An economic study

of a thermal conveyance system

as applied to waste heat utilization

by

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INTRODUCTION

Waste heat is energy that has been discharged to the environment on the assumption that it is no longer useful for the process under consideration. Generally, this waste heat is discharged in the form of condenser water with temperatures ranging from $90^{\circ}F$ ($32.2^{\circ}C$) to $110^{\circ}F$ ($43.3^{\circ}C$).

Largest and most widely distributed sources of waste heat are steam electric power plants. Energy stored in nuclear or fossil fuel is converted to electricity with efficiencies ranging from thirty to forty percent. Thus, for every three units of energy consumed only one appears as electricity and the other two are discarded as waste heat. According to Karkheck and Powell (1) today's electric generating plants produce waste heat in amounts exceeding the total U.S. space and water heat net demand. It must be noted that due to its low temperature the waste heat could not practically supply all the space and water heat demand, however, because of the vast amount available it could be utilized to provide some of the demand for energy.

District heating, based on utilizing heat currently wasted, is a potential partial solution to the problem of finding energy to meet projected U.S. demands. Heat energy required for this application is provided at temperatures higher than the normal outlet temperature of the plant's condenser water (2). Heat at this temperature is called low grade heat, (100^oF (37.8^oC) to 400^oF (204.4^oC)), and is extracted as steam from the last stages of the turbine. Removal of this low grade heat from the turbine reduces the power plant's electrical generating capacity and hence there is a trade-off between the advantage of supplying the high temperature heat required for specific applications and the cost penalty associated with lower plant capacity. It has been shown that, through the use of multipurpose central generating stations which are a source of electricity, low grade heat, and waste heat, overall efficiencies of between 65 and 70 percent can be obtained (3).

It is unlikely that power plant thermal efficiencies of 30 to 40 percent can be improved significantly, unless higher operating temperatures are made possible through major breakthroughs in the metallurgical field. Therefore, higher energy-use efficiency in the electric power industry centers upon the application of waste heat in various integrated energy use systems.

Literature Review

Generally speaking, energy is produced conveniently or economically at some distance from the place where it can be used. The task confronting the engineer is to transport the energy from where it is to where it is needed.

In this study the energy source is water heated to some temperature above the ambient, usually by a power plant. It must be conveyed in some manner to customers located at some distance from the source. This can be done indirectly, such as by converting the energy produced as heated water at the plant to electricity and conveying it over power lines to the customer. If necessary, the electrical energy can be turned back into heat at an appropriate temperature. The energy can also be conveyed directly by transporting the heated water through pipelines to the customer. This method of energy conveyance is studied in this investigation.

Examples of pipelines conveyance of heat can be found all over the world. In 1957, Bulgaria began using waste heat from industry and electric generating stations to heat greenhouses [4] . Nearly all Danish towns have district heating pipe networks and in Sweden considerable effort has gone into the design of heat transportation and distribution systems using waste heat from nuclear reactors [4] . Also, large district heating systems have been operated in cities like New York, Detroit, Chicago, Pittsburgh, and Rochester, New York [5].

Oak Ridge National Laboratory has studied the technical and economic feasibility of district heating in the Minneapolis-St. Paul area [6]. As of 1962, 54 U.S. cities had district heating systems [7].

Different applications require heat at different temperatures. The water or steam used in district heating is usually delivered at a temperature of 300^{O}F. Certain chemical operations require heat at higher temperatures, whereas greenhouse heating needs can be satisfied by water heated to a temperature 10 to 20 degrees above the normal growing temperature. Due to this temperature variation, there is a difference in the piping systems used for district heating from that used for waste heat applications. A district heating transport system may cost more because of the precautions which must be taken to maintain the water at a higher temperature. This requirement might seem to imply that the transport system for district heating is more important than that for the lower temperature waste heat transport system. Actually, since the low temperature characteristics of a waste heat utilization system requires that much more water must be conveyed to deliver the required energy, the piping system will be larger and more expensive. Therefore, the piping system may contribute significantly to the cost of both district heating and waste heat utilization systems.

The cost analysis of these systems is vital. It was shown in the ORNL study that the transportation and distribution costs were 66 percent of the total investment [6]. Also, when in 1964 the Bulgarian experiment on greenhouse heating with warm water from industrial and electric generating stations was discontinued, one of the reasons given was the extensive piping requirements with associated costs [4]. Most underground heat delivery systems feature high installation and maintenance costs [8]. Ellwood Clymer, Jr. noted in this connection that "if combined heat and power usage is to become a significant part of our total energy picture, then one of the goals is to reduce main installation costs" [9]. Thus, the purpose of this research is to explore the cost composition of thermal conveyance systems as applied to waste heat utilization and to reveal how the overall costs can be reduced.

STATEMENT OF THE PROBLEM

Once the technical practicality of a waste heat utilization project has been established, the economic aspects of the problem must be investigated. Potential hidden costs in terms of both energy and dollars are important.

Guidelines and Utilization Costs

First of all there must be a set of basic guidelines to help define the problem and direct the way to a possible solution.

Guidelines

(1) No major plant modifications or design changes should result from the use of waste heat.

(2) If the operational efficiency of the power plant is reduced, the cost will be borne by the heating customers.

(3) Total cost of the waste heat delivered to the user should be competitive with alternative heat sources.

(4) The power plant operator can not be held responsible for outages, consequently the waste heat customer must provide his own standby capacity (8).

With these guidelines in mind, a practical waste heat utilization system must be economically attractive to investors.

