1 of 1

RESEARCH REPORT

INCREASED USE OF REJECT HEAT FROM ELECTRIC GENERATION

by:

Dr. Richard W. Leigh
461 Central Park West
New York NY 10025

and

Michael Piraino
The Gas Research Institute
8600 Bryn Mawr Avenue
Chicago IL 60631

Work Supported by the
Utility Systems Division
Office of Utility Technologies
UNITED STATES DEPARTMENT OF ENERGY

Submitted to:

The Energy Efficiency and Conservation Division
Department of Applied Science
Brookhaven National Laboratory
Upton NY 11973

This work was performed under the auspices of the U. S. Department of Energy,
Washington, D.C., under Contract No. DE-AC02-76CH00016

MASTER

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

ABSTRACT

This study aims to determine existing barriers to greater use of reject heat by electric power producers, including utilities and cogenerators. It includes analytical studies of the technical and economic issues and a survey of several electric power producers.

The core analytic findings of the study are that although electric utility-based, cogenerated district heating is sometimes cost competitive with currently common furnaces and boilers, it is not clearly less expensive, and is often more expensive. Since market penetration by a new technology depends on strong perceived advantages, district heating will remain at a disadvantage unless its benefits, such as lowered emissions and decreased reliance on foreign oil, are given overt financial form through subsidies or tax incentives.

The central finding from the survey was that electric utilities have arrived at the same conclusion by their own routes; we present a substantial list of their reasons for not engaging in district heating or for not pursuing it more vigorously, and many of them can be summarized as the lack of a clear cost advantage for district heat.

We also note that small-scale district heating systems, based on diesel generators and located near the thermal load center, show very clear cost advantages over individual furnaces. This cost advantage is consistent with the explosive growth currently observed in private cogeneration systems.

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

TABLE OF CONTENTS

1. SUMMARY AND CONCLUSIONS	1
2. THE COSTS OF REJECT HEAT	5
2.1 Thermal/Electric Tradeoffs in Cogeneration	5
2.2 Condensing Steam Turbine Generators	6
2.3 Backpressure Steam Turbine Generators	8
2.4 Combustion Turbine Generators	9
2.5 Reciprocating Engine Generators	9
3. THE COSTS OF DELIVERING THERMAL ENERGY	11
3.1 Transmission Costs - BNL Generic Model	11
3.2 Transmission Costs - PSE&G	11
3.3 Distribution Costs - BNL Generic Model	13
3.4 Distribution Costs - PSE&G	14
3.5 Distribution Costs - Reconciliation	15
4. DISTRICT COOLING	15
4.1 Cost of Electric Compression Cooling	16
4.2 Cost of Absorption Chillers	17
4.3 Distributed District Cooling	18
4.4 Central District Cooling - Four Pipe	20
5. AVAILABLE REJECT HEAT	22
5.1 Condensing Steam Turbine Generators	22
5.2 Backpressure Steam Turbine Generators	23
5.3 Combustion Turbine Generators	24
5.4 Reciprocating Engine Generators	24
6. REASONS UTILITIES DO NOT FULLY EXPLOIT THE RESOURCE	25
6.1 Large Eastern Utility #1 (LEU#1)	25
6.2 West Coast Utility (WU)	26
6.3 Large Eastern Utility #2 (LEU#2)	26
6.4 Large Eastern Utility #3 (LEU#3)	27
6.5 District Heating Utility (DHU)	27
6.6 District Heating Developer	27
APPENDIX A: INFLATION FACTORS	31
APPENDIX B: EXTRACTION TURBINE HEAT RATES	32
APPENDIX C: THERMAL ENERGY TRANSMISSION COSTS FROM PSE&G	32
APPENDIX D: MANIPULATION OF UNCERTAINTY AS DATA ARE COMBINED	33

LIST OF FIGURES

Figure 2-1: The Cost of Condenser Heat.	8
Figure 3-1: Distribution System Costs from BNL Study.	13

LIST OF TABLES

Table 1-1: Costs of Cogenerated Thermal Energy.	2
Table 1-2: Costs of Distributing Thermal Energy.	2
Table 1-3: Quantity and Cost of Cogenerated District Thermal Energy. . .	2
Table 2-1: Characteristics of Extraction Steam Turbine Retrofit.	7
Table 2-2: Costs of Thermal Energy from Extraction Retrofit.	7
Table 2-3: Power and Heat Balance, Large Diesel Engine at Full Load. . .	10
Table 2-4: Costs of Thermal Energy from Diesel Engines.	10
Table 3-1: Unit costs of transmission of thermal energy from PSE&G study for underground, dual steel pipe (1992 \$/mmBtu - mile).	12
Table 3-2: Generic Model Estimates of Distribution-Related Unit Costs for FRP and Steel Pipe (1992\$/mmBtu).	14
Table 3-3: Distribution Costs from PSE&G Study.	15
Table 4-1: Unit Costs of Cooling - Electric Chillers.	17
Table 4-2: Costs of Cooling - Single Effect Absorption Chillers.	17
Table 4-3: Costs of Cooling - Double Effect Absorption Chillers.	18
Table 4-4: Costs of District Cooling - Distributed Chillers.	19
Table 4-5: Costs of District Cooling - Four-Pipe, Central Chillers. . . .	21
Table 5-1: Condensing Turbine Thermal Resource at Selected Utilities. .	23

ACKNOWLEDGEMENT

The authors would like to thank Dr. John Andrews of BNL for his support, encouragement and many helpful suggestions during the preparation of this report, and Floyd Collins of the U.S. Dept. of Energy, Office of Utility Technologies, for its support.

1. SUMMARY AND CONCLUSIONS

This study aims to determine existing barriers to greater use of reject heat by electric power producers, including utilities and cogenerators. It includes analytical studies of the technical and economic issues, a broad survey of several electric power producers and a detailed survey of those found to have the largest unused resource. The core finding of the study is that their reasons for not engaging in district heating or for not pursuing it more vigorously are often well-founded: if the power producer is using a large, efficient steam turbine to generate electricity, the losses involved in running it so as to produce thermal energy at a useful temperature are sufficient, when combined with the cost of a substantial transportation and distribution system for the thermal energy, to make cogenerated thermal energy only marginally competitive with distributed alternatives such as furnaces and boilers. This is not a sufficiently strong inducement to persuade customers to switch to what is, to them, a "new" technology. The situation is much more positive for smaller, distributed systems based on diesel prime movers, systems which are the exclusive province of independent power producers. In the remainder of this Summary we outline the steps leading to this conclusion; the details are in the succeeding sections.

As a basic ground rule, we have examined only currently proven technologies in detail. Electric utilities are technically conservative, and have made their decisions to implement or not to implement district heating and cooling on the basis of technologies they could buy with confidence*. Where appropriate, we will point out where the availability of a new technology might have changed the situation, but for the most part this will not be a key issue.

The costs of district heating systems and of their components vary widely, depending on the physical circumstances of the deployment, the economic nature of the utility or other entity developing the system and the assumptions used by the analyst. While aware that we may provoke controversy, we have attempted to reduce the various cost components of district heating systems to unit costs (dollars per million Btu, or \$/mmBtu[#]) and to show, wherever possible, a range of values within which the costs of the majority of systems would fall. Given this wide range of values, we have used very simple models (explained below) to relate capital costs and annual expenses. The costs we arrive at are therefore "ballpark estimates"; technologies which look promising or implausible in our analysis probably are just that, while technologies which appear uncertain will require case-by-case analysis.

In Section 2 we examine the four major technologies serving as sources of heat for cogenerated district heating and cooling: Steam extraction from condensing turbines, backpressure turbines and heat recovery from combustion turbines and from reciprocating diesel engines. Technically, there are no barriers to the design and construction of efficient and readily operated systems, many of which are discussed below. However, the economic results, summarized in Table 1-1, indicate that thermal energy supplied by combustion turbines and diesel engines is significantly less expensive than that supplied by systems based on steam turbines. As discussed in Section 2-3, combustion turbines are not suitable sources of heat for district heating and cooling since they are used only sporadically, to meet peak electric loads, and the cost of

*For example, BNL researchers have suggested (see Ref. 10) a very low temperature (120°F) district heating system, feeding high efficiency heat pumps at the user's end. Such a system could turn out to be very competitive, but due to a lack of study and demonstration to date, it lacks credibility with the utility community.

Conventions can be confusing. "M" indicates "million" in the SI system, but not in conventional American units, where "m" indicates "thousand", rather than 1/1000, as it does in the SI system. We have elected to be 100% American, so "mmBtu" will refer to one million Btu, etc.

appropriate storage is prohibitive. (Combined cycle plants are base loaded, but are appropriately modelled as steam turbines, not as combustion turbines.)

In Section 3 we draw on both analytic studies and engineering experience to estimate the cost of delivering this thermal energy to residential and commercial loads; the results are in Table 1-2.

"Transmission" involves transport of large amounts of thermal energy over substantial distances - more than 1000 feet, for example, while "distribution"

describes the smaller piping that branches out to reach each individual customer. ("Distribution" costs here include heat exchangers on the customer's premises.) Systems involving smaller, local generators may not even need transmission, and in Section 3 we found the cost per mile of transmission line traversed so that systems could be compared appropriately. All district heating systems require distribution systems; interestingly, the wide range of possible costs does not reflect different load densities, which we find result in very comparable costs, but rather, uncertainty in the load factor and differences in different groups' estimates of the actual construction costs.

The results of these various estimates may be added together to give estimates of the total cost of supplying district heat to a consumer, presented in Table 1-3. For the large steam turbine-based systems we include the cost of one mile of transmission line. This is an item which can only be determined on a site-specific basis, but the cost is sufficiently low that lines

between one-half and two miles will have very similar total costs. Similarly, any system where the thermal energy must be transported ten miles or more will be at a serious disadvantage. For the combustion turbines and diesel engines the transmission costs are taken as zero, since they can (in principle) be sited near the thermal load. The uncertainty ranges in the totals are derived by assuming that the three uncertainties are independent, and that the ranges in the individual results represent a most probable value plus or minus one standard deviation, as explained in Appendix D.

For competitive, individual systems, current fuel prices are \$5-6 per million Btu for natural gas in the residential and commercial sectors, and distillate oil is about \$7/mmBtu. Efficiencies of 85% (furnace efficiencies above 90% are available) raise the fuel cost to \$5.90-7.00/mmBtu for gas and \$8.25 for oil. Capital and maintenance costs may add another \$2/mmBtu, giving the totals shown in Table 1-3.

	\$/mmBtu
Condensing Turbine (Extraction):	4.60-5.30
Backpressure Turbine:	3.00-4.00
Combustion Turbine:	0.50-1.00
Diesel Engine:	0.10-0.30

Table 1-1: Costs of Cogenerated Thermal Energy.

Transmission (\$/mmBtu-mile):	0.14 - 0.35
Distribution (\$/mmBtu):	2.60 - 7.00

Table 1-2: Costs of Distributing Thermal Energy.

	Quads	\$/mmBtu
Condensing Turbine:	Many	7.80-12.20
Backpressure Turbine:	Fraction	6.30-10.80
Combustion Turbine:	0	3.30-7.80
Diesel Engine:	0.1	2.80-7.20
Distributed Furnace/Boilers:		8.40-10.75

Table 1-3: Quantity and Cost of Cogenerated District Thermal Energy.

