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# TECHNICAL FEASIBILITY AND ECONOMICS OF RETROFITTING AN EXISTING NUCLEAR POWER PLANT TO COGENERATION FOR HOT WATER DISTRICT HEATING

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## FOREWORD

This study was undertaken to help define the possible role of nuclear power plants as a source of heat for district heating systems in nearby cities. It was inspired largely by the rapid development of hot-water energy transport in Europe, in which water at the temperature required for district heating is transported economically in pipelines for distances of several tens of miles. Perhaps the most ambitious example of this was the proposed transport of hot water from the Forsmark nuclear station in Sweden, to Stockholm, a distance of 120 km. A number of smaller projects have been built and are in service throughout northern and central Europe.

A number of nuclear power plants in the United States are sited reasonably close to potential heating loads, and the U.S. Department of Energy became interested in the technical and economic feasibility of using U.S. nuclear plants as a source of thermal energy as well as electricity. Among the incentives for doing this, of course, would be to reduce the need for expensive fossil fuels and to reduce the emission of combustion products into the atmosphere.

The approach taken was to study a representative city with a need for district heating, and with one or more nuclear power plants nearby. Minneapolis/St. Paul was a natural choice, since a comprehensive study of district heating for the Twin Cities was already underway, and the two-unit Prairie Island nuclear station was in service on a convenient site about 58 km (36 miles) from the center of St. Paul. Thus, this study, while completely independent, was able to make use of the relevant data being generated in the DOE-sponsored "District Heating/Cogeneration Application Studies for the Minneapolis-St. Paul Area." Another plus for this area was the generous cooperation of Northern States Power Company, owner and operator of the Prairie Island station, who provided assistance in the study of the power plant modifications and changes in operation, and participated in the economics portion of the study. The organization of the study is described further in the Introduction. For additional information on the history and development of nuclear power as a source of thermal energy, refer to Appendix A.

## CONTENTS

## CONVERSION FACTORS

| To convert from | To* | Multiply by |
|---|---|---|
| lb/h | kg/s | 0.0001260 |
| $ft^2$ | $m^2$ | 0.09290 |
| in. | cm | 2.5400 |
| MBtu or $10^6$ Btu | GJ | 1.055 |
| Btu/h | kW(t) | 0.0002931 |
| MBtu/h or $10^6$ Btu/h | MW(t) | 0.2931 |
| psi | kPa | 6.895 |
| °F | K | $T_K = [(T_{°F} - 32)/18] + 273$ |
| $/MBtu | $/GJ | 0.9479 |

*Prefixes are used in the SI system to form decimal multiples of the base units (factors of $10^3$): $k = 10^3$, $M = 10^6$, and $G = 10^9$.

## LIST OF ABBREVIATIONS AND ACRONYMS

| | |
|---|---|
| A&E | architect & engineer |
| AEH | analog electrohydraulic |
| AFDC | allowance for funds during construction |
| BPT | back pressure turbine |
| CCGT | closed-cycle gas turbine |
| CV | control valve |
| DHHX | district heating heat exchangers |
| FCR | fixed change rate |
| FVB | Fjarrvarmebyran |
| HP | high pressure |
| LP | low pressure |
| LWR | light water reactor |
| MSR | moisture separator reheater |
| NSP | Northern States Power |
| PWR | pressurized water reactor |
| TCF | transmission capacity factor |
| TPC | total project cost |
| TTD | terminal temperaure difference |
| TV | throttle valve |

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## ACKNOWLEDGMENTS

This study involved the cooperative effort of several engineering organizations who performed the various technical and economic studies. The study was managed during the first year through the Midwest Office of KVB, Inc., Minneapolis, by Michael Barnes and Bart Murphy—now of Scantec, Inc. The Westinghouse turbine and plant modification study was directed by John D. Rogers with technical contributions from George J. Silvestri, Jr., J. H. Donegan, J. A. Polacheck, R. D. Brown, and R. G. Slebodnik; the Fluor Power Service, Inc., contributions to this study were by Charles E. Agan and Joseph J. Lula. The transmission system design study was directed by James J. Powers of Metcalf & Eddy, Inc., and Birger Abrahamson of FVB, with technical contributions from Mike Gilbert, Sue Pierce, and James Donn of Metcalf & Eddy, Inc., and Bjorn Anderson of FVB.