Waste heat utilization costs can be categorized into three major groups:

A) Energy costs

B) Utilization costs

C) Transportation and distribution costs

Energy Costs   The cost of the energy, in a waste heat system, will depend on the temperature that is requested by the user. In the case of waste heat being used, for example, to improve fish yields in water warmed by power plant condenser coolant(10,11), the cost could be zero as long as the power plant operation is normal with no decrease in generating capacity. Higher temperature heat would be priced to compensate the utility for the loss of revenue created by lower electrical output. This area has been explored previously(12) and will not be considered in this investigation.

Utilization Costs   Utilization costs will be made up of costs for heat exchangers, heat pumps, valves, and controls required for waste heat utilization. The cost of this equipment can be expensive because of the large volume of low temperature fluid used. This area has also been studied (6,12,13) and will not be discussed in this study.

Cost comparisons that would be initiated by the third guideline have been conducted by various people(6,12) and therefore, will not be part of this paper.

Transportation and Distribution Costs   Pumps, pipes, installation costs, and the cost of running and maintaining the system are what make up the transportation and distribution costs. The importance of this area to the overall economic success of a waste heat utilization system must be addressed and therefore will be the main theme of this study.

OBJECTIVE

Transportation and distribution costs are the largest fraction of the total cost of a waste heat utilization system. These costs have been shown to be as high as 66 percent of the total investment (6). Therefore, the objective of this research is to establish the most economical thermal conveyance system as applied to waste heat usage.

To accomplish this the capital costs, such as pipes and pumps, will be investigated, but more importantly the long term costs, such as operating costs, will also be studied.

Reasons for pipes and pumps being important in this water supply system is that in a waste heat environment the usable temperature difference is so low that a large volume of water must be delivered to the customer to supply the heat demand and therefore the supply system becomes large and important. Also, exploring operating costs over a long period of time is necessary in an economic study because of the effect it may have on the outcome of the study to determine which concept is the best.

If the use of waste heat is to play a role in helping to alleviate the world's energy crisis, then an economical thermal conveyance system must be developed. Then, when the associated conveyance system cost, which is a large fraction of the total, is reduced, the idea of waste heat utilization may be viable.

ANALYSIS OF THERMAL CONVEYANCE SYSTEMS

System Description

For this study of waste heat utilization, the type of system that will be analyzed is shown in Figure 1. Also, the system parameters held constant and those that vary are listed below. Flowrate and the coolant temperature of the plant will be held constant at 2000 gal/min and 100°F (37.7°C) respectively. The pipe diameter will change so that the trade off between high head pumps and small diameter pipes and low head pumps and large diameter pipes can be investigated.

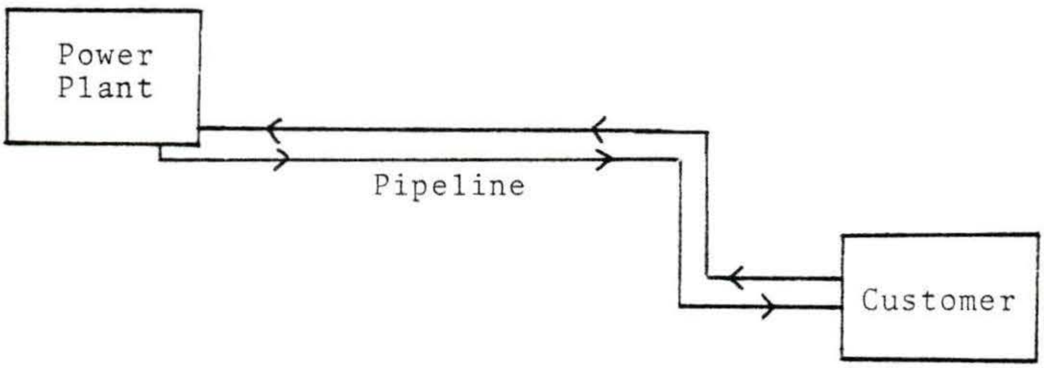

Figure 1. Conveyance System

Systems Parameters

A. Fixed

1. Flow in gallons/minute
2. Coolant temperature at plant

B. Variable

1. Pipe diameter
2. Pipe material
3. Distance between power plant and customer
4. Pipe above or below ground

Pressure Losses

In a particular application the pump chosen must provide the necessary flow through the pipe and therefore develop the required pressure head. Thus, before selecting a combination of pipe and pumps, it is necessary to calculate the pressure drop through the system. This calculation will determine the head which must be developed by the pump to distribute the water to the customers.

The pressure drop is found by using the equation (14):

$$P_f = f \frac{L}{D} \frac{\rho v^2}{2 g_c}$$

where P_f = pressure loss - lb_f/ft^2

f = friction factor - dimensionless

L	=	length of pipe - feet
D	=	diameter of pipe - feet
ρ	=	density of the fluid - lb_m/ft^3
v	=	velocity of the fluid - ft/sec
g_c	=	gravitational constant - 32.2 $lb_m\ ft/lb_f\ sec^2$

The friction factor f depends on pipe inside surface characteristics and the Reynolds number, which is defined as (14):

$$Re = \frac{\rho v D}{\mu}$$

where

Re	=	Reynolds number - dimensionless
ρ	=	density of the fluid - lb_m/ft^3
v	=	velocity of the fluid - ft/sec
μ	=	viscosity of the fluid - lb_m/ft sec
D	=	inside diameter of the pipe - feet

Table 1. Reynolds Numbers

Diameter	Velocity	Re #	Type of Flow
8"	12.8 ft/sec	1.15×10^6	Turbulent
10"	8.16 "	9.21×10^5	Turbulent
12"	5.68 "	7.69×10^5	Turbulent
14"	4.17 "	6.58×10^5	Turbulent
16"	3.19 "	5.76×10^5	Turbulent

A Reynolds number less than 2300 indicates laminar flow, whereas a Reynolds number greater than 4000 is characteristic of turbulent flow. As can be seen from Table 1, the flow is turbulent in every case.