Clearly the condensing turbine source is of the most interest, due to its large potential. We find that carefully engineered designs under the best of circumstances will be competitive, but that there will also be many cases where individual systems are significantly less expensive. Also, because in the U.S., district heating is a "new" technology trying to displace an established technology (distributed furnaces and boilers), it is not enough for it to be "competitive". The fact that district heat costs no more than individual systems in many cases is not sufficient to lead to its widespread adaptation. Any new technology must offer some clear-cut advantage to the decision maker. If this advantage is significantly lower cost, the transition will be more or less spontaneous. Our results indicate that it is not surprising that this has not yet occurred. Other advantages of district heat, such as lower emissions or lower oil imports, must be translated through tax structures, subsidies or regulations into a clear signal to decision makers if they are to be realized. The utilities' responses to this situation will be discussed below.

Although the main purpose of this study is to determine why better use is not made of reject heat from these large, utility-based systems, we have to point out in passing that more localized systems, based on diesel engines or combustion turbines, appear to be cost effective under almost all circumstances. Although the available energy is currently small, independent power production is undergoing explosive growth, and the development of localized systems based on diesel engines with heat recovery appears to be extremely cost effective, with backpressure turbine-based systems a close second.

Section 4 contains a discussion of district cooling. After dissecting the market into the various technical possibilities, we show that the only system that appears to be economically viable is a large, centrally located double effect chiller, especially when it is competing with smaller, distributed electric compression systems. Even this system is only viable as an add-on to a presumed heat distribution system, and, when both capital and operating expenses are accounted for, it has no significant effect on the costs for thermal energy. That is, the presence of the cooling system does not increase the viability of the heating system.

Section 5 examines national statistics on electric utility plant and generation to determine several locations with large quantities of reject heat and a market for that heat which is, at least in principle, large enough to make good use of it. A summary sketch of the results has been presented in Table 1-3. It became rapidly apparent that there were many such examples, and the only difficulty lay in choosing which ones to contact. At this point the survey became less than random, since we ended up interviewing utilities for which we could make contact with a knowledgeable representative after a modest number of phone calls. We believe this means we contacted utilities which are more interested than average in district heating, and may have derived a perspective which is more optimistic about the future of district heating than the industry at large.

In Section 6, several of the utilities selected as likely to profit from increased use of reject heat are questioned about the current status of district heating in their service territories. Despite a few optimistic assessments of the technical possibilities, expectations were uniformly very low for any expansion of cogeneration-based district heating systems in the near term. The most commonly cited problems were:

- High capital costs for piping;
- Significant capital costs for retrofit of power plant;
- The cost of lost electricity due to cogeneration;
- Complexities of operating the cogeneration system;

- Complexities of allocating costs in the cogeneration system;
- Difficulties in getting consumers to recognize the true total costs of their individual systems (that is, including capital and maintenance costs);
- Inadequacies in the Environmental Protection Agency's (EPA's) emissions policies, which do not recognize reductions due to the shut-down of individual units; and
- Lack of interest on the part of top management, due in part to the small size of even successful district heating operations compared to the electric delivery system.

This list of problems seems to multiply without limit as one converses with utility personnel, and it bears out the limited competitiveness of utility-based district heat we found in the financial analyses of Sections 1-4. The days of significant expansion of large, utility-operated, steam-turbine based district heating systems in the U.S. are over, at least until there is a significant change in circumstances, such as a dramatic run-up in fuel prices.

District heating systems offer substantial environmental benefits, but they often do so at a significant cost (which could be developed from these figures). Without dramatic changes in current law, it is unreasonable to expect either utilities or their customers to voluntarily undertake these added costs. Indeed, it is possible that comparable environmental benefits could be achieved at lower cost through improvements in individual boiler and furnace emission controls.

At the same time, it appears that local, diesel-based systems are in a very competitive position and could enjoy strong growth over the next few years with minimal encouragement. This possibility will only be beneficial if the problems of pollution from these systems can be overcome through combustion control, more complete recuperation and (perhaps) small-scale scrubbing techniques. It is in areas such as these that R&D emphasis should be placed over the next few years.

2. THE COSTS OF REJECT HEAT

This section reviews the technical and economic aspects of increased use of reject heat by electric utilities and cogenerators. Different sorts of generation equipment require different decrements in electrical production to allow extraction of reject heat at useful temperatures, and the resulting economic trade-offs must be readily quantified. We first discuss the technical and economic quantities to be determined, and then examine the four major thermal conversion electric generation technologies.

2.1 Thermal/Electric Tradeoffs in Cogeneration

Any process converting thermal energy to mechanical or electrical energy involves certain unavoidable inefficiencies determined by the second law of thermodynamics and further limited by the particular thermodynamic cycle used to effect the conversion. The energy not converted to mechanical or electrical form is then available as heat, but at a temperature which is lower than the heat source, and which approaches ambient temperatures as the efficiency of the conversion process is increased towards its maximum. However, thermal energy has greater value the more its temperature exceeds ambient temperatures, since it may then be used for a wider variety of processes, including cooling and chemical processing. Even when space heating alone is considered, higher temperature thermal energy allows the use of smaller heat exchangers, and so is more valuable. Clearly, thermal energy available at ambient temperature is of no value in any application.

Cogeneration therefore poses a complex problem of engineering optimization: if thermal energy is extracted at higher temperatures, making it more valuable, the extraction process tends to reduce the production of electricity. For existing systems, fuel, maintenance and capital charges remain the same, so the unit production cost of electricity rises for that machine. More important, the foregone electric energy must be obtained elsewhere, and for an electric utility, the value of that foregone energy will be the marginal cost of energy to the utility at any given time. Rather than the relatively low cost of energy from an efficient, existing coal-steam unit, the marginal cost of energy is much more likely to be the higher cost of energy from a combustion turbine, at least during the ten to fourteen hours considered "on peak".

The case of a new machine which is designed to cogenerate might appear to be different, since it can be "oversized" from the start to provide whatever amount of electric power is optimal for the utility at that time. However, it is still producing less electricity for a given amount of fuel, and the capital costs of a new plant are such that its total costs of production will be close to the utility's time-averaged marginal cost in any event. (If they were much lower, the plant would have been built sooner.) (A new machine which is to have a certain electric capacity must have larger boilers since its electric efficiency will be lower. This will be somewhat offset by savings associated with the lower costs of the smaller, "backpressure" turbines.)

Cogenerated thermal energy, in the form of steam or hot water leaving a generating station, is therefore not in any sense "free" in most applications. Contributions to the variable costs include the marginal price of foregone electric energy and increased operations and maintenance costs chargeable to the operation of heat-recovery equipment. Contributions to the fixed costs include the annualized capital expenses of heat exchangers and piping within the generating station and any costs incurred adapting the generating unit to permit use of the thermal energy, such as extra steam-extraction ports. In addition, if a generating unit is adapted to operate far from its design point (such as converting a condensing turbine to backpressure use) then parts of the machine (the low-pressure stages of the turbine) are no longer being used productively, and their residual capital costs should also be assigned to the thermal energy

recovered. For the sake of simplicity and generality, we will often not include some of the smaller of these factors in our calculations, so our estimates of the cost of thermal energy will be, in this respect, on the low side.

It will be necessary to combine capital and operating expenses in the following analysis. Electric utility economics are complex and particular to the utility being analyzed, due to the variations in regulatory climate and other external factors. Rather than try to construct a detailed model of the utility's economics, we will simply annualize capital expenses at 10% per year and add them to variable expenses to obtain a cost of cogenerated thermal energy. (This is equivalent to amortizing over 14 years at 5% interest, 20 years at 8%, and so on.) Also, costs originally estimated in several different years must be escalated to 1992 dollars for comparison with each other and with current prices for alternatives. This calculation is described in detail in Appendix A.

2.2 Condensing Steam Turbine Generators

These technically mature devices are the standard large electric utility energy conversion system, used in both fossil fuel and nuclear stations¹. They range in size up to well over 1000 MW in a single unit, although tandem installations are also common. Steam is delivered at as high a temperature and pressure as possible - up to 1000°F and 3500 psia. It then cascades through several stages of turbine blades, producing the rotational motion which powers the alternator, in the process undergoing at least one reheat cycle. At the downstream end the steam gives up its latent heat to cooling water in a condenser, and the temperature of the condenser determines the pressure of the steam at this point.

In the condensing steam turbines commonly used by electric utilities, the condenser is cooled to the lowest practical temperature, usually 90 to 110°F, by cooling water from a river or cooling tower. This results in traditional, non-cogenerating operation, where every possible kWh of electric energy is wrung from the steam. To cogenerate, the most efficient process is to remove some steam from one or more "extraction points" along the turbine, lessening the flow through the turbine and lowering its output⁺. The extracted steam is then either fed directly to the load for a steam distribution system, or used to heat water in heat exchangers for a hot water distribution system. By removing steam at various pressures and temperatures along the turbine and using it to heat system water or serve load in appropriate "counterflow" configurations of heat exchangers the quality of the thermal energy can be preserved.

Analysis of a turbine to determine its efficiency when operating with steam extraction can only be done as a specific case study for a particular turbine. Such a study has been done as part of a feasibility study for a district heating system in metropolitan New Jersey² based on a large condensing steam turbine supplying, at full-load design point, 57,200 gallons per minute of 221°F water, with a return temperature of 165°F. The technical characteristics of the system are summarized in Table 2-1 for both the existing turbine as it produces only electricity, and for the turbine as retrofitted to supply the district heating system. From the perspective of "First Law" efficiency^{**} the implementation of district heating looks like a significant improvement.

⁺This means that the existing turbine will not be used optimally, since blades adequate for greater flow are now under-utilized. We ignore any associated costs.

^{**}"First Law" (of thermodynamics) efficiency simply reports what fraction of supplied energy was delivered to load, without regard for its quality.

	Electric Only	Full District Heating Load
Fuel Input (mmBtu/hr):	5355	5432
Electricity		
Gross Capacity (MW):	652	560
Parasitics & Environmental (MW):	-52	-52
Net Capacity (MW):	600	508
Net Heat Rate (Btu/kWh):	8920	10685
Electric Efficiency:	38.3%	31.9%
Availability:	62%	62%
Thermal Supply		
DH Load (mmBtu/hr):		1600
Thermal Efficiency:		29.5%
Total Efficiency:	38.3%	61.4%

Table 2-1: Characteristics of Extraction Steam Turbine Retrofit.

However, an examination of the costs of actually operating this system, outlined in Table 2-2, indicates that the situation is not so clear-cut. The electricity not produced must be paid for, and in their assessment PSE&G applied their time-averaged internal rate of \$0.069/kWh in 1982. This is the rate one division of the utility charges another for electricity, and represents the company's best estimate of its "marginal cost" of replacement. Although ten years have passed, wholesale electricity rates have risen very little³ and we use the same value for 1992. Their construction costs for the retrofit include heat exchangers and in-plant piping but omit significant work on the existing turbine and control system (and are therefore labelled "partial" and represent a lower limit on the cost). These costs do not include work on the district heating system; they have been escalated by 34% as described in Appendix A. Factoring in the uncertainties discussed above, the result is that thermal energy from this plant would cost \$4.60 - 5.30/mmBtu today.

	1981 \$	1992 \$
Electricity Cost (¢/kWh)	6.9	6.9
Partial Retrofit Costs (M\$):	43-80	57-107
Cost of Thermal Energy (\$/mmBtu)		
Fixed costs at 10%/yr:	.49-.91	0.70-1.31
Electricity:	3.96	3.96
TOTAL:	4.45-4.87	4.66-5.27

Table 2-2: Costs of Thermal Energy from Extraction Retrofit.