The study was greatly facilitated by the extensive support and technical assistance of the Northern States Power Company, who cooperated in the use of the Prairie Island Nuclear Plant as a model for the study and also provided technical and economic input and review by Tom Thomas, Lee Tuskey, Steve Caskey, and Tom Matson.

Acknowledgments are also due to W. Savage and R. G. Oehl of the Department of Energy and, from Oak Ridge National Laboratory, to M. A. Karnitz and T. E. Cole for their support and guidance, and to T. K. Stovall and G. D. Pine for their critical review of the report. L. F. Truett and Jacqueline Miller were the technical editors.

## ABSTRACT

This report gives the results of a study of the hypothetical conversion of the Prairie Island Nuclear Plant of the Northern States Power Company to cogeneration operation to supply a future hot water district heating system load in the Twin Cities of Minneapolis–St. Paul. The conceptual design of the nuclear turbine retrofitted for cogeneration and of a hot water transmission system has been performed, and the capital investment and annual owning and operating costs have been estimated for thermal energy capacities of 600 and 1200 MW(t). Unit costs of thermal energy (in mid-1982 dollars/million Btu) have been estimated for cogenerated hot water at the plant gate and also for the most economic transmission system from Prairie Island to the Twin Cities. The economic results from the analysis of the Prairie Island plant and transmission route have been generalized for other transmission distances in other locations.

# 1. INTRODUCTION

## 1.1 BACKGROUND

The improved utilization of fuel energy content brought about by the use of domestic-fueled, cogeneration power plants has received much study and attention in the United States, especially since the 1973 Arab oil embargo. Potential advantages in cogenerated thermal energy from domestic coal and uranium include (1) reduced dependence on imported oil, hence an improved national balance of payments and security of energy supply; (2) extension of the domestic reserves of the premium fossil fuels, oil, and natural gas; (3) reduction in long-term costs of thermal energy compared with the increasing costs from oil and natural gas; and (4) a reduction in the thermal discharge from a cogeneration power plant.

The retrofit of an existing nuclear power plant to cogeneration operation has additional advantages when producing thermal energy for large-scale, hot water district heating applications. These advantages are

1. reduced production cost for thermal energy from cogeneration operation,
2. reduced emissions of fossil fuel pollutants with the consequent reduction of local air pollution and regional acid rain, and
3. the ability to utilize nuclear energy more efficiently without requiring new plant sites.

However, this concept also has several inherent requirements, either economic or logistical, which would have to be met before a practical application could be feasible. First, a previous study[1] has shown that the high capital cost of the long-distance transmission system from a power plant to a district heating load center requires a large—>200-300-MW(t)—heat transmission capacity to have a potential for economic feasibility. Second, a large district heating distribution system must either exist or be planned to create a demand for the transmission system output. Third, the cost of cogenerated hot water from the nuclear plant must be as low as or lower than that from other available heat sources—either power plants or heat-only plants—that are closer to the load center.

These "real world" constraints are stated at the outset to establish a proper understanding of the motivation for this study. Since there is currently no U.S. metropolitan area with a large hot water district heating system in operation, the results of this study are intended to provide a generic technical and economic evaluation of retrofitting an existing nuclear power plant for a considerable time period into the future. Also, the study of a retrofitted nuclear plant heat source for the Minneapolis–St. Paul (Twin Cities) area complements other studies of fossil-fueled power plant heat sources for a large, metropolitan-scale district heating system performed for this area.

The remainder of this section presents a brief summary of nuclear power plant heating application studies, as a historical setting for this study, and also the specific background of this study in the Twin Cities area.

### 1.1.1 Nuclear Power Plant Heating Applications

Currently, the United States produces energy for heating from the Experimental Breeder Reactor-2. Russia produces heat for district heating from three nuclear reactors at Bilibino. In Canada, the Bruce heavy-water reactor (HWR) is used to supply thermal energy for separating heavy water. In Sweden, a small nuclear power plant was used for district heating from 1964 to 1974.