Now that the Reynolds number is known, the friction factor can be found and the pressure losses calculated. Values in Table 2 are taken from Cameron Hydraulic Data (15) in which the results of the above calculations are tabulated. To obtain the head loss used in determining the pump size, a representative value was added to account for the head losses in bends, valves, and flow area changes in the pipe system. In the actual case this head loss may be greater than or less than the assumed value, but the pump selected will be able to handle the difference (See Appendix B). Two pipe materials were considered, polyvinyl chloride (PVC) and ductile cast iron (DCI) and the distance is the total circuit distance to and from the customer.

Pump and Pipe Selection

Pump selection depends on the total head loss, the corresponding flow, and the pressure range at which the pump operates most efficiently. With this information available, a suitable pump can be identified with the aid of a supplier's catalogue. For example, a 10x12x20 is a pump with a 10 inch inlet, a 12 inch outlet, and a maximum impeller size of 20 inches.

Table 2. Total Head Loss

Distance (miles)	Diameter (inches)	Velocity (ft/sec)	PVC Head Loss (ft)	DCI Head Loss (ft)
	8	12.80	282	285
1/2	10	8.16	95	96
	12	5.68	39	40
	14	4.17	18	18
	16	3.19	9	9
	10	8.16	190	192
1	12	5.68	78	79
	14	4.17	34	35
	16	3.19	19	20
	10	8.16	381	384
2	12	5.68	152	153
	14	4.17	71	72
	16	3.19	38	39

The pump for this type of application must develop a high head, be compatible with water (no corrosion), be inexpensive, and have an efficiency of at least eighty percent. The best choice is a cast iron horizontal-split case pump with bronze fittings.

Table 3 shows the cost data for different sized pumps needed for various pipe diameters and different system lengths.

Table 3. Pump Costs

Distance (miles)	Diameter (inches)	Head Loss (ft)	Pump Size (inches)	Motor Size (hp)	Total Costs ($)
	8	290	10x8x20	200	13,000
1/2	10	100	10x8x20	75	10,000
	12	41	10x8x20	40	9,000
	14	21	12x10x12	20	7,000
	10	200	8x6x17	150	9,000
1	12	82	8x6x14½	75	6,000
	14	38	10x8x12	25	5,000
	16	25	10x8x12	20	5,000
	10	390	10x8x20	300	14,000
2	12	155	10x8x17	125	10,000
	14	74	10x8x17	50	8,000
	16	45	10x8x17	40	8,000

Selection of the most economical pipe depends on unit price and the desired corrosion resistance to both the fluid flowing inside the pipe and the environment outside. Polyvinyl chloride (PVC) and ductile cast iron pipe (DCI) were chosen for this study over steel, stainless steel and aluminum because they best fit these criteria, with the price being the determining factor. Both types of pipe are rated at pressures higher than that to be expected at the outlet of the pumps. The pipe specifications can be seen in Appendix A.

Pipe costs make up the largest part of the initial capital costs of the thermal conveyance system. Table 4 compares the prices of both PVC and DCI pipe.

The costs of various pump and pipe combinations can be seen in Figure 2. Data presented in Tables 3 and 4 are combined and plotted as a function of piping system length. For both pipe materials, identical pumps were used since the pump characteristics cover the range of required heads and flows needed for both types of pipe.

It can be seen from the graph that the smallest diameter and least expensive ductile cast iron pipe has the lowest combined cost. Therefore, the pipe is the determining factor as was noted previously. However, the inclusion of installation, maintenance, and operating costs will alter the outcome.

Table 4. Pipe Costs

	PVC			DCI		
Distance (miles)	Diameter (inches)	Cost/ft (dollars)	Total Cost (dollars)	Diameter (inches)	Cost/ft (dollars)	Total Cost (dollars)
1/2	8	12.32	36,000	8	8.06	21,000
	10	21.11	56,000	10	10.61	28,000
	12	29.41	78,000	12	13.43	36,000
1	10	21.11	112,000	10	10.61	56,000
	12	29.41	155,000	12	13.43	71,000
				14	16.29	86,000
				16	19.01	100,000
2	10	21.11	223,000	10	10.61	112,000
	12	29.41	311,000	12	13.43	142,000
				14	16.29	172,000
				16	19.01	201,000

