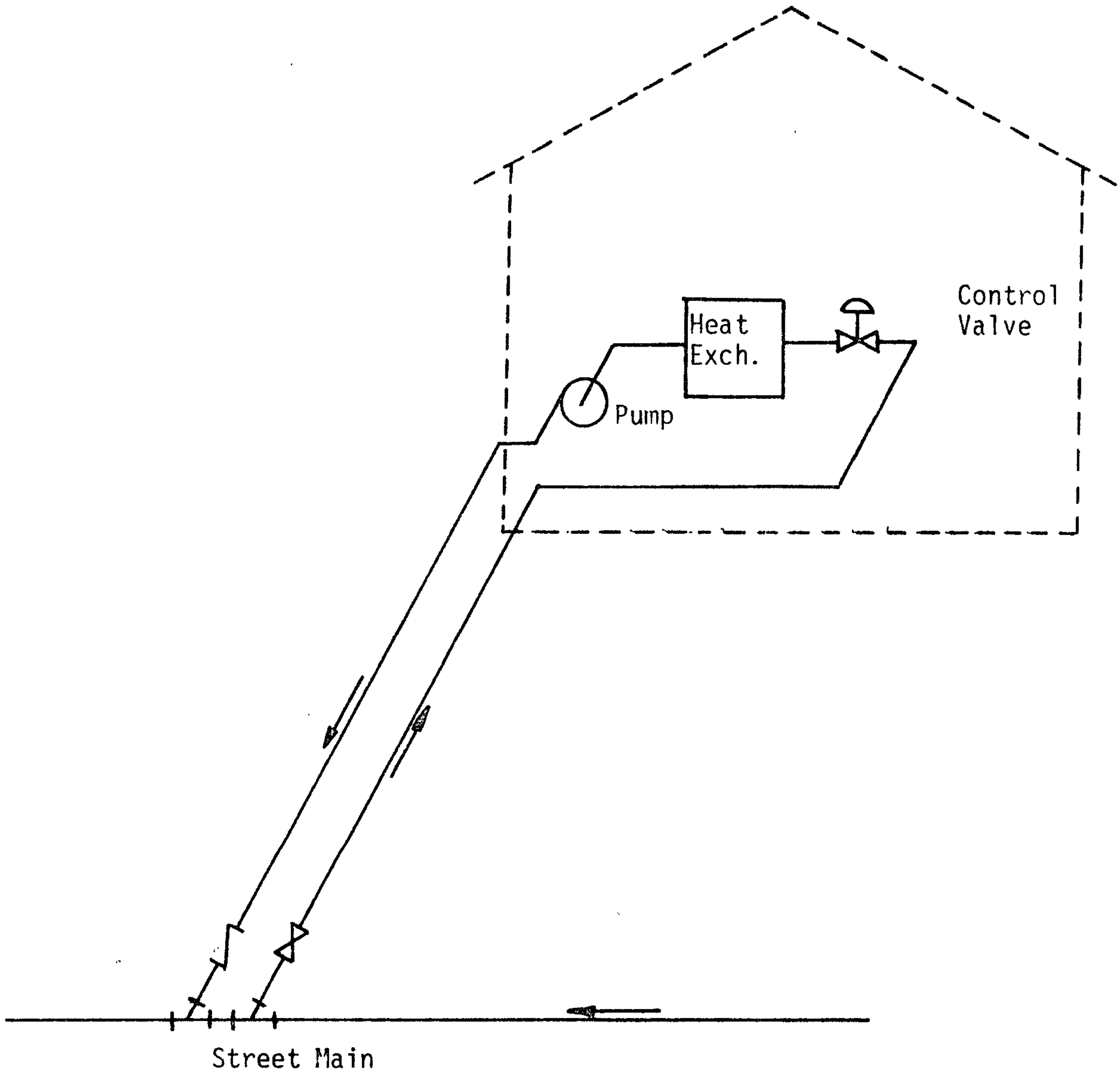


Figure 17. Schematic of Individual Residential Piping System.