2.3 Backpressure Steam Turbine Generators

The turbines in these devices are constructed so that the steam exits at a temperature significantly above ambient so that it may be used for some downstream process; here the process will be district heating (or the operation of thermally activated chillers for district cooling) and the extraction temperatures will be in the range of 220-300°F. Heat rates for various condenser temperatures up to 275°F are developed in Appendix B and are used to calculate how many kilowatt-hours are not produced per million Btu as the condenser temperature is increased (shown as the central line in Fig. 2-1) and what the cost of replacing that electricity is, again at internal rates of 6.9¢/kWh and a lower bound of 5.0¢/kWh (shown as the upper and lower lines).

The relatively high cost of extracted thermal energy is due in part to the assumption that the boiler provides high quality steam (1000°F = 540°C), as it does in large utility installations. If low pressure boilers (for example, 200 psig = 1400 kPa) are assumed, the low conversion efficiency results in smaller electric sacrifices when steam is extracted, making thermally activated technologies more economically attractive⁴. Such assessments may be useful in privately owned cogeneration systems operating under these conditions; for this study of electric utilities, steam is normally provided at the higher quality assumed in deriving Fig. 2-1.

Figure 2-1: The Cost of Condenser Heat.

To supply the same district heating system as in Section 2.2, cooling water will enter the condensers at 165°F and leave at 221°F; typical condenser calculations⁵ show that the condenser temperature (i.e. the temperature of the condensing vapor) will be about 20°F above the exit temperature of the water, or

240°F. Figure 2-1 then indicates that the cost of foregone electricity will be about \$2.80/mmBtu. Capital costs will vary, depending in part on whether the plant is designed to handle the thermal load from the beginning or is a retrofit. In either case, it must be able to cycle the thermal load on and off, to cope with seasonal variations, and when all incremental capital costs are added and annualized, \$3 - 4/mmBtu brackets the probable range. If higher temperature thermal energy is needed, as for thermally activated district cooling, the cost could rise by another \$1/mmBtu.

2.4 Combustion Turbine Generators

Combustion turbines operate on the Brayton cycle, with the exhaust gasses from the direct combustion of natural gas or oil passing through the turbine to produce rotary motion of the alternator. Intake air is first compressed by turbine stages which are coaxial and rotate with the power stages. It is economically impractical to build turbines with rotors sufficiently large to cool the working fluid to ambient levels while still extracting work; practical limits are such that the gasses leave the turbine about 1000°F, and the recovery of this heat requires only the addition of a suitable boiler or heat exchanger.

This is often done in what is called a "combined cycle" installation, where the exhaust gasses from a combustion turbine boil water into steam which is then used to drive a condensing turbine, also producing electricity^{1,6}. Such combined cycle systems are efficient and clean, operating at efficiencies above 40% (heat rates of less than 8600 Btu/kWh). Several gigawatts (thousands of megawatts) of such systems have been installed in recent years by independent power producers, with the output being sold to utilities for resale. However, heat recovered for district heating from these systems, by using either extraction or a backpressure turbine, would be produced by the same methods and at the same costs as was the case for the systems of Sections 2-1 and 2-2, and for the purpose of analysis, these combined cycle plants should be regarded as special cases of steam turbines, rather than as combustion turbines.

Electric utilities also have large numbers of combustion turbines installed with no heat recovery equipment (other than regenerators contained within the turbine system). Heat exchangers could be added to these systems, and heat recovered from them at high temperatures at an annualized, unit cost of \$0.50 - 1.00/mmBtu for the heat exchangers.

For example, PSE&G estimated the retrofit of 8 22 MW(e) combustion turbines at about \$12 million when converted from 1981 to 1992 dollars with the 50-50 escalation factor in Appendix B'. They also estimated that this unit would produce about 1.85×10^{12} Btu/year⁸ (70% capacity factor). Annualizing at 10%/year and dividing we find \$0.66/mmBtu; the range of \$0.50 - 1.00/mmBtu reflects various site-specific contingencies and includes a slight de-rating of the turbines due to backpressure from the heat exchangers.

2.5 Reciprocating Engine Generators

Internal combustion engines based on reciprocating pistons coupled to a rotating alternator through a crankshaft are a mature and widely used technology.^{1,9,10} They are available in sizes ranging from one to several thousand kilowatts, in designs based on both diesel and spark ignition, burning gasoline, natural gas or fuel oil. We will concentrate on large, four-cycle diesel cycle systems designed for fuel oil, natural gas or both ("dual-fuel"). Natural gas-fired diesel engines often use fuel oil to start up, as it is difficult to initiate gas ignition in a cold engine.

Reciprocating engines are well-adapted to cogeneration, since waste heat is very easily recovered from the various components summarized in Table 2-3.

	Available	Recoverable	Temperature	
	(%)	(%)	°F	°C
Mechanical Work:	38	38	na	na
Jacket Water:	17	17	170-250	75-120
Lube Oil:	4	4	160-190	70-90
Exhaust Gas:	34	24	600-850	315-450
Radiation, leaks:	7	0	na	na
Total:	100	83		

Table 2-3: Power and Heat Balance, Large Diesel Engine at Full Load.

The systems are clearly efficient; 38% corresponds to a heat rate of less than 10,000 Btu/kWh even after the 95% generator efficiency is factored in. (Generator efficiencies can drop to 85% for smaller systems or part-load conditions.) Heat is easily recovered from the water jacket by circulation, with or without isolating heat exchangers, near the lower end of the indicated range. A 10°F (6°C) temperature rise in the circulating water is typical. The high end of the temperature range for jacket water range corresponds to 15 psig (2.0 atmosphere) steam produced by ebullient cooling. Here pressurized water boils throughout the engine; the steam then rises up and out. This results in very uniform temperatures, but they are nonetheless high temperatures, and the environment is stressful for lubricating oil, resulting in a shorter life. Oil cooling involves the straightforward installation of oil to water heat exchangers. The exhaust gasses contain 30 to 35% of the fuel input energy, but only a portion of this can be recovered unless heat exchangers that can tolerate acid condensate are employed, not a common practice today¹¹. This limits the exhaust temperature to 325°F (160°C) or above, producing steam at 15 psig.

In Table 2-4 we develop costs for the recovery of heat from the water jacket and exhaust at 250°F (120°C) and 15 psig (2 atmospheres), with the exhaust heat recovery system tuned to avoid condensation. The costs are taken from Refs. 9, 10, escalated to 1992 dollars as described in Appendix A. Equipment capital costs and thermal energy costs can vary by at least 25%, depending on the installation. Lower temperature heat recovery is even less costly. The unit energy costs were based on the recovery of 41% of the fuel input energy and 7884 hours per year of operation (90%); most diesels have higher availabilities¹². Operating and maintenance costs are assumed billed to electric production except those directly associated with heat recovery, such as heat exchanger cleaning, which are taken to be 30% of total maintenance costs for these engines.

Electric Power	Thermal Power	Equipment Cost		Maintenance	Total
(kW)	(mmBtu/hr)	\$/kW(e)	\$/mmBtu	\$/mmBtu	\$/mmBtu
150	0.55	52-88	.17-.28	.11	.28-.39
500	1.80	27-45	.08-.14	.09	.17-.23
1000	3.70	23-39	.08-.13	.07	.15-.20
5000	18.4	23-39	.08-.13	.02	.10-.15

Table 2-4: Costs of Thermal Energy from Diesel Engines.

3. THE COSTS OF DELIVERING THERMAL ENERGY

In this section, the costs of creating or expanding thermal energy delivery systems in various locations and under various conditions are developed from industry experience to date. We distinguish between "transmission costs", corresponding to the transport of thermal energy through large lines with little or no branching, over distances of, say, one thousand feet or more, and "distribution costs" incurred in the multiply branched networks that deliver the energy to each customer. We derive our estimates from two sources: an earlier BNL study¹³ based on generic construction cost estimates for a variety of community types and applied in Milwaukee¹⁴, and a detailed study carried out by PSE&G for metropolitan New Jersey².

3.1 Transmission Costs - BNL Generic Model

The BNL DH model incorporates a piping optimization module which determines the lowest cost pipe size for a given installation, taking account of capital costs and pumping expenses. The unit costs of delivered energy attributable to transmission can only be derived from such an optimization process, since the total cost of the line must be divided by the energy transmitted through it, and this will be determined from trade-offs between the costs of pumping and the cost of pipe. However, in studies to date using the BNL model, the transmission lines have either been left out or are not described in sufficient detail to allow us to break these optimized costs out in a useful manner. We will therefore not try to develop overall costs of thermal energy transmission from these reports.

Two results from the BNL studies are worth noting, however. First, they developed costs for Schedule 40 steel pipe as well as for Fiberglass reinforced polyester (FRP), in both cases looking at buried, appropriately insulated double pipes. The steel pipe costs from two and one-half to three and one-half times as much as the FRP, for four inch to ten inch diameter pipe. Clearly the FRP could make the difference between viability and non-viability for a marginal project.

Second, the BNL costs per foot for installed steel pipe in rural installations are just about twice the costs estimated for the same size pipe by PSE&G, after correcting for inflation. We will incorporate this in the uncertainty of probable costs we derive from the PSE&G reports in the next section.

3.2 Transmission Costs - PSE&G

PSE&G performed a detailed cost estimate for DH pipe installations¹⁵. A spreadsheet printout showing detailed calculations from their numbers is included as Appendix C; here we will summarize the results. The estimates are based on work done for them as part of the project by Stone and Webster and by the gas pipeline division of PSE&G. PSE&G compared these estimates with previous estimates by Burns & Roe and found good agreement, and included corroborating reviews by various experts; the costs per foot to put pipe in the ground appear to be on firm ground. As usual, they were here escalated up to 1992 dollars.

However, a problem appeared: the flow rates derived by PSE&G from their optimization process were very high (25 feet per second), and as a result, a given pipe carried a great deal of energy. However, the electric consumption of the pumps and its cost were also great, too great to be an optimal result. Scrutiny of the numbers indicated that they probably optimized using an erroneous cost of \$1.00 per megawatt-hour, rather than the \$69/MWh they said they were using (and which we are using here). Appendix C shows the peak thermal power specified by PSE&G, and the peak power found when our spreadsheet was used to optimize the system using the realistic electricity price. Although our

optimization cut the flow (and delivered thermal power) by about a factor of three, it also cut the total unit cost of delivered energy in half.

The load factor of 70% is appropriate to transmission lines (where it would be too high for distribution) because they will be used to bring thermal energy from a "baseload" plant, from which heat is being extracted whenever possible. In the PSE&G system design, various smaller plants and local boilers are used to meet peak loads, but these are dispersed and do not use the transmission lines. Variation in total unit costs was close to variation in the load factor: if it was increased by 10%, to 77%, the total unit cost fell by about 9% in all cases, and when it was decreased by 10%, to 63%, the total unit cost increased by about 11%. The transmission lines were planned to range in size from 18 to 42 inches in diameter. Calculations were not provided for these pipe sizes, however, and we show results for the closest available sizes, 20 and 36 inch diameters.

The results of the spreadsheet calculation are summarized in Table 3.1, where we show results for dual pipes in rural and urban environments. The results are presented as dollars per million Btu delivered to the customer and per mile of transmission line. After multiplying by an appropriate line length, they may be added to the cost of thermal supply derived earlier and that of distribution, to follow.