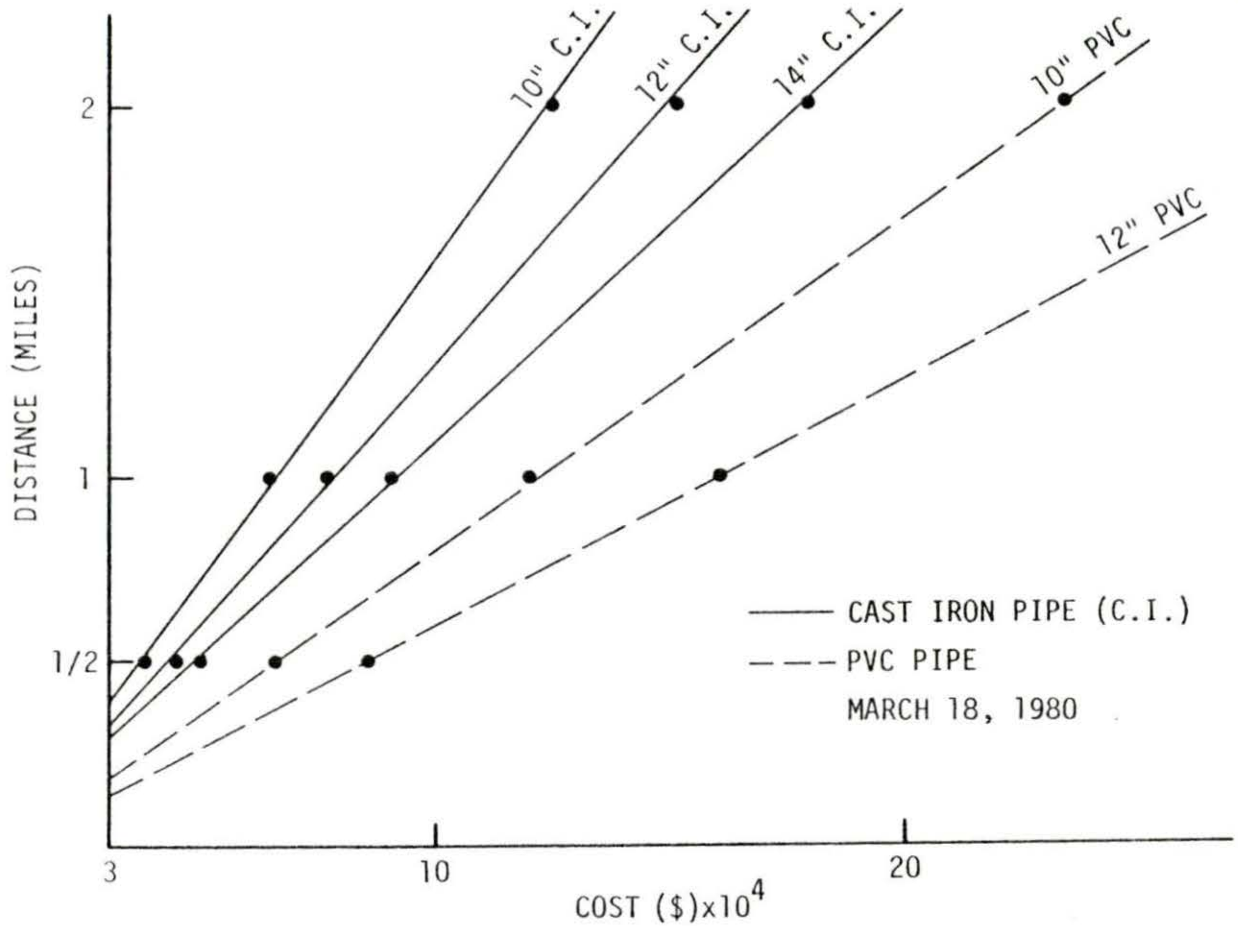

Figure 2. Pump and Pipe Costs

Heat Losses

A thermal conveyance system's purpose is to supply water to the customer at some selected temperature. Therefore, the heat loss to the environment from the system must be considered so that the appropriate inlet temperature can be selected. To determine the best system, both in terms of cost and heat losses, different approaches were studied.

Because the water is being pumped around a circuit, the heat gained by the water from the pump must be included. The equation used to calculate the contribution is given as:

$$Q = \frac{(V)(h)}{778}$$

where Q = heat gained - BTU/hr

V = volumetric flowrate - ft^3/hr

h = head generated - lb_f/ft^2

The temperature changes in the water due to the heat gained from pumping are so small ($\sim .3^{\circ}F$) in comparison to the initial temperature of the water ($100^{\circ}F$) that this quantity can be neglected.

Heat losses corresponding to four different distribution systems were investigated. The systems were insulated and uninsulated aboveground systems and insulated and uninsulated buried systems. The heat transfer analysis for the aboveground cases is different from that for the buried cases as is evident from the following equations. For the aboveground systems the heat losses are calculated using the following equation (12):

$$Q/L = \frac{Tw - Ta}{\frac{1}{\pi D h_w} + \frac{\ln\left[(R+c)/R\right]}{2\pi K_p} + \frac{\ln\left[(R+c+d)/(R+c)\right]}{2\pi K_{in}} + \frac{1}{\pi (OD) h_a}}$$

where Q/L = heat loss per unit pipe length - BTU/hr ft

T_w = temperature of water - ^oF

T_a = temperature of air - ^oF

D = inside diameter of pipe - feet

OD = outside diameter of pipe - feet

h_w = heat transfer coefficient of water - BTU/hr ft^2 ^oF

h_a = heat transfer coefficient of air - BTU/hr ft^2 ^oF

R = inside radius of pipe - feet

c = thickness of pipe - feet

d = thickness of insulation - feet

K_p = thermal conductivity of pipe - BTU/hr ft ^oF

K_{in} = thermal conductivity of insulation - BTU/hr ft ^oF

In both underground examples, insulated and uninsulated, the heat transfer mode is heat conduction in a semi-infinite body. The equation for calculating the heat losses is (17):

Q/L =

$$\frac{T_w - T_s}{\frac{1}{\pi D h_w} + \frac{\ln\left[(R+c)/R\right]}{2\pi K_p} + \frac{\ln\left[(R+c+d)/(R+c)\right]}{2\pi K_{in}} + \frac{\ln\left[4N/(OD)\right]}{2\pi K_e}}$$

where T_s = temperature at the surface - ^{o}F

N = depth of pipe centerline below the surface - feet

K_e = thermal conductivity of the earth - BTU/hr ft ^{o}F.

The above method used to calculate the heat losses is a simple one. It is a steady state application of conduction and convection. Values obtained for the heat loss are the maximum values that can be calculated. But, because the temperature of the fluid is so low this method approximates the heat loss very well. For a more detailed analysis the reader may refer to [18].