APPENDIX

LIMITS ESTABLISHED ACCORDING TO U. S. ENVIRONMENTAL PROTECTION AGENCY INTERIM PRIMARY DRINKING WATER STANDARDS

Total Solids	500 ppm
Turbidity	5 t.u.
Color	15 c.u.
Alkalinity	No est. limits
Calcium	No est. limits
Magnesium	No est. limits
Hardness	No est. limits (0-50 ppm Soft) (51-100 ppm Med.-Hard) (101-200 ppm Hard)
Sodium	No est. limits
Iron	0.3 ppm
Chlorides	250 ppm
pH	6.5-8.5 pH units acceptable
Manganese	0.05 ppm
Copper	1.0 ppm
Zinc	5.0 ppm
*Fluoride	Controlled (0.7-1.0 ppm) Natural (1.6 ppm)
Potassium	No est. limits.
*Mercury	2.0 ppb
*Chromium	0.05 ppm
*Cadmium	0.01 ppm
*Lead	0.05 ppm
*Arsenic	0.05 ppm
*Barium	1.0 ppm
Cyanide	0.2 ppm
*Selenium	0.01 ppm
*Silver	0.05 ppm
MBAS	0.5 ppm
CCE	0.2 ppm
*Nitrate	10 ppm
Nitrite	1 ppm
Phosphate	No est. limits
Sulfate	250 ppm
Odor	3 t.o.

PESTICIDES

Hydrocarbons:	
Aldrin	0.001 ppm
Chlordane	0.003 ppm
DDT	0.05 ppm
Dieldrin	0.001 ppm
* Endrin	0.0002 ppm
Heptachlor	0.0001 ppm
Heptachlor Epoxide	0.0001 ppm
*Lindane	0.0004 ppm
*Methoxychlor	0.10 ppm
*Toxaphene	0.005 ppm
Organophosphate & Carbamate:	0.1 ppm (parathion)

HERBICIDES

*2,4-D	0.1 ppm
2,4,5,T	0.002 ppm
*2,4,5 TP (Silvex)	0.01

ppm = parts per million
ppb = parts per billion
t.u. = turbidity units
c.u. = color units
t.o. = threshold odor

*Indicates maximum contaminant levels. All other parameters established by the U. S. Public Health Service, 1973.