Dia- meter	Delivered Energy	Capital Cost	Cost of Thermal Losses	Cost of Pumping Energy	Total Unit Cost
(in)	mmBtu/yr				
RURAL:					
20	1,300	0.17	0.01	0.07	0.25
36	4,050	0.09	0.01	0.04	0.14
URBAN:					
20	1,540	0.26	0.01	0.10	0.37
36	4,810	0.13	0.01	0.06	0.20

Table 3-1: Unit costs of transmission of thermal energy from PSE&G study for underground, dual steel pipe (1992 \$/mmBtu - mile).

Capital costs clearly dominate the expenses, but the cost of electricity for the pumps is not negligible. This is the basis for current research on friction-reducing additives. Success in reducing pumping power requirements would require re-optimization, allowing the use of smaller pipe for a given thermal power. Thermal losses are evidently a very small part of the cost and were here calculated assuming the cost of thermal energy supply was \$5 per million Btu; for less expensive sources, such as diesel engines, the cost of thermal losses becomes completely negligible.

There remain three factors which could affect these estimates of the cost of thermal energy transmission. First, PSE&G indicated that wherever possible, they would employ transmission lines raised on supports rather than buried in the ground. They estimate this would result in about a 20% reduction in capital costs, and would make maintenance and repair easier. Since capital costs are 65-70% of total unit costs, this could result in about a 14% reduction in total unit costs before re-optimization, and slightly more after.

Second, comparison with the BNL study discussed in Section 3.1 raises the question of whether PSE&G's cost estimates for steel pipe systems are accurate, since the BNL assessment, based on actual quotes from pipe vendors and excavation costs from estimation manuals, came up with installed per mile costs about 80% higher than PSE&G's. As in the previous paragraph, this would suggest that unit energy costs could be higher than in Table 3-1 by up to perhaps 50%.

Finally, the BNL study highlighted the strong cost advantages of FRP piping systems - reductions of 60-70% in capital costs seem quite possible. However, the FRP piping has not yet enjoyed widespread acceptance and may not even be available in the 20" - 36" sizes appropriate to long distance lines. Also, it cannot withstand as high pressures as steel can, and the PSE&G system design had pressures up to 15 atmospheres in the transmission lines.

Rather than fold all these possibilities in quantitatively, we will use them to highlight the uncertainty of this process and price transmission of thermal energy at the outer limits of our formal estimates:

Final Estimate of Transmission Costs: \$0.14 - 0.35 per million Btu delivered per mile of line.

3.3 Distribution Costs - BNL Generic Model

The BNL DH model uses the piping optimization model to derive cost estimates for the distribution system, given information on peak load per customer, number of customers per block and system size in blocks. It allows specification of four different installation classes - high-density urban, low-density urban, suburban and rural - and incorporates various impacts these different location types will have on installation costs, such as the amount of hand or machine digging and the type of surface to be broken through and restored. The model also incorporates costing algorithms for dual pipe water systems made of low-temperature polyethylene pipe, fiberglass-reinforced polyester (FRP) and Schedule 40 steel pipe, and a single pipe steam system of steel.

In reference 10 the model was applied to a wide variety of loads and community sizes, corresponding to load densities ranging from very low up to 150 mBtu/ft²-year. (For comparison, a Swedish rule-of-thumb for load densities appropriate to district heating is from 10 mBtu/ft²(land)-yr on up.) In each case the portion of the unit cost of delivering energy attributable to the distribution system was presented (in Appendix II).

In Fig. 3-1 these costs (escalated to 1992 dollars) are shown as a function of load density. All systems where the distribution costs are over \$5.00/mmBtu are omitted as impractical; potentially "practical" systems emerged, however, over a wide range of load densities. All costs

Figure 3-1: Distribution System Costs from BNL Study.

were developed for FRP piping systems, and correspond to hot water distribution. The horizontal lines correspond to the expected range of costs, taken as the average plus or minus the standard deviation of the 24 systems included here.

These systems must incorporate a water-to-water heat exchanger at the customer end to ensure water quality. The BNL study took the price for such heat exchangers to be \$11.54 per thermal kilowatt in 1992 dollars. Annualizing, converting units and assuming a customer load factor of 26% (see next paragraph), this corresponds to a unit charge of \$0.15/mmBtu which must be added to the distribution costs.

The systems examined all had the same load structure, resulting in a 26% load factor; without re-running the model we cannot determine sensitivity to this parameter, but because the primary dependence of unit costs on load factor is through the division by total annual load, we assume that the approximate one-for-one dependence observed for the transmission lines will hold here also. That is, if we take the likely variation in load factor to be 20% (i.e. ranging from 21 to 31%), we would expect an additional variation of 20% in the costs attributable to distribution. Since this source of uncertainty is independent of the factors giving rise to the scatter among the various systems, we must take the root-mean-square sum of the two fractional errors; this gives rise to the unit cost estimates for FRP piping shown in Table 3-2.

Steel pipe is more expensive. Examination of the pipe cost tables in Ref. 13 indicates that in various installation conditions relevant to distribution, it is more expensive by a factor of 3.0 ± 0.5 . Treating this as another independent variation, we take the root-mean-square sum of the fractional errors to derive the range shown for steel in Table 3-2.

	Load Factor	
	.31	.21
FRP:	0.60	1.70
Steel:	1.45	4.85

Table 3-2: Generic Model Estimates of Distribution-Related Unit Costs for FRP and Steel Pipe (1992\$/mmBtu).

3.4 Distribution Costs - PSE&G

PSE&G's subcontractor, Stone & Webster, in consultation with the gas pipeline division of PSE&G, worked up detailed estimates of the cost of installing distribution piping in Jersey City, Newark and a new development in the same area of New Jersey¹⁶. The estimates for Jersey City were also reviewed by personnel from the recently constructed St. Paul, Minnesota district heating system and by Danpower, a Danish district heating company, who both arrived at very similar estimates. A Swedish expert proposed considerably lower figures, based on "shorter pipe lengths". This led PSE&G to suggest that only studies based on contractor's quotations could decrease uncertainties further.

Table 3-3 shows the original capital cost estimates for one typical square mile of distribution system in each location, (in 1981 dollars), and the peak loads for each region. The costs are then multiplied by the "50-50" inflation factor from Appendix B and divided by the annual energy delivered in each representative square mile, based on the system-wide load factor of 27%.

To this is added 24¢ in 1981 dollars for the customer heat exchanger, or \$0.33 in 1992 dollars, after inflation by the producer price index for capital goods. The unit cost was derived from PSE&G's capital cost estimates¹⁷ by again assuming that the 27% load factor applied to each individual user.

Pumping costs attributable to distribution are developed from estimates by Transflux for PSE&G¹⁸, based on \$69/MWh. Of the 65¢/mmBtu total pumping bill,

	Capital	Peak Load	Load Factor	
			0.32	0.22
	'81M\$	mmBtu/hr	'92\$/mmBtu	
Jersey City	29.3	300	5.80	8.50
Newark	30.7	360	5.20	7.50
New Devel.	18.0	300	3.80	5.60
Weighted Average:			5.20	7.60

Table 3-3: Distribution Costs from PSE&G Study.

18¢/mmBtu is attributable to distribution in the lines and 19¢/kWh in the customer heat exchangers. Since this cost is energy-related, it does not scale with load factor. The cost range is then developed by assuming (as in Section 3.3) that the likely variation in load factor is 20%, from 22 to 32%, and that the costs will scale in inverse proportion to this. The results are in the rightmost two columns of Table 3-3.

3.5 Distribution Costs - Reconciliation

Even for steel pipe, PSE&G's distribution cost estimates are higher than those from the BNL generic model by a disturbing margin, and the difference shows up in a variety of places: PSE&G's heat exchangers cost twice as much, their pumping energy requirement is much larger, and so forth. It is true that one of PSE&G's consultants felt that their prospective costs were high, but with three estimates lying close together and only one proposing much more optimistic numbers, the PSE&G estimates seem well-founded.

On the other hand, the fact that when PSE&G were conducting the study there was some chance that they would actually end up building the system could have led them to err on the side of prudence, so as to avoid the possibility of financial embarrassment later. Also, the PSE&G results are based solely on steel pipe, and the FRP systems do stand a reasonable chance of being accepted in the smaller diameters and pressures appropriate to distribution systems.

So in the end we will compromise, taking the average of the BNL and PSE&G estimates for steel pipe, and ignoring the BNL estimates for FRP not because FRP may not be viable, but because we give more credibility to PSE&G's estimates for site work and all other non-pipe items. To get the projected unit price range, as before, we take the mean of the BNL estimate ($\$3.15 \pm 1.20$) and the mean of the PSE&G weighted average ($\$6.40 \pm 1.20$), average them and compute the root mean square variation of that average ($\$4.80 \pm 2.20$) to generate our estimate:

Final Estimate of Unit Distribution Costs: \$2.60 - \$7.00 per million Btu delivered.

4. DISTRICT COOLING

So far in this report we have discussed district heating systems without including the district cooling option, and have found promising economics to be somewhat elusive. In principle, district cooling could improve the situation, since by making greater use of the capital-intensive piping system, it would allow the fixed costs to be spread over more delivered energy services. In this chapter we examine this possibility in some detail; in short, we find that district cooling may be worthwhile as an add-on if district heating is itself viable, but that it cannot make non-viable district heating viable.

To analyze district cooling accurately, several distinct possibilities must be broken out and examined separately. In the first case, the district heating system is operated as it would be in the pure heating case, with pressurized hot water supplied at 220-250°F (105-120°C) and used to operate absorption chillers at the customer site; we call this "distributed district cooling". Alternatively, large absorption chillers can be operated off reject heat in proximity to the generating station, and cold water then distributed by a separate set of pipes to the customers. We call this "central district cooling", and examine in detail four-pipe systems capable of supplying both hot and cold water simultaneously; we briefly discuss two-pipe systems which are suitable only for district cooling.

We omit central district cooling systems based on mechanical heat pumps because we are interested only in increased use of reject heat from electric utilities. Various forms of "pulse" systems have been suggested, where the same pipes are used first for hot water, then for cold water on a cycle of a few hours or seasonally. The systems operating on short cycles will have high losses associated with heating the entire piping system up and down, and the annual cycle systems cannot deliver domestic hot water in the summer. To our knowledge, no such system has been demonstrated, and for these reasons we doubt that one will be.

4.1 Cost of Electric Compression Cooling

First, consider the competition: any form of district cooling must have overall costs comparable to or lower than electric compression chillers. There are three components to these overall costs, associated with electricity, capital equipment and maintenance, and if these are calculated as costs per unit of heat removed from the conditioned space, they may be added to find the overall costs. We will look at two size ranges: a 20-25 ton unit appropriate for a modest building, and a 400 ton unit sized to a large commercial or residential building.

Electricity in the metropolitan areas suitable for district heating costs from 12-16¢/kWh. New electric compression air conditioning systems in the twenty ton range, with dry evaporators, have seasonal energy efficiencies ("SEER") of 14 Btu/Wh, or a COP of 4.1. The 400 ton systems, with wet cooling of the evaporator, have SEER's of up to 18 Btu/Wh, for a COP 5.3. The resulting energy related costs of cooling are then \$8.60-11.40 per mmBtu cooling of the conditioned space (mmBtu_{clg}) for the three ton unit and \$6.60-8.80/mmBtu_{clg} for the 400 ton unit.