In the underground cases the thermal conductivity of the pipe is so large, compared to the thermal conductivity of earth (Ke), that its effect is negligible. This makes the value of the thermal conductivity of the earth the dominant factor in the heat loss equation. Therefore, the value for Ke was taken as 1.0 BTU/hr ft^{o}F. This value was used by Kendrick and Havens [18] as the steady state thermal conductivity of dry soil. Other values have been calculated [19] , but this value will be used in the heat loss calculations.

In each case, a 12 inch pipe carrying 100^{o}F water will be used as the reference case. For the insulated examples, 1 1/2 inches of urethane foam, thermal conductivity of .019 BTU/hr ft^{o}F, surrounds the pipe and for the buried cases the pipe will be buried six feet deep to ensure that it lies below the frostline.

Heat loss for the aboveground pipe system involves both conduction and forced convection. Heat conduction through the pipe wall and insulation is a straightforward analysis, but the calculations of the forced convection heat transfer from the pipe to the ambient air and from the water to the pipe are more involved.

The forced convection heat transfer coefficient depends on the Reynolds number and the average Nusselt number. In the case of the aboveground system there are two heat transfer coefficients that need to be evaluated, the one between the water and the inside of the pipe, h_w, and the other between the exterior of the pipe and the outside air, h_a. For the underground analysis the only heat transfer coefficient that is needed is h_w, and it is found in the same manner as in the aboveground case.

The connection between the forced convection heat transfer coefficient, h, is made through correlations of the form

$$Nu = K\ Re^n\ Pr^m$$

where

$$Nu = \text{Nussett number}\ \frac{hD}{k}$$

$$Re = \text{Reynolds number}\ \frac{\rho VD}{\mu}$$

$$Pr = \text{Prandtl number}\ \frac{c_p \mu}{k}$$

K, n and m are well established, empirically determined constants which correspond to specific applications such as whether the flow is inside or outside the pipes; if outside whether the flow is parallel to, or perpendicular to, the pipe axis and whether the pipes are held at a constant temperature or conduct a constant heat flux.

For the case of interest in this study, with flow inside pipes and a constant wall temperature, the Dittus and Boelter correlation (12) gives K = 0.023, n = 0.8 and m = 1/3. The value of h_w was found to be 673.0 BTU/ft^2 hr ^oF in all cases.

For the case of air flow over the outer surface of the pipes the correlation to be used depends on the Reynolds number. Typical speed and air temperature data (12) as presented in Appendix C were used to calculate the Reynolds numbers shown in Table 5. The appropriate correlation for the range of Reynolds numbers is (16)

$$Nu = 0.0266\ Re^{0.805}\ Pr^{1/3}$$

For the aboveground cases the Nusselt numbers and the heat transfer coefficients for the outside air are presented in Table 5. The values for the heat losses in both aboveground cases are shown in Tables 6 and 7.

Values obtained for the heat losses in both underground systems can be found in Tables 8 and 9.

Table 5. Heat Transfer Coefficients for Outside Air

Month	Re	Nu	h_a	
January	144887	339.1	4.14	BTU/hr ft^2 ^oF
February	146893	342.8	4.24	"
March	162129	271.1	4.60	"
April	155795	359.0	4.51	"
May	135518	320.0	4.05	"
June	123049	296.8	3.78	"
July	111692	274.5	3.51	"
August	92287	235.4	3.00	"
September	108741	268.8	3.41	"
October	122255	295.5	3.70	"
November	147399	343.7	4.24	"
December	150459	349.6	4.27	"
AVERAGE	132886	316.0	3.95	"

Table 6. Heat Loss from Uninsulated Aboveground Pipe

Month	PVC	DCI
January	2835 BTU/hr ft	12085 BUT/hr ft
February	2409 "	10567 "
March	2256 "	10448 "
April	1687 "	7709 "
May	1464 "	6225 "
June	1084 "	4405 "
July	876 "	3391 "
August	955 "	3334 "
September	1239 "	4703 "
October	1861 "	7453 "
November	2482 "	10883 "
December	2910 "	12824 "
AVERAGE	1828 "	7645 "

Table 7. Heat Loss From Insulated Aboveground Pipe

Month	PVC		DCI	
January	43.6	BTU/hr ft	43.4	BTU/hr ft
February	37.3	"	37.2	"
March	34.4	"	34.3	"
April	25.8	"	25.7	"
May	22.9	"	27.7	"
June	17.2	"	17.1	"
July	14.1	"	14.0	"
August	15.9	"	15.9	"
September	20.1	"	20.0	"
October	29.6	"	29.5	"
November	38.4	"	38.3	"
December	45.0	"	44.8	"
AVERAGE	28.7	"	28.6	"

Table 8. Heat Loss From Buried Insulated Pipe

Month	PVC	DCI
January	35.6 BTU/hr ft	34.3 BTU/hr ft
February	30.5 "	29.4 "
March	28.2 "	27.1 "
April	21.1 "	20.4 "
May	18.7 "	18.0 "
June	17.1 "	13.5 "
July	11.5 "	11.1 "
August	13.0 "	12.6 "
September	16.4 "	15.2 "
October	24.2 "	23.3 "
November	31.4 "	30.3 "
December	36.8 "	35.5 "
AVERAGE	23.5 "	22.6 "

Table 9. Heat Loss From Buried Uninsulated Pipe

Month	PVC		DCI	
January	155	BTU/hr ft	161	BTU/hr ft
February	133	"	138	"
March	123	"	128	"
April	92	"	96	"
May	82	"	85	"
June	61	"	64	"
July	50	"	52	"
August	57	"	59	"
September	72	"	74	"
October	106	"	110	"
November	137	"	142	"
December	161	"	167	"
AVERAGE	103	"	106	"

To get a better idea of what these heat losses mean, Table 10 contains data for the heat losses in terms of a water temperature drop in degrees Fahrenheit for a two mile circuit. Since December had the highest heat loss rate, the temperature drop calculations will be made for that month only. From the data in the table, the most economical installation method can be determined.