Many countries—Sweden, Switzerland, Finland, and West Germany—have plans and studies for the use of thermal energy from nuclear power plants for district heating. The obstacles to such developments seem to be the remoteness of most nuclear power plants, public opinion, and the vast size of most nuclear power plants. Descriptions of existing applications and studies are presented in more detail in Appendix A.

### 1.1.2 Twin Cities Study

This study, sponsored by the Department of Energy (DOE) was initiated by the Oak Ridge National Laboratory (ORNL) to explore with the Northern States Power (NSP) Company the feasibility of a hypothetical hot water district heating system serving the metropolitan Twin Cities area. Initially, the overall technical and economic feasibility of a hot water district heating system for the Twin Cities based on cogenerated thermal energy was examined under the sponsorship of the DOE in 1978-79 by the Swedish laboratory, Studsvik Energiteknik AB, in conjunction with NSP, the Minnesota Energy Agency, and ORNL.[2] With the results of the Studsvik study as a foundation (Sect. 2), a series of studies of cogeneration power plant applications was performed for an existing NSP coal-fueled plant, a new coal-fueled plant, and a new light-water reactor (LWR) nuclear plant. This study completes the assessment of the range of options for future sources of base-load thermal energy by evaluating the retrofit of the NSP Prairie Island Nuclear Plant (Prairie Island) to provide thermal energy for a hypothetical Twin Cities hot water district heating system.

The study was first proposed by a consortium of private firms led by the Midwestern Office of KVB, Inc., the initial project director. The technical analysis tasks were organized to utilize the expertise from specific firms that had prior experience with Prairie Island. The analysis of turbine and plant modifications was performed by the Westinghouse Electric Corporation, supplier of the nuclear steam supply system and the turbine-generators for Prairie Island with assistance from

Fluor Power Services, Inc., the architect-engineer for Prairie Island. Analysis of the hot water transmission system was performed jointly by a U.S. firm and a Swedish firm—Metcalf & Eddy, Inc., and FVB. These engineering consulting firms combine extensive experience with design and operation of U.S. water distribution and treatment systems and Swedish hot water district heating piping distribution and transmission systems. Finally, NSP performed the plant production cost allocation and also contributed to the review of the plant modification analysis.

ORNL served as the technical project manager for DOE throughout this study. In addition, ORNL assumed the role of project director when KVB withdrew from that role in 1981. ORNL subsequently performed the economic analysis with the assistance of NSP and was responsible for this final report. Figure 1.1 shows an overall task organization for this study.