Capital costs are taken from an earlier BNL study (CRAC-1)¹⁹ and a study by Arthur D. Little for the Gas Research Institute (ADL/GRI)²⁰. For a 25 ton unit, Table 4-2 of CRAC-1 gives an installed capital cost of \$18,900; converting from 1984 to 1992 dollars with the producer price index (App. A), dividing by the capacity, assuming 2200 hours per year of operation and converting units, we find a capital-related cost of \$3.50/mmBtu_{clg}. For the 400 ton unit, the cost of the separate cooling tower must be included. Following the same procedure, we find \$1.80/mmBtu_{clg}; for the same size machine, Figure 2-2 of ADL/GRI gives \$2.00, and these two values are used as upper and lower limits in Table 4-1.

Maintenance costs are derived from CRAC-1 and treated similarly, but use the "50-50 factor" from App. A for price escalation. Because they are considerably smaller and are not given in ADL/GRI, we did not seek out corroboration. The results are summarized in Table 4-1, where the totals were obtained by combining uncertainties as discussed in App. D.

	Cost in 1992\$ per mmBtu cooling	
	20 ton	400 ton
Energy-related costs	8.60 - 11.40	6.60 - 8.80
Capital-related costs	3.50	1.80 - 2.00
Maintenance	2.40	0.50
Total:	14.50 - 17.30	9.00 - 11.20

Table 4-1: Unit Costs of Cooling - Electric Chillers.

4.2 Cost of Absorption Chillers

We will consider only lithium-bromide based, steam or hot water driven units. These are currently only available in larger sizes, from 20 to 400 tons, since smaller units have not been cost-effective and are no longer made. System characteristics and data will be taken from the same previous BNL study, CRAC-1¹⁹ and from ADL/GRI²⁰. We will derive costs for both single-effect machines, with COP's of 0.70, and the more expensive double-effect machines, with seasonal COP's of 1.2. Single effect machines can operate effectively with source temperatures as low as 200°F (93°C), while double effect machines normally require a minimum of 300°F (150°C) steam or pressurized water. This is greater than the nominal distribution temperature assumed for the district heating system, and the cost of supplying this higher temperature energy must be accounted for.

Since single-effect absorption chillers have COP's of about 0.7, and the range of costs of thermal energy from Table 1-3 above for condensing turbines is \$7.80-12.80/mmBtu, the cost for thermal energy removed from the room will be \$11-\$17.40/mmBtu_{c1g}, used in Table 4-2.

	Cost in 1992\$ per mmBtu removed	
	Small	400 ton
Energy-related costs	11.00 - 17.40	11.00 - 17.40
Capital-related costs	5.10 - 11.60	2.30
Maintenance	2.40	0.50
Total:	20.40 - 29.50	13.80 - 20.20

Table 4-2: Costs of Cooling - Single Effect Absorption Chillers.

For double effect machines we must take into account the required temperature of 300°F, 80°F higher than planned for the system described in Section 2.1. Determining the cost of 300°F thermal energy from this condensing turbine is too case-specific to be carried out here. We will use the fact that

at 6.9¢/kWh, Fig. 2-1 above indicates that the cost of thermal energy from a backpressure turbine increases at 2.1¢/mmBtu-°F, serving as a lower limit to costs from a condensing turbine. Whenever chillers are running, then, the system must supply thermal energy at 300°F, at a cost increment of at least \$1.70/mmBtu and a total cost of at least \$9.50-14.50/mmBtu. Since the COP of the double effect chillers is 1.2, the cost of thermal energy will be at least \$7.90-12.10/mmBtu_{clg}, shown in Table 4-3.

Capital costs derived from CRAC-1 form the upper limits in Tables 4-2 and 4-3. They are based on manufacturer's equipment costs in Figure 2-2 of CRAC-1 added to installation and cooling tower costs from Table 4-2 of CRAC-1. Costs for 20-25 ton machines are referred to as "small", and a 400 ton capacity was used at the upper limit. Although machines sized up to 1000 tons are available, economies of scale are modest above 400 tons, and the unit costs would not decline greatly for larger machines. Again, 2200 hours of operation per year were assumed, and costs were escalated with the capital goods factor of the PPI and annualized at 10% per year. The ADL/GRI estimates came out consistently lower, and we feel this range represents real uncertainty in this volatile market.

The maintenance estimates are from CRAC-1 and were escalated with the "5-0-50 factor". The total costs in Tables 4-2 and 4-3 again approximately represent 67% confidence intervals on the actual range of possible costs - that is, of all the systems that could reasonably be constructed, 67% of them would end up with costs in this bracket.

	Cost in 1992\$ per mmBtu removed	
	Small	400 ton
Energy-related costs	7.90 - 12.10	7.90 - 12.10
Capital-related costs	6.90 - 14.40	2.90 - 3.30
Maintenance	2.40	0.50
Total:	18.70 - 27.40	11.50 - 15.70

Table 4-3: Costs of Cooling - Double Effect Absorption Chillers.

Comparison of Tables 4-2 and 4-3 with 4-1 indicates that absorption cooling, powered by thermal energy from cogenerated district heat, is likely to be cost competitive only in some restricted niche markets. However, we will examine each of the possible deployment schemes in turn to see if the situation can be improved. Of course, if absorption chiller costs are lowered significantly^{19,21} or efficiencies increased significantly²², the situation would have to be re-assessed.

4.3 Distributed District Cooling

The term "district cooling" is really a misnomer for this application, since there is nothing district-wide about the cooling. Thermal energy customers simply buy an appliance (an absorption chiller) which allows them to use centrally supplied thermal energy to cool commercial or residential space. One question we must ask is whether exercising this option significantly lowers the cost of the centrally supplied thermal energy through larger consumption.

To answer this we look back at Chapter 2 and the summaries of Chapter 3 found in Tables 1-2 and at the final cost of thermal energy in Table 1-3.

Assuming that the demand for cooling would come in summer, ideally no new pipes or hardware would be required, and load factors for the various components would go up, allowing fixed costs to be spread over more delivered Btu's. How much would the cost of thermal energy go down?

First, the electricity-related cost (payment for electricity not produced) would be unchanged, or would be increased when double effect chillers demanded even higher extraction temperatures. From Table 2-2, this amounted to \$3.96/mmBtu (of heat). Of the capital items, the extraction retrofit and the transmission line were both assumed to have relatively high load factors (62% and 70%, respectively). We generously assume these can be increased to 71% and 80%, corresponding to a drop of 15% in the resulting energy costs, to \$0.60-1.11/mmBtu for the retrofit (see Table 2-2) and to \$0.12-0.30/mmBtu for the transmission line (see Table 1-2). This leaves the distribution costs, with a load factor in the range of 26%. If we assume half of this is heat and half hot water, implementing district cooling could increase the load and load factor by 50%, corresponding to a 33% decrease in the fixed costs in Table 1-2 to \$1.74-4.66/mmBtu. Adding these quantities together using the rules of Appendix D, we get a new cost of thermal energy of \$6.70-9.70/mmBtu if single effect chillers enjoy substantial market penetration. Compared to the condensing turbine costs of Table 1-3, this is a significant drop, which would make the basic district heating system more attractive to potential customers for thermal energy.

For systems supplying double effect chillers, the incremental \$1.70/mmBtu for 300°F supply temperatures must be added, for a total cost of \$8.40-11.40/mmBtu. Thus despite the increased usage, thermal energy customers would see no significant change in costs, and their thermal energy would arrive at a higher temperature. (Lower supply temperatures in winter could perhaps lower costs somewhat.)

However, we have to ask if these prices make district cooling cost-effective, else the assumed increase in load factors will never take place. If we re-do the arithmetic behind Table 4-2 (single effect chillers) with the new lower thermal energy costs, we find the cost range for the small chiller shown on the second line of Table 4-4, while that for the double effect units, developed like Table 4-3, is shown on the third line.

	Cost in 1992\$ per mmBtu cooling	
	Small	400 ton
Single Effect Absorption:	18.60 - 26.40	12.40 - 16.70
Double Effect Absorption:	17.30 - 25.30	10.60 - 13.10
Distributed Electric Chillers:	14.50 - 17.30	9.00 - 11.20

Table 4-4: Costs of District Cooling - Distributed Chillers.

For ease of comparison the costs of distributed electric chillers is shown on the bottom line; we see modest overlap between the probable cost ranges of absorption chillers and the likely costs of cooling electrically only in the case of large double effect machines. We conclude that while the presence of distributed absorption chillers in a two-pipe district heating system would lower the costs of thermal energy from that system for single effect chillers, significant penetration is unlikely to take place. For double effect chillers, there may be some markets where they can compete if 300°F thermal energy is available, but their presence will not lower costs for the balance of the system.

These results are consistent with the results of an earlier BNL study of district heating and cooling¹³, which found that the incremental costs of distributed cooling, not including any charge for the heating-associated capital items, was narrowly competitive with electric systems. We have more realistically charged all customers for their share of the capital as described just above.

4.4 Central District Cooling - Four Pipe

In these systems, four pipes are laid in the transmission line rather than two, and two of them deliver and retrieve chilled water from each customer. Commonly, the water is delivered at 45°F (7°C) and returned at 55°F (13°C). Cold water is supplied at a central location by a large chiller, and in the case of reject heat utilization, this must be an absorption chiller located somewhere near the power plant. Considerations to be raised below will make it clear that the double effect chiller has great advantages with respect to efficiency and cost, and it is the only case we will examine. The question we must answer is whether the cost of supplying chilled water plus the cost of the extra piping is competitive with some mix of small scale and large scale distributed electric cooling, since some of the customers will be large buildings capable of their own economies of scale and others will be smaller.

We start with estimates of cooling costs for centrally located 400 ton machines, examining only incremental costs attributable to district cooling. Energy-related costs, transmission and distribution costs, and the capital costs associated with the chiller are displayed and totaled in Table 4-5.

For double effect chillers, the central location of the chiller provides a tremendous bonus: although the thermal energy must be extracted from the electric generator at 300°F, with the attendant incremental cost of \$1.70/mmBtu discussed above, that energy is released from the chiller at 250°F, having done its work there, and may then be directly shipped out in the hot water pipes to supply normal heating loads. The only energy-related cost for central double effect chillers is then the incremental \$1.70/mmBtu, or, at a COP of 1.2, \$1.42/mmBtu_{clg}, which we will call \$1.60/mmBtu_{clg} to include capital costs for the extra steam fittings at the turbine.##

For transmission and distribution costs, we ask what the cost of two additional pipes is at construction time. Examination of detailed costs for steel pipe¹⁵ indicate that the costs directly associated with the pipe (pipe, joining and fittings) constitute from 50% of the cost for small pipes in dense urban areas to 78% for large pipes in rural areas. Although the chilled water pipes may be larger or smaller than the hot water pipes, depending on site-specific conditions, we will simply charge a premium of 60% of normal transmission and distribution costs, amounting to \$1.70-4.40/mmBtu if the pipes were transporting hot water with a temperature drop of 50°F (28°C), as the hot water pipes are. For district cooling, however, the temperature drop is normally restricted to 10°F (6°C) by the danger of freeze-up at the low end and the need to dehumidify at the return temperature. To give the technology the benefit of the doubt, we will assume this can be pushed up to 12°F, but we must still increase the unit costs by a factor of four (50/12) to give \$6.80-17.60/mmBtu_{clg}, shown in Table 4-5.

Capital costs and maintenance for the double effect chiller are as in Table 4-3, and the total is compared with the costs of electric chillers. We see that there is a significant area of competitive overlap for small systems, but very

Distributed double effect chillers could reap the same benefit, but only if there were an exact temporal match between thermal and chilling loads at the individual sites; load diversity makes this a near certainty for the central chillers.

little chance that district cooling would be more cost-effective than large distributed chillers. This result is driven by the high distribution costs, and any advances that could lower these, such as slush ice distribution systems^{23,24} could have a significant impact.