In the PVC and DCI uninsulated-aboveground-case the coolant temperature drop is $16^{\circ}F$ and $68^{\circ}F$ respectively. To make up this temperature loss the inlet coolant temperature would have to be raised which lowers the electrical generating efficiency and requires that a high charge be made for the waste heat. This cost, calculated over an operating period of 1.15×10^5 hours, is more than the cost of burying the uninsulated pipe. Also, if there were any outages in the winter months, to prevent freezing the water would have to be drained from the pipes placed aboveground. Thus, the pipes must be buried below the frost line for service in the Midwest.

From the table it can be seen that the advantages of insulating pipe to be buried are negligible, consequently the appropriate technique will be to use an uninsulated buried pipe to carry the coolant from the power plant to the user.

Installation Costs

To install an underground piping system, a ditch must be dug, the pipe put in and assembled, and the ditch backfilled. The costs associated with these activities are based on data taken from (20) and are compared in Tables 11 and 12 for the different systems.

Table 10. Water Temperature Drops

	PVC	DCI
Uninsulated aboveground	15.5 °F	68.3 °F
Insulated aboveground	.24 °F	.24 °F
Insulated buried	.20 °F	.19 °F
Uninsulated buried	.86 °F	.89 °F

Table 11. Costs for the Installed PVC Pipe System

Distance (miles)	Pipe Size (inches)	Cost of Installation (dollars)
1/2	8	23,000
	10	24,000
	12	27,000
1	10	28,000
	12	55,000
2	10	96,000
	12	110,000

Table 12. Costs for the Installed DCI Pipe System

Distance (miles)	Pipe Size (inches)	Cost of Installation (dollars)
1/2	8	27,000
	10	29,000
	12	32,000
	14	41,000
1	10	58,000
	12	65,000
	14	82,000
	16	87,000
2	10	116,000
	12	129,000
	14	163,000
	16	175,000

The type of trench to be used is shown in Figure 3. A common ditch, where both supply and return pipes are in the same hole, is utilized because of the low temperature of the water being pumped. Damp, sandy loam, typical of much of the Midwest is the soil type assumed in the calculations.

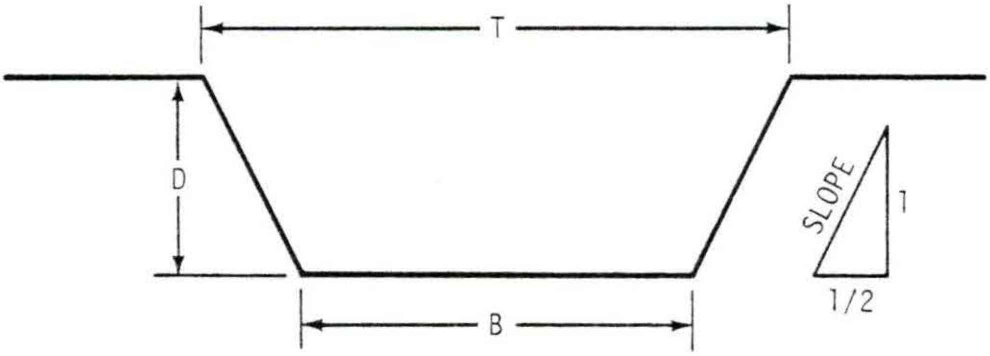

Figure 3. Ditch Configuration

By calculating the amount of earth removed, the cost of digging and backfilling the trench is obtained. The depth is six feet and the width at the bottom of the trench is found by adding both pipe diameters, a minimum of one-half foot between the side slope and the closest pipe, and a minimum of one foot between the pipes. The volume of earth to be removed per foot of trench is given as (20):

$$\text{Cubic yards/lineal foot} = \frac{1/2(B+T) \times D}{27}$$

where B = width at bottom of trench

T = width at top of trench

D = depth of trench

Electrolytic corrosion is a possible concern when burying any type of iron pipe. Cathodic protection is protection against pipe corrosion due to interactions between the pipe and soil acids. The type of ductile cast iron pipe considered in this report withstands this type of corrosion very well (21) and the suppliers recommend that no cathodic protection be provided.

Operating Costs

The importance of operating and maintenance costs on the economic evaluation and comparison of heat transport systems cannot be overestimated. Decisions between possible systems based on equipment costs alone may prove to be wrong when actual operating and maintenance costs are introduced. Meador (22) notes that if these costs are not carefully predicted, then the choice of the pumping system reflects an acceptance of a trial and error process or, at best, an impression of past experience leading usually to an unnecessarily high overall cost.

The costs of operating the pump to transport the water around the circuit can be significant in determining which system is the most economical. Operating costs are based on the head and flow needed for the system. To calculate the number of kilowatts needed, the following equations were used (23):

$$\frac{Q \times 10 \times H}{33{,}000} = \text{W.H.P.}$$

$$\frac{\text{W. H. P.}}{\text{pump efficiency}} = \text{B.H.P.}$$

$$\frac{\text{B.H.P.}}{\text{motor efficiency}} = \text{E.H.P.}$$

$$\text{E.H.P} \times (.746) = \text{input kilowatts}$$

Where	Q	=	flow - gal/min
	H	=	total head - feet
	W.H.P.	=	working horsepower
	B.H.P.	=	brake horsepower
	E.H.P.	=	electrical horsepower

The actual time during which the heat transport system would be expected to be in operation reflects the fact that during at least two summer months no heat would be required and that routine plant maintenance would involve shutdowns lasting an additional two months during the year. Thus 24 hours a day, 8 months per year and a 20 year life totals to 1.15×10^5 hours. The associated operating costs are shown in Table 13.