DWS
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$JOB           P=20
      DIMENSION  L(100), H(100), D(100), B(100),QPRT(100)
      REAL MU,L,NH
      INTEGER B,H
      READ, RHO,MU,E,QD,QH,NH
      READ,N,(L(I),H(I),D(I),B(I),I=1,N)
      Q= NH*(QD + QH)
      PRINT 70
   70 FORMAT('1',//,12X,'N',15X,'L',7X,'D',7X,'Q',8X,'V',10X,'RE',8X,'F
     $,8X,'HF',7X,'P',10X,'PT')
      PT = 0.0
      DO 15 I=1,N
C  B=0 PROGRAM ITERATES ALONG SELECTED PIPE SEGMENT
C  B=1 PROGRAM CALCULATES FLOW IN BRANCH  PIPE LINE
C  B=-1 PROGRAM ITERATES ALONG MAIN PIPE LINE LEAVIVING SUB-DIVISION
      IF(B(I)) 1,2,3
    2 Q=Q-H(I)*QD
      CALL FRCTNF(Q,RHO,MU,E,D,V,F,RE,HF,P,PT,L,I)
      QPRT(I)=Q
      GO TO 14
C  B=2 PROGRAM ITERATES  ALONG BRANCH PIPE LINE
    3 IF(B(I).EQ.2) GO TO 4
C  B=3 PROGRAM CALCULATES FLOW IN SECONDARY BRANCH PIPE LINE
      IF(B(I).EQ.3) GO TO 5
C  B=4 PROGRAM ITERATES ALONG SECONDARY BRANCH PIPE LINE
      IF(B(I).EQ.4) GO TO 6
C  B=5 PROGRAM CALCULATES FLOW IN RIGHT-HAND BRANCH FROM MAIN PIPE LINE
      IF(B(I).EQ.5) GO TO 11
C  B=6 PROGRAM ITERATES ALONG RIGHT-HAND BRANCH
      IF(B(I).EQ.6) GO TO 12
      READ, IREQ,HT
      QE= Q-HT*QD
      QB=IREQ*(QD+QH)
      Q=Q-QB
    4 QB= QB-H(I)*QD
      CALL FRCTNF(QB,RHO,MU,E,D,V,F,RE,HF,P,PT,L,I)
      QPRT(I)=QB
      GO TO 14
C  B=-2 PROGRAM CALCULATES FLOW IN MAIN PIPE LINE LEAVING SECONDARY
C  BRANCH
    1 IF(B(I).EQ.-2) GO TO 7
C  B=-3 PROGRAM ITERATES ALONG MAIN PIPE LINE LEAVING SECONDARY BRANCH
      IF(B(I).EQ.-3) GO TO 8
C  B=-4 PROGRAM CALCULATES FLOW IN LEFT-HAND BRANCH  FROM MAIN PIPE LIN
      IF(B(I).EQ.-4) GO TO 9
C  B=-5 PROGRAM ITERATES ALONG LEFT-HAND BRANCH
      IF(B(I).EQ.-5) GO TO 10
C  B=-6 PROGRAM ADDS FLOW FROM BRANCH BACK INTO 16 INCH LINE
      IF(B(I).EQ.-6) GO TO 13
      QE= QE-H(I)*QD
      CALL FRCTNF(QE,RHO,MU,E,D,V,F,RE,HF,P,PT,L,I)
      QPRT(I)=QE
      GO TO 14
    5 READ, IREQ
      QBB=IREQ*(QD+QH)
      QB= QB-QBB
    6 QBB=QBB-H(I)*QD
      CALL FRCTNF(QBB,RHO,MU,E,D,V,F,RE,HF,P,PT,L,I)
      QPRT(I)=QBB
      GO TO 14
    7 QEB=QBB+(Q-HT*QD)
    8 QEB =QEB-H(I)*QD
      CALL FRCTNF(QEB,RHO,MU,E,D,V,F,RE,HF,P,PT,L,I)
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```
      QPRT(I)=QEB
      GO TO 14
    9 READ,HL,HR
      QL=HL*(QD+QH)
      QL=HL*(QD+QH)
   10 QL=QL-H(I)*QD
      CALL FRCTNF(QL,RHO,MU,E,D,V,F,RE,HF,P,PT,L,I)
      QPRT(I)=QL
      GO TO 14
   11 QR=QL+HR*(QD+QH)
   12 QR=QR-H(I)*QD
      CALL FRCTNF(QR,RHO,MU,E,D,V,F,RE,HF,P,PT,L,I)
      QPRT(I)=QR
      GO TO 14
   13 Q=Q+QB
      CALL FRCTNF(Q,RHO,MU,E,D,V,F,RE,HF,P,PT,L,I)
      QPRT(I)=Q
   14 PRINT 30,I,L(I),D(I),QPRT(I),V,RE,F,HF,P,PT
   30 FORMAT('0',10X,I2,10X,2F8.2,2F9.4,F12.2,F9.4,F8.2,F10.5,F12.5)
   15 CONTINUE
      STOP
      END

      SUBROUTINE FRCTNF(Q,RHO,MU,E,D,V,F,RE,HF,P,PT,L,I)
      DIMENSION L(100), D(100)
      REAL MU,L
      DFT = D(I)/12.0
      AREA = (3.1415926*DFT**2.)/4.0
      V = Q/AREA
      RE = (RHO*V* DFT)/MU
      RR = E/DFT
      F= 0.019
    2 A= 1./SQRT(F)
      B = -0.86*ALOG(RR/3.7 + 2.51/(RE*SQRT(F)))
      IF(ABS(A-B).LE. 0.05  ) GO TO 1
      F = F+ 0.0001
      GO TO 2
    1 HF = F*(L(I)/DFT)*(V**2./64.4)
      P = (Q*RHO*HF)/550.0
      PT= PT+P
      RETURN
      END

$ENTRY
```