Double Effect Absorption	Cost in 1992\$ per mmBtu cooling	
Electricity:	\$1.60	
Transmission & Distribution	6.80 - 17.60	
Chiller Capital:	2.90 - 3.30	
Maintenance:	0.50	
TOTAL:	12.00 - 22.80	
	Small	400 ton
Distributed Electric Chillers:	14.50 - 17.30	9.00 - 11.20

Table 4-5: Costs of District Cooling - Four-Pipe, Central Chillers.

It should now be clear why we omitted single effect chillers from consideration - they would use energy at 250°F (or less), but would have to pay the full price for it, since they would release it at too low a temperature for further distribution. (It might prove possible to make a cascade system work for single effect chillers, but the margins are sufficiently close that we must call it speculative.) The result would be costs significantly higher than those of the double effect chiller.

Similar problems apply to two-pipe systems dedicated to district cooling alone. The cold water delivery must now pay all the costs of the transmission and distribution systems, still multiplied by a factor of four relative to heat delivery lines, and the result will be extremely high costs per unit of delivered cooling energy.

5. AVAILABLE REJECT HEAT

Using the costs and characteristics of reject heat and thermal energy transmission systems developed earlier, we are now in a position to estimate the quantities of thermal energy available from utility owned power plants which could in principle be used for district heating systems. Overall estimates of district heating potential have been made more than once, and we will not attempt anything so detailed here.

For reference, studies at Argonne National Laboratory²⁵, cited by the National Academy of Science²⁶ studied scenarios where the annual energy to be supplied by district heating in the early 2000's ranged from 0.4 to 3.5 quads⁺⁺, depending on the amount of government support the technology received. This corresponds to 12 to 100 systems the size of current large networks, discussed in Chapter 6, below. Although significant amounts of thermal energy would come from peaking boilers, geothermal springs, heat pumps and other sources, the bulk would be supplied by cogeneration, owned either by utilities or independent power producers.

Rather than attempt to make estimates at this level of sophistication, we will look at a selected set of utilities that would appear to have appropriate markets and adequate supplies of reject heat. For these, we will estimate from Department of Energy data how much reject heat is likely to be available from generators located in urban areas served by these utilities. What we find, in each case, is that there is a very large amount of thermal energy being discarded. The conclusion is uniform (and probably not surprising): the failure of electric utilities to construct or expand cogeneration-based district heating systems has little to do with availability of the energy.

5.1 Condensing Steam Turbine Generators

These are the workhorse units for electric generation from fossil fuels. (They are also used in nuclear plants, but we omit these from consideration.) We include only units with more than 100 MW summer electric capability, and assume that an amount of thermal power equal to the electric power can be extracted at reasonable cost. (This is slightly more optimistic than the detailed estimates of Section 2.1) Fossil-fueled steam-electric plants averaged a 48% capacity factor in the U.S.²⁷, so each megawatt of electric capacity corresponds, when averaged across all units, to 1.43×10^{10} Btu per year.

Of course it is wildly optimistic to assume that all of these plants can be retro-fitted for cogeneration - poor location, on-site space problems, transmission corridors and a host of other problems are likely in any particular case. Nonetheless, we will calculate this "theoretical potential" for several electric utilities of interest, discussed in detail in Section 6, below. The results, more of an upper limit on available thermal energy than an estimate of practical extraction, are in Table 5-1.

An inventory of a large eastern utility's (LEU#1) in-city steam plants²⁸ indicates a capacity of over 4400 MW; this implies the availability of 15,000 mmBtu/hr, or 6.3×10^{13} Btu per year at a 48% capacity factor. Since they actually have a peak thermal capacity (including directly fired boilers) of about 12,000 mmBtu/hr²⁹, they are making substantial use of the resource.

A large west coast utility (WU) has almost 600 MW of steam turbine capacity in urban environments, and in neighboring counties another 3280 MW. Where transporting the thermal energy from suburban plants into urban areas would

⁺⁺One "quad" = 10^{15} Btu; the U.S. consumed 81.5 quads in 1991.

require several miles of transmission line, we drop it from consideration. We also omit stations in smaller cities. The available urban resource is still impressive.

Large Eastern Utility #2 (LEU#2) has 2200 MW of steam capacity in densely settled industrial, commercial and residential areas, almost all of which is classified as urban. (They also have over 1000 MW in less densely settled areas, which we ignore.) Their examination of the possibility of developing district heating is discussed in Section 6.3 below.

Large Eastern Utility #3 (LEU#3) has 350 MW of steam capacity in densely populated areas and 1900 MW in a mixed urban/suburban area. Because we found in Section 3.3 that distribution costs did not depend strongly on density (if the density were above critical levels that are exceeded here) we will use the total of 2250 MW for this utility.

Our last example, eastern utility # 4 (LEU#4) has 500 MW in urban areas and over 2000 MW in neighboring suburbs. The suburban area is fairly densely populated (although less so than EU2), but it is also uniform, without large load centers; since we are searching for easily exploited resources, we will include only the urban resource.

Table 5-1: Condensing Turbine Thermal Resource at Selected Utilities.

	Summer Capability	Thermal Power	Thermal Energy
	(MW)	(mmBtu/hr)	(10^{12} Btu/year)
EU#1	4416	15,000	63.4
WU	584	2000	8.4
EU#2	2200	7500	31.6
EU#3	2250	7680	32.3
EU#4	480	1640	6.9

5.2 Backpressure Steam Turbine Generators

Since these units are only employed in systems where there is a demand for thermal energy at temperatures significantly above ambient, they are very rare in larger sizes in electric utility installations, and do not constitute a statistically significant source of currently unused thermal energy. Some may have been constructed as part of a district heating system, or some site-specific installation involving a market or local use for the thermal energy, but if they are still operating as such, they are not a source of unused thermal energy, and if they have been de-commissioned, it is likely there is some technical problem or major inefficiency associated with substantial age. Searching through the references listed at the end of this report, we have not found any evidence of significant utility use of backpressure turbines, and accordingly drop them from further consideration.

To avoid confusion, it is worth noting that PSE&G³⁰ had decided it was worthwhile to add backpressure turbines to an existing boiler-condensing turbine system, to provide an efficient route for thermal energy, extracted in the form of high pressure steam, into the pressurized hot water of their proposed district heating system. Had they had any backpressure turbines available, this step might not have been necessary.

5.3 Combustion Turbine Generators

The discussion of Section 2.4 showed that heat from combustion turbines could cost \$0.50 - 1.00/mmBtu, a very attractive price. However, that calculation was based on the assumption that the unit was used regularly, at a 70% capacity factor. The actual amount of heat available is small and would generally be considerably more expensive since most of the turbines are used as "peaking units". That is, they are run only a few hours a day, on an "as needed" basis, to meet peak loads. In fact, the national average for gas turbines is below 5%³¹, which leads directly to an annualized cost which is $70/5 = 12$ times larger, or \$6-12/mmBtu, more expensive than any of the options discussed in Chapter 2 and probably prohibitively expensive.

In addition, the heat would only be available at times of peak electric load, which might or might not correspond with peak thermal loads, depending on geography and many other factors. Storage could be added, but storage on the scale needed for district heating is moderately expensive, and regular, diurnal use of combustion turbines is rare. Storage of capacity adequate to match the actual irregular use of turbines to thermal loads would have to be substantially larger than for regular diurnal cycles, would be proportionately more expensive and would have a proportionately lower duty cycle, making delivered unit energy costs quite high.

The combination of high actual costs and irregular availability seems to be an insurmountable wall to greater deployment of combustion turbines at this time. Consequently, they will not be considered further as a source of heat for district heating and cooling despite the seductively low apparent price.

5.4 Reciprocating Engine Generators

There were about 2900 reciprocating prime movers in use by U.S. electric utilities in 1989/90³², with a net capability of 4600 megawatts; thus the average size was about 1500 kW, and almost all were at least a few hundred kilowatts. They were equally divided between units that could burn only fuel oil, and dual fuel systems that could switch to natural gas. Using the estimates of Table 2-3, this corresponds to 4.2 mmBtu/hr for the average 1500 kW unit, or 26,000 mmBtu per year at a 70% capacity factor. This is enough to provide heat to 200-300 residences, a sort of "micro" district heating system consisting only of local distribution piping. As an upper limit, if all 2900 units were integrated into such systems, they would provide almost 1/10 of a Quad per year.

However, we found no evidence that any of these units were being used for cogeneration. Also, generation statistics³¹ indicate that the capacity factor for these diesels was actually 3-4%, considerably lower than that for combustion turbines. Essentially, these systems are used only for peaking or in emergencies and consequently, they cannot be regarded as significant source of reject heat at present.

Perusal of Table 20 in Ref. 32 indicates that the bulk of these systems belong to small municipal utilities who are currently purchasing most of their power from some electric wholesaler, such as a major investor owned utility or a power authority. Even assuming that they do this because it is economically optimal, a few case-by-case studies might be merited to see if the addition of district heating could turn the economics around. The ease with which water jacket heat can be recovered, and the flexibility that is possible within a small organization argue that these "mini-utilities" might be a promising place to develop community-sized energy systems, since the prime movers are already present. Attempts to contact the utilities led to incredulity or complete lack of interest, and time constraints have prevented us from pursuing this question further.

6. REASONS UTILITIES DO NOT FULLY EXPLOIT THE RESOURCE

Here we report on the results of direct contact with several utilities who were found in Section 4 to have large, unexploited thermal resources and a substantial market in proximity to those resources.

To anticipate the results, the largest single barrier was always high capital costs, both for piping and for retrofit, which should come as no surprise at this point in the study. Other frequently mentioned problems were the cost of replacement electricity, the technical complexities of operating turbines after the retrofit, the difficulty of convincing customers to compare the cost of district heat with the sum of the fuel, operations and capital expenses associated with their individual systems, rather than just the fuel cost, and inadequacies in the EPA's emissions policies. The district heating enthusiasts we spoke with all expressed a certain sense of isolation within their company on this issue, and indicated uniformly that top management was just not interested.

In several cases the spokesperson requested anonymity for themselves and the company, and we have honored that request by withholding the names of all the utilities. We have also adjusted data slightly to make these utilities less recognizable, while still presenting the points the respondent wished to make.

6.1 Large Eastern Utility #1 (LEU#1)

LEU#1 operates the a large district heating system, feeding up to 12 billion Btu's per hour to some 2000 customers in the form of steam at various pressures - low for building heating and domestic hot water and higher pressures for various commercial and industrial operations. Of this, about 55% is produced by cogeneration and the remainder from steam-only plants scattered around the city. As discussed in Section 5.1, they is making better use of its reject heat than other U.S. utilities. Nonetheless, they still reject a comparable amount of heat to rivers and other sinks. Why don't they expand their district heating system, either with steam or with hot water? The customers are there, if the thermal energy is appropriately priced.

Price is certainly a serious problem. LEU#1's rates now run from about \$20 per thousand pounds of steam for "small" users down to \$10 per thousand pounds for "large" users. (Note that even a "small" user will comprise a multi-apartment or equivalent commercial structure.) This works out to \$17 and \$9 per mmBtu, respectively. In a city where fuel oil costs \$1.10 per gallon, or about \$8 per mmBtu, the cost savings associated with not having to purchase or maintain a furnace or boiler are completely overwhelmed by the price differential for small customers. Unless they can lower their rates, it is unlikely they can develop the customer base further.