Maintenance Costs

Maintenance costs may be estimated in various ways. The technique used in this research is to take a fixed percentage, 2.5 percent per year, of the capital investment (24). Table 14 shows the total cost for maintenance over a twenty year period for both types of systems.

Table 13. Operating Costs for Pumping System

Distance (miles)	Pipe Size (inches)	Operating Costs (dollars)
1/2	8	733,000
	10	253,000
	12	104,000
	14	53,000
1	10	506,000
	12	207,000
	14	96,000
	16	63,000
2	10	986,000
	12	392,000
	14	187,000
	16	114,000

Table 14. Maintenance Costs

	PVC		DCI	
Distance (miles)	Diameter (inches)	Maintenance Cost (dollars)	Diameter (inches)	Maintenance Cost (dollars)
1/2	8	34,000	8	31,000
	10	47,000	10	34,000
	12	57,000	12	38,000
			14	45,000
1	10	84,000	10	61,000
	12	108,000	12	71,000
			14	86,000
			16	96,000
2	10	166,000	10	121,000
	12	215,000	12	141,000
			14	172,000
			16	192,000

Gross Costs

In viewing the total costs presented in Tables 15 and 16, a conclusion can be drawn about which system is the most economical. The costs of each system are shown both before and after the operating and maintenance costs were added. As was stated earlier, the operating and maintenance costs are so large that the cost ranking of the systems established on the basis of equipment cost alone has been changed. Ductile cast iron is still favored over polyvinylchloride for the pipes but now it can be seen that large sized pipes provide the most economical system.

Table 15. PVC Gross Costs

Distance (miles)	Diameter (inches)	Cost Before Operating Maintenance (dollars)	Operating and Maintenance Costs (dollars)	Gross Cost (dollars)
1/2	8	69,000	767,000	836,000
	10	95,000	300,000	395,000
	12	114,000	161,000	275,000
1	10	168,000	590,000	758,000
	12	216,000	315,000	531,000
2	10	332,000	1,152,000	1,484,000
	12	431,000	607,000	1,038,000

Table 16. DCI Gross Costs

Distance (miles)	Diameter (inches)	Cost Before Operating & Maintenance (dollars)	Operating and Maintenance (dollars)	Gross Cost (dollars)
1/2	8	61,000	764,000	825,000
	10	67,000	287,000	354,000
	12	77,000	142,000	219,000
	14	90,000	98,000	188,000
1	10	122,000	567,000	689,000
	12	142,000	278,000	420,000
	14	172,000	182,000	354,000
	16	192,000	159,000	351,000
2	10	241,000	1,107,000	1,348,000
	12	281,000	533,000	814,000
	14	343,000	359,000	702,000
	16	383,000	306,000	689,000

CONCLUSIONS

The objective of this research was to establish the most economical thermal conveyance system. By doing this the largest fraction of the total cost of a waste heat utilization system will be reduced and therefore help waste heat utilization become economically feasible.

In this study, the most economical thermal conveyance system featured ductile cast iron pipes of 14-inch diameter for a flow distance of one-half mile and 16-inch diameter for flow distances of one and two miles.

Comparing this thermal conveyance system to other studies that have been done (8,22), it can be seen that this system is less expensive. The reason is that the temperature of the water in the other studies is higher (300°F) than the temperature of the water used in the study. Also, in the Minneapolis-St. Paul study, the pipe was being installed in the city which would account for the higher cost.

Since this study is a continuation of the work of Roberts and Bahr (7), it would be sensible to compare results and determine if the idea of waste heat utilization is economical.

In the Roberts and Bahr study, it was calculated that for certain configurations that using waste heat was cheaper than using conventional fuel to heat greenhouses or raise fish. Utilizing the most economical conveyance system, it was found that the net savings gained over a 20 year period, when using both greenhouse and aquaculture facilities, is more than twice the cost of the conveyance system.

Therefore, by looking at these results it can be said that waste heat utilization is economical and useful. But, as was stated by Daugard and Sundaram (2), the impacts of waste heat utilization should not be viewed in terms of projections based on present-day engineering, social and economic considerations, but should be viewed in terms of the basic changes in the coming years in the nature of generation and utilization of energy.

SUGGESTIONS FOR FUTURE WORK

This research has covered a small but significant area of the work necessary in the study of waste heat utilization. Some of the work that should be done to further investigate the technical and economic feasibility of waste heat utilization is stated as follows:

(1) Study existing systems, both in the U.S. and in other countries, to obtain actual cost data on maintenance costs, pipe placement techniques, the nature of the ground in which the pipe must be laid, the effect of existing ground use, and actual savings over conventional local heating methods.

(2) Investigate the economic effects of introducing locally produced fish and vegetables into the market.

(3) Study the possibility of reduced operating costs associated with a system employing a number of small pumps in parallel, with only the pumps needed at any given time in service.

(4) Examine the regulations that may govern the transport and/or use of waste heat and the economical and technical problems associated with those regulations.