A discussion with a company official indicated that lowered prices are very unlikely, since they consider the primary problem with expanded district heating to be very high capital costs, especially for pipe. As a ballpark figure they find it costs \$1800 per foot to put pipe in the ground. They were aware of a private developer in the metropolitan area who put a transmission line across an empty park, and it still cost \$1000 per foot.

It is hard to know if anecdotal cases like this are the norm or the exception. Our estimates in Section 3 ran from \$250-500 per foot in rural areas and \$400-900 per foot in urban areas for large (16-36 inch) pipes. In any event, there is no question that they see piping costs as a large and perhaps insurmountable barrier.

The market is somewhat inelastic, so it is also unlikely that they will lose large numbers of customers quickly. This occurs because decisions made when large buildings are constructed can be difficult to reverse later. Many large

buildings have no boiler room and no place to put one in; they were designed around the steam supply. Many have substantial investments in steam-driven absorption chillers, and even where it is technically feasible, they are unlikely to remove the chiller until near the end of its useful life. The LEU#1 representative corroborated this analysis by emphasizing that their greatest need was for better maintenance technology, such as re-insulating pipe which is already buried and other in-situ repairs.

Although LEU#1 does not operate incinerators, they have stayed abreast of what has happened at local installations that attempted to use a district thermal distribution system and here, too, they think the capital costs are overwhelming for large systems. Various other problems such as community resistance and variations in supply add to the difficulties.

So this representative of an existing large district heating system has concluded that the future of district heating belongs to small, localized, private systems driven by modest cogenerators! Even then, he felt the economics were difficult enough to require a case-by-case feasibility study.

6.2 West Coast Utility (WU)

A representative of this utility reported that while they do have a district steam system serving some 200 downtown buildings, it is not driven by cogeneration but by separate steam-only boilers. The system is profitable, but management is interested in selling it since it is such a small part of their business. They do actively recruit customers, with brochures and planning assistance, but see very little prospect for growth. He saw several other barriers to expansion, including (again) the expense and difficulty in laying pipe and the question of cost allocation: it must be very clear that the heating system is self-sustaining, since no part of its cost can fairly be charged to electric customers. PG&E is definitely planning electric expansion, but there is no interest in cogeneration at new plants.

6.3 Large Eastern Utility #2 (LEU#2)

This large eastern utility has examined district heating in the past but has never entered the market. An acknowledged district heating advocate within the company said that management was dissuaded by the high capital costs, but that if he were mounting a campaign for DH today, he could make a strong case for lower first costs than were found in past estimates for several reasons, including using European-style pipe, which is thinner-walled and less expensive than is traditional in the U.S. He had no knowledge of the FRP pipe discussed in Section 4, above. He also felt that contractors were being cautious in their estimates of pipe installation costs, and that a little bargaining could lower the costs significantly below values based on formal and abstract quotations from the same contractors. As an example, he offered the Jamesport, NY, system, which he asserted had been constructed at a cost effective price, well below the costs his company had predicted for comparable installations. (Jamesport is also discussed in Section 6.6, below.)

Aide from high capital costs, he recognized the cost of replacement electricity as a significant problem. He also commented on the inadequacy of the EPA's "bubble" policy, which does not give the utility any emission credit for the individual boilers shut off when district heating is provided despite a 3:1 reduction in overall emissions (see also Section 6.6 below).

The respondent admitted that he was in a small minority within the company with respect to his interest in district heating, and did not expect any developments without a significant change in circumstances.

6.4 Large Eastern Utility #3 (LEU#3)

LEU#3 makes no current use of reject heat other than to heat their own plants. When asked about barriers to greater cogeneration, the representative expressed concern about unbalancing the turbine/condenser combination, which was optimized for low temperature and low pressure conditions. (This is discussed at length in Section 2, above.) They are summer peaking, but have large winter loads as well, so the load curve would not by itself be a barrier to cogeneration. The representative gave the clear impression that there was no corporate interest in increased cogeneration or district heating at this time.

6.5 District Heating Utility (DHU)

Although not an electric utility, this east coast thermal energy company is in a sufficiently comparable position to make their experience directly relevant to an electric utility considering expanding their district heating market. They burn 2200 tons of trash per day collected by the city and so can be said to have "free" fuel. Operating expenses are of course far from free, since they must be more vigilant concerning emissions than a single-fuel plant, and the residual solid waste must be disposed of carefully.

The incinerator produces a maximum of 320,000 lb/hr (310 mmBtu/hr), but this capacity cannot now be used since the transmission line is limited to 250,000 lb/hr. They face a winter peak load of 700,000 lb/hr and a summer peak of 200,000 (presumably using peaking boilers for the difference), but due to rapid growth in internal heat sources and building insulation, they expect the summer cooling load to grow much more rapidly than the winter heating load.

When queried about expansion, the representative turned the discussion to the importance of low-interest loans for piping system expansion, mentioning the U.S. Department of Housing and Urban Development as a possible sponsor. Although he did not explicitly say so, his remarks seemed based on a sense that without some form of subsidy, the piping costs were too great to be competitive.

6.6 District Heating Developer

We also queried a well-known consultant on district heating both in the U.S. and Europe; he was interviewed in the early stages of this study and made three major points relevant to the future of district heating.

First, district heating systems do not now receive credit from the Environmental Protection Agency (EPA) for reductions in emissions that occur as individual boilers and furnaces are shut down. Thus they must produce the heat with essentially zero emissions, even though they must bring more plant on line to make up for the lower generation efficiency. This means more stack clean-up devices and greater expense (an expense not counted in the economic analysis of this report!).

As an aside, the authors of this report have heard this complaint independently raised by two other knowledgeable professionals³⁵ who asserted that district heating must meet strict industrial emissions standards when it is replacing commercial boilers having essentially no emission standards. Furthermore, the EPA will only give emission credit if the replaced boiler is physically destroyed, a step most owners are reluctant to take, since the boilers provide a level of back-up protection and cost money to remove.

Second, the developer asserted that the Public Utility Regulatory Policy Act of 1978 (PURPA) does not give independent power producers sufficient incentive to cogenerate, since it only requires an overall efficiency of 40.5%. Given combined cycle efficiencies approaching 40%, it takes very little heat

distribution to qualify under PURPA, so cogeneration is being pursued in a half-hearted manner in many cases.

Third, he mentioned the one million dollar retrofit of the Jamestown, NY, bituminous coal steam turbines to district heating as an example of the feasibility of cogeneration. However, because these are two 25 MW turbines, this argues more for the viability of small systems than large. Using the assumptions of Section 2, and a 25 MW(thermal) capacity, this gives a fixed cost contribution of only \$0.10/mmBtu, far less than the values derived in Table 2-2 for PSE&G's projected retrofit, but we have not been able to research the reasons for the difference. The system uses only 1.5 miles of transmission line, and this proximity to the load is another point in its favor.

REFERENCES

1. T. Baumeister, ed., "Marks' Standard Handbook for Mechanical Engineers", Eighth Edition, Ch. 9, McGraw-Hill, New York.
2. PSE&G (Public Service Electric & Gas Co.), "District Heating and Cooling Systems for Communities through Power Plant Retrofit Distribution Network"; DOE/CS/20071-1; January 1984.
3. U.S. Energy Information Administration, "Electric Power Annual - 1990", DOE/EIA-0348(90), Washington DC 20585, Table 40.
4. P.E. Hufford, "Absorption Chillers Maximize Cogeneration Value", ASHRAE Transactions, Vol. 97, Pt. 1, 428-433 (1991); "Absorption Chillers Improve Cogeneration Energy Efficiency", ASHRAE Journal, March 1992.
5. Baumeister, op.cit., pp. 9-62 through 9-64.
6. J. E. Cristian, "Gas-Steam Turbine Combined Cycle Power Plants", Oak Ridge National Laboratory, ANL/CES/TE 78-4, Oak Ridge TN 37830 (1978).
7. PSE&G, op. cit., Vol 7, Section 7.4, p. 36.
8. PSE&G, op. cit., Vol 6, pp 4,5. (21% of 8.8×10^{12} is 1.85×10^{12} .)
9. C. L. Segaser, "Internal Combustion Piston Engines", ANL/CES/TE 77-1, Oak Ridge National Laboratory, Oak Ridge TN (1977). (Avail: NTIS)
10. C. L. Segaser, "Heat Recovery Equipment for Engines", ANL/CES/TE 77-4, Oak Ridge National Laboratory, Oak Ridge TN (1977). (Avail: NTIS)
11. T. A. Butcher N. Park and W. L. Litzky, "Condensing Economizers: Thermal Performance and Particulate Removal Efficiencies", BNL 47391 (1992).
12. H. W. Brown and F. S. Stuber, "Reliability of Natural Gas Cogeneration Systems", GRI 93/0020; ARINC Research Corp. for The Gas Research Institute, 8600 Bryn Mawr Ave., Chicago IL 60631 (Sept. 1992).
13. P. D. Metz, M. Catan, M. Piraino, R. W. Timmerman and J. Gleason, "District Heating and Cooling Technology Selection and Characterization", BNL 52001, Brookhaven National Laboratory, Upton NY, (1986).
14. R. T. Coughlan, J. W. Andrews and P. D. Metz and J. Gleason, "District Heating Technology Characterization Case Study: Milwaukee, Wisconsin", BNL 52156, Brookhaven National Laboratory, Upton NY, (1988).
15. PSE&G, op. cit., Section 7, Vol. 4 and Section 6.5, Vol 3.
16. PSE&G, op. cit., Vol 4: Table 7-III (p. 6) and Section 7.6.4 (p. 47-48).
17. PSE&G, op. cit., Vol 4, pp. 37 ff.
18. PSE&G, op. cit., Vol. 4, pp. 47-48 and Table 8.3-V on p. 147.
19. R. W. Leigh, D. W. Huszagh, J. G. Murray, A. Ravve, J. H. Saunders, H. Teoh and W. G. Wilhelm, Cost Reductions in Absorption Chillers, BNL 38154 (1986).

20. Assessment of Large-Tonnage, Gas-Fired Cooling Technologies for the Commercial Sector, by Arthur D. Little for the Gas Research Institute, GRI-88/0162 (1989).
21. R.W.Leigh, H.S.Isaacs, J. Kirley, J.G.Murray and A. Ravve, "Cost Reductions in Absorption Chillers, Phase II", BNL 42502 (1989).
22. R.C.DeVault, "Triple Effect Absorption Chiller Utilizing Two Refrigeration Circuits", U.S. Patent 4,732,008 (March 1988).
23. P. J. Winters and R. J. Kooy, "Direct Freeze Slurry District Cooling System Evaluation", International District Heating and Cooling Association, 82nd Annual Conference, San Francisco CA (1991); Chicago Bridge and Iron Technical Services Co. Research Center, Plainfield IL; CBT-5557.
24. R.W.Leigh, "Economic Assessment of Thin Polymer Ice Maker", BNL-48103 (1992); "Thin Polymer Icemaker Development and Test Program: Final Report on Technology", BNL-61866 (1991).
25. A. Teotia et. al., "Cogeneration and District Heating Show Promising Market Potential..", Proceedings of the 16th International Energy Conversion engineering Conference, Atlanta GA 1981; American Society of Mechanical Engineers.
26. "District Heating and Cooling in the United States", Committee on District Heating and Cooling of the National Research Council, D.O. Meeker, Chair, National Academy Press, Washington DC 1985.
27. Energy Information Administration, "Electric Plant Cost and Power Production Expenses 1990" (Table 1, p. 4), DOE/EIA-0455(90), U.S. Government Printing Office, Washington DC 20402 (1992).
28. U.S. Energy Information Administration, "Inventory of Power Plants in the United States 1989" (Table 20, p. 164-166), DOE/EIA-0095(89), U.S.Government Printing Office, Washington DC 20402 (1990).
29. "District Heating and Cooling in the United States", Table 2-1, p. 29, Committee on District Heating and Cooling of the National Research Council, D.O. Meeker, Chair, National Academy Press, Washington DC 1985.
30. PSE&G, Ibid, pp. 42-43.
- 31.31. Energy Information Administration, "Electric Plant Cost and Power Production Expenses 1990" (Table 1 and p. 73), DOE/EIA-0455(90), U.S. Government Printing Office, Washington DC 20402 (1992).
32. U.S. Energy Information Administration, "Inventory of Power Plants in the United States 1989" (Table 5), DOE/EIA-0095(89), U.S.Government Printing Office, Washington DC 20402 (1990).
33. Mr. David Brewer and Mr. Ralph Cavanaugh, speaking in a discussion session at the 1991 International District Heating Association annual meeting, San Francisco.