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Appendix A

Pipe Specifications

Ductile Cast Iron (21)

Inside Diameter (in)	Thick-ness (in)	Weight per foot (lbs)	Cost per foot (dollars)	Pressure Range (psi)
8	.27	24.1	8.10	350
10	.29	31.8	10.60	350
12	.31	40.6	13.40	350
14	.33	50.6	16.30	350
16	.34	59.6	19.00	350

Ductile Cast Iron Thermal Conductivity - 19 BTU/hr ft ^{o}F

Polyvinyl Chloride *

Inside Diameter (in)	Thick-ness (in)	Weight per foot (lbs)	Cost per foot (dollars)	Pressure Range (psi)
8	.50	8.2	12.30	250
10	.59	12.3	21.10	250
12	.69	17.1	29.40	250

Polyvinyl Chloride Thermal Conductivity - .8 BTU/hr ft ^{o}F

* PVC Pressure Pipe Sales Brochure, Robintech Corp., Industrial Ave. Grinnell, Iowa 50112

Appendix B

Pump Characteristic Curves

In Figure 4 the characteristics for a 10 x 8 x 12 horizontal split case pump are shown. On these curves the head losses, for the 14 and 16 inch ductile cast iron pipe, for a distance of 1 mile have been plotted. From this figure it can be seen for the 14 inch case that the pump, along with a 25 horsepower motor can supply up to 4 more feet of head than is needed. Also, for the 16 inch case this pump with a 20 horsepower motor can supply up to 12 more feet than is needed. Therefore, if there is a greater head loss than was originally calculated, the pump will be able to supply the need.

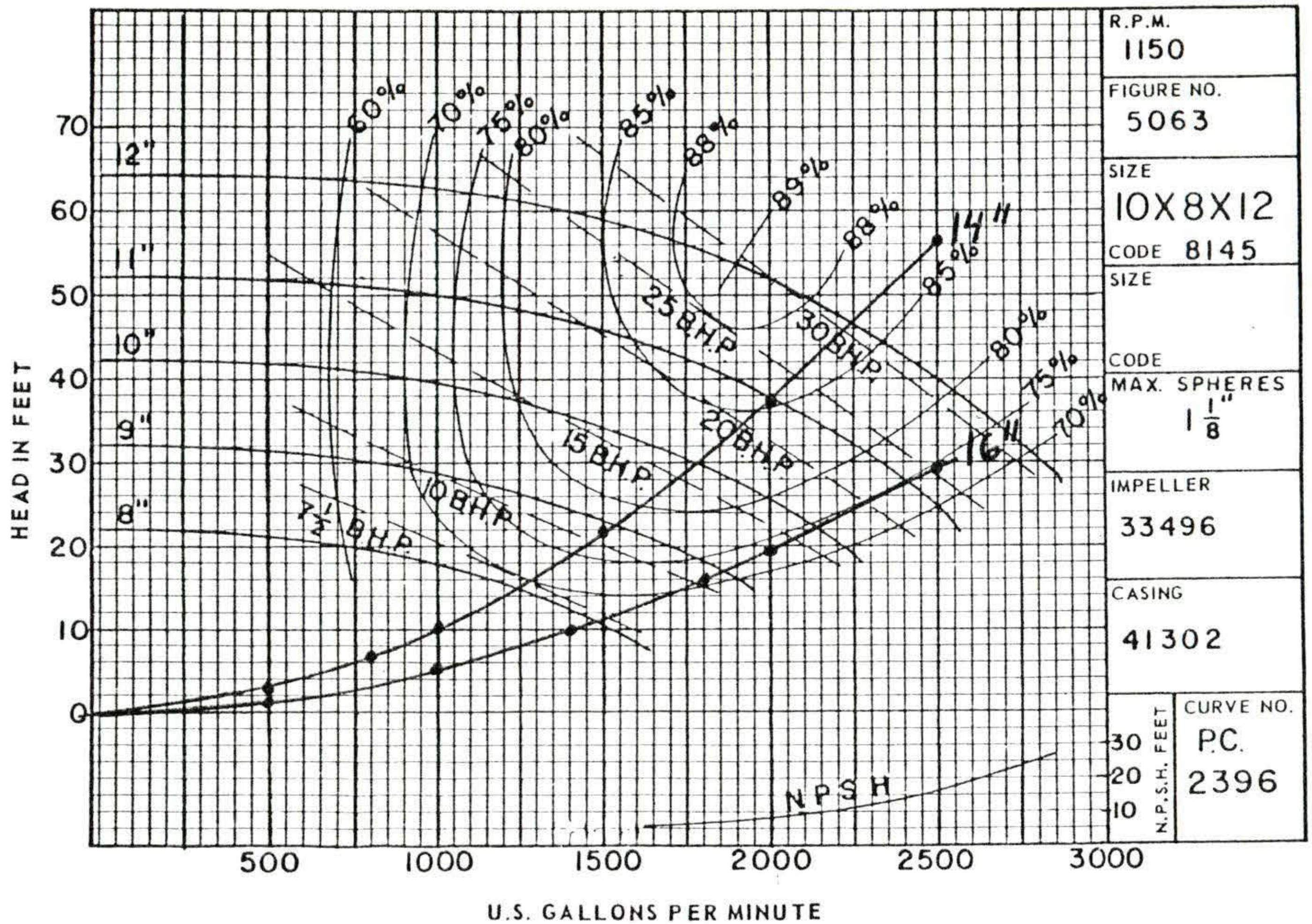

Figure 4. Pump Characteristic Curves

Appendix C

Average Monthly Air Temperatures and Velocities [7]

Month	Outside Air Temperature	Wind Velocity
January	21.9 °F	13.0 mph
Feburary	33.1 °F	13.5 mph
March	38.3 °F	15.0 mph
April	53.7 °F	14.9 mph
May	59.0 °F	13.0 mph
June	69.2 °F	12.0 mph
July	74.7 °F	10.8 mph
August	71.4 °F	9.0 mph
September	64.0 °F	10.2 mph
October	46.9 °F	11.5 mph
November	31.1 °F	14.6 mph
December	19.3 °F	14.5 mph
AVERAGE	48.6 °F	12.5 mph