APPENDIX A: INFLATION FACTORS

Financial and cost data used in this report was generated in reports written at various points in the past, using what was then contemporary data. To make all costs comparable and familiar, they have been escalated to the equivalent cost in 1992 dollars using indices from "Statistical Abstract of the United States - 1992"¹. Two sorts of costs are considered. For items that are clearly "capital equipment", such as steam turbines and heat exchangers, the "Producer Price Index for Capital Equipment" (Table 746) is used. The "PPI Factor" is the 1992 index divided by the index for the year in question.

For other items, such as installed transmission and distribution piping, labor constitutes a significant part of the cost, and the labor cost index (Table 657) for that year was divided into that for 1992 to obtain an "labor factor". The "labor factor" and "PPI factor" were then averaged with equal weights, and the resulting "50-50 Factor" was used to inflate piping costs. The calculations are displayed in detail in Table A-1.

Year of Data	PPI Index	PPI Factor	Labor Index	50-50 Factor
1977	66.1	1.96		
1980	85.8	1.51		
1981	94.6	1.36	71.0	1.49
1983	102.8	1.26	78.8	1.46
1984	105.2	1.23	82.8	1.31
1985	107.5	1.21	86.8	1.27
1987	111.7	1.16	93.1	1.20
1992	129.6	1.00	114.9	1.00

REFERENCE

1. Statistical Abstract of the United States 1992, U.S. Dept. of Commerce, U. S. Government Printing Office, Washington DC20402, pp. 476-477; New York Times, 9 April 1993 for 1992 indices.

Appendix B: Steam Turbine Heat Rates

----- ("Hg)	Pc (psia)	----- (psig)	Tc (oF)	TSR lb/kW	NHR from TSR	NHR Btu/kWh	(P-P')/Q' kWh/mmBtu	Cost \$/mmBtu	
1.5	0.7	-14.0	90			7920			
2	1.0	-13.7	101	5.633		7930	0.25	0.02	0.01
3	1.5	-13.2	114	5.819		8000	2.00	0.14	0.10
4	2.0	-12.7	126	5.963		8100	4.42	0.30	0.22
5	2.5	-12.2	133			8250	7.86	0.54	0.39
	19.7	5.0	227	7.713	10477	10477	42.37	2.92	2.12
	24.7	10.0	239	7.975	10833	10833	46.03	3.18	2.30
	34.7	20.0	259	8.421	11439	11439	51.54	3.56	2.58
	44.7	30.0	274	8.799	11952	11952	55.62	3.84	2.78
		115	381	11.06	15024				

Electric: 6.9 5.00 c/kWh

Marks' Standard Handbook for Mechanical Engineers - Ch. 9

(T. Baumeister, ed.; Eighth Edition, McGraw Hill)

Theoretical Steam Rate (TSR) from Table 9-2

Net Heat Rate (NHR) from Table 9-4.

or scaled from NHR at 133F using TSR's

All data for 1000 oF, 2400 psig supply

Boiler efficiency (BE): 90%

Auxiliary Power (AP): 4%

Cost of Electricity: 6.9 c/kWh (PSE&G, p.112, and lower bound)

$(P-P')/Q' = (NHR'/NHR - 1)/((NHR'/3413)*(1+AP)-1+AP)$

Appendix C: Thermal Energy Transmission Costs from PSE&G

(All carbon steel pipe, polyurethane insulation, polypropylene cover)

All data from PSE&G study DOE/CS/20071-1 V3 (1984)

Air data from PSE&G study, DOE, 1993, 2007												
LdFctr:				PPI(92/83):		HtCst:		DelT:				
Our		70%		1.46		\$5.00 /mmBt		50 F				
PSE&G Optimised				Capital:		10% /yr						
Diam	Peak	Peak	AnnEn	CapCst	CapCst	HtLoss		Vel	PmpCst	--Total Cost--		Delta
(in)	(mmBtu/hr)		(E9Btu)	1983\$/ft	\$/yr-mi	mmBtu/ yr-ft	\$/yr-mi	ft/s	\$/yr-mi @ 0.069 \$/kWh	Our	PSEG	for \$5 or \$.15 /mmBtu
p.122				p.117		p. 119				Opt.	Opt.	
RURAL:												
3	20	6	37	83	63,598	0.17	4,356	11	37,324	2.869	4.859	0.120
4	30	10	61	95	73,234	0.19	4,990	10	35,715	1.872	3.405	0.082
6	60	22	132	120	92,506	0.23	5,966	10	46,763	1.098	1.556	0.045
8	100	37	225	145	111,778	0.25	6,468	9	55,477	0.774	1.183	0.029
10	160	57	349	170	131,050	0.30	7,814	9	62,291	0.577	0.954	0.023
12	260	79	484	195	150,322	0.34	9,002	9	65,305	0.464	0.958	0.019
16	320	139	854	245	188,866	0.39	10,349	9	75,967	0.322	0.525	0.012
20	470	212	1,299	295	227,410	0.53	13,913	9	85,994	0.252	0.364	0.011
24	800	299	1,835	345	265,954	0.59	15,602	8	110,009	0.213	0.342	0.009
30	1400	463	2,838	420	323,770	0.86	22,598	8	120,218	0.164	0.317	0.008
36	2000	661	4,054	495	381,586	1.24	32,762	8	162,981	0.142	0.291	0.007
URBAN:												
3	20	8	48	162	125,191	0.17	4,356	14	79,243	4.317	5.460	0.092
4	30	13	77	183	141,225	0.19	4,990	13	74,720	2.855	3.811	0.065
6	60	27	167	225	173,294	0.23	5,966	12	93,792	1.639	1.835	0.036
8	100	46	281	266	205,362	0.25	6,468	12	109,840	1.144	1.360	0.023
10	160	70	430	308	237,431	0.30	7,814	11	120,704	0.850	1.077	0.018
12	260	96	588	350	269,500	0.34	9,002	11	124,839	0.686	1.044	0.015
16	320	167	1,026	433	333,637	0.39	10,349	11	144,767	0.477	0.598	0.010
20	470	251	1,540	516	397,774	0.53	13,913	10	161,198	0.372	0.429	0.009
24	800	357	2,187	599	461,911	0.59	15,602	10	202,451	0.311	0.389	0.007
30	1400	543	3,332	724	558,117	0.86	22,598	10	220,238	0.240	0.350	0.007
36	2000	784	4,809	849	654,323	1.24	32,762	10	293,815	0.204	0.315	0.005

Costs of Rural Thermal Transmission

APPENDIX D: MANIPULATION OF UNCERTAINTY AS DATA ARE COMBINED

Throughout this report various quantities, usually costs, have appeared which are the sum, difference, product or quotient of primary quantities which are themselves uncertain. For example, we might say the energy-related costs of thermal energy are \$5.00-8.00/mmBtu while the capital equipment-related costs are \$3.00-5.00/mmBtu. The total cost must be found by adding these ranges together somehow. In this appendix we point out the complexities involved in this addition and present our method for dealing with them.

First, what do we mean by offering a range of values? We are discussing district heating somewhat generically, and we mean that while the energy-related costs might be as low as \$5.00/mmBtu in Texas, where gas is cheap, they could be as high as \$8.00 in Maine, where fuel is more limited. We do not even mean that these numbers are absolute limits: if a utility owns its own gas well, the cost could be well below \$5.00; in a region under severe air pollution constraints, the costs could be higher than \$8.00 when sulphur removal and abatement are taken into account. Thus \$5.00-8.00/mmBtu is a range of costs which will be found for most, but not all, projects of the sort being examined.

In order to manipulate these quantities, we will pretend to understand their statistics better than we actually do: we will assume that the costs are distributed according to a Gaussian probability curve¹, and that the numbers quoted are the mean minus one standard deviation and the mean plus one standard deviation. There is then approximately a 68% probability that any given project will find costs within the range, a 16% probability that it will lie above it, and 16% that it will lie below.

We can now apply the theory of errors¹ to determine how to manipulate these quantities. We let

ℓ = lower limit and
 u = upper limit;

then

$m = (u + \ell)/2$ = mean, and
 $s = (u - \ell)/2$ = standard deviation.

Clearly $s = 0$ if $u = \ell$, as one would expect.

Now, with several uncertain quantities (ℓ_1, u_1) , (ℓ_2, u_2) , (ℓ_3, u_3) , ... to be added (or subtracted), find the m_i and s_i for $i=1,2,3...$. Then the mean and standard deviation of the sum, M and S , will be given by

$M = m_1 + m_2 + m_3 + \dots$, and

$S = \sqrt{s_1^2 + s_2^2 + s_3^2 + \dots}$.

If some of the quantities are to be subtracted, or are negative, the sign will affect the answer for M , but will have no effect on S , since it vanishes in the squaring operation.

For the example of the first paragraph, $\ell_1 = 5.00$, $u_1 = 8.00$, $\ell_2 = 4.00$ and $u_2 = 5.00$. $m_1 = 6.50$, $s_1 = 1.50$, $m_2 = 4.00$ and $s_2 = 1.00$. Then $M = 10.50$ and $S = 1.80$ (using a "√" key on a calculator). The limiting values will then be $L = M - S = 8.70$ and $U = M + S = 12.30$, so the range of the sum will be \$8.70-12.30/mmBtu.

Sometimes similar quantities must be multiplied and divided. In this case, the framework is the same, but the error is the "root-mean-square sum" of the fractional errors, made up of the standard deviations:

$M = m_1 * m_2 * m_3 * \dots$, and

$$S = M * \sqrt{(s_1/m_1)^2 + (s_2/m_2)^2 + (s_3/m_2)^2 + \dots}.$$

If division is involved, the formula for M must be modified in the obvious way, but that for S is unchanged.

Barford's book (see footnote) gives a complete explanation of the theory of errors. We may be presumptuous in assuming that we know enough about our data to manipulate it as if it were truly on a Gaussian distribution, but it is far better to have some recognition of the uncertainty intrinsic in assessments of this kind than to present one number answers, with little or no discussion of the accuracy with which they are known, as is so often the case.

REFERENCE

1. N. C. Barford, "Experimental Measurements: Precision, Error and Truth", John Wiley & Sons, New York, 1985.

DATE

FILMED

4/29/94

END

[REDACTED]

[REDACTED]