ORNL DWG 83-15434

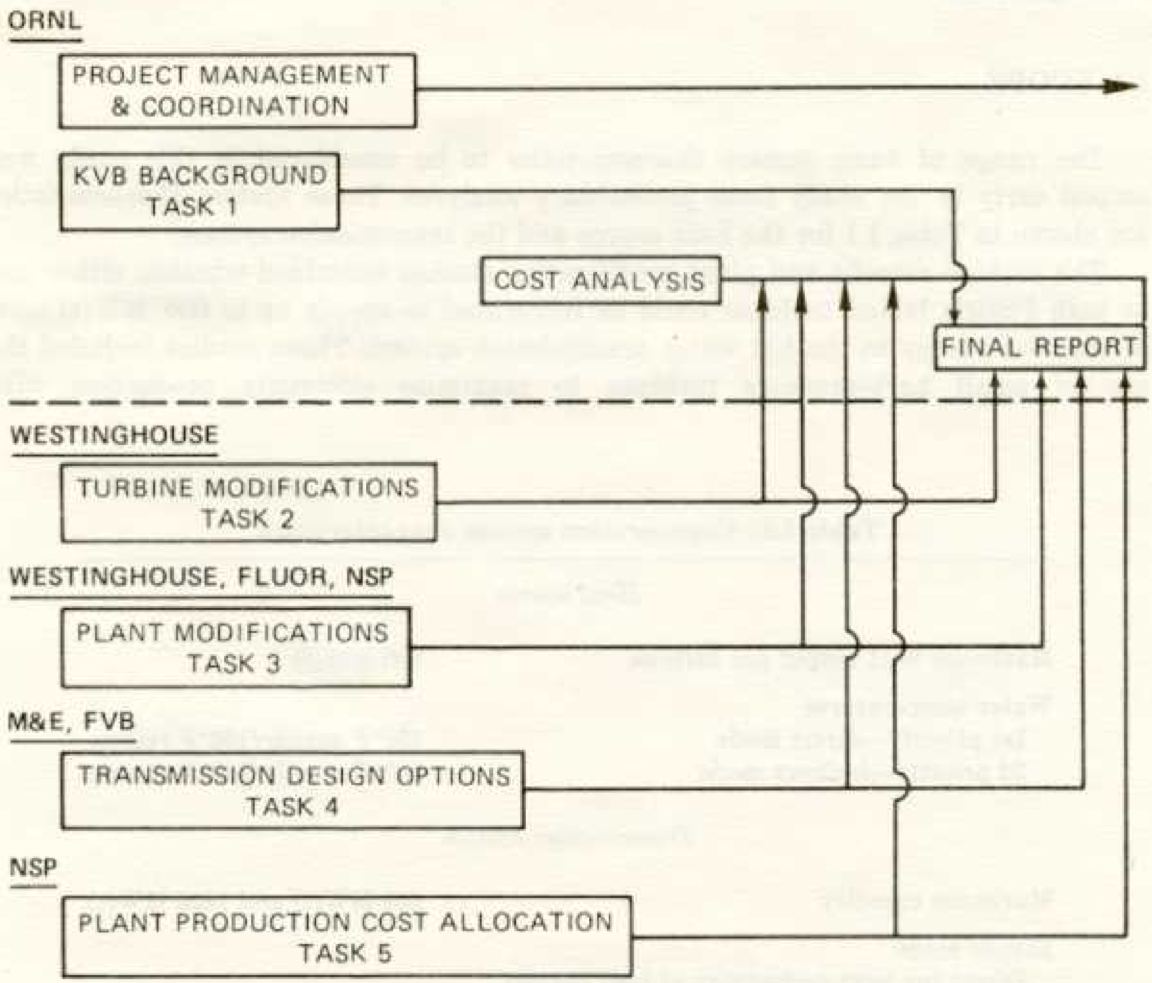


**Fig. 1.1. Nuclear district heating study task organization.**

## 1.2 PURPOSE

The purpose of this study is threefold: (1) to develop the preliminary design for retrofitting an existing LWR power plant for operation in an extraction cogeneration mode, (2) to develop capital cost estimates for retrofitting the LWR unit to cogeneration and for construction of a hot water transmission system, and (3) to estimate the unit cost of thermal energy at the power plant gate and thermal cost of transmission to the load center on the basis of an assumed system annual utilization factor.

Although this study is based on a specific power plant—the NSP Prairie Island Nuclear Plant—the study results are intended to have generic or general application. Although the analysis of Prairie Island as a cogeneration heat source and the Twin Cities as a future load center was based on the potential district heating load for the Twin Cities and depended on the cooperation of NSP in the Twin Cities district heating application studies, this study does not represent a plan on the part of any participant to utilize the Prairie Island plant for a district heating source.

## 1.3 SCOPE

The range of basic system characteristics to be considered in this study was scoped early in the study from preliminary analyses. These system characteristics are shown in Table 1.1 for the heat source and the transmission system.

The turbine retrofit and plant modification studies examined whether either one or both Prairie Island turbines could be retrofitted to supply up to 600 MW(t) each of thermal energy to the hot water transmission system. These studies included the use of small back-pressure turbines to maximize electricity production with

**Table 1.1. Cogeneration system characteristics**

| | |
|---|---|
| *Heat source* | |
| Maximum heat output per turbine | 600 MW(t) |
| Water temperatures | |
| 1st priority—direct mode | 250°F supply/160°F return |
| 2d priority—indirect mode | 350°F supply/180°F return |
| *Transmission system* | |
| Maximum capacity | 600 MW(t) and 1200 MW(t) |
| Supply mode | |
| Direct (no heat exchangers at load center) | |
| Indirect (heat exchangers at load center) | |

cogeneration operation. Two levels of transmission system capacity were established to allow examination of the economic trade-offs between 600- and 1200-MW(t)-sized transmission systems. Two transmission temperature levels were established to provide a general comparison of supplying thermal energy directly from the transmission system to a load center with the alternative of indirect supply through heat exchangers isolating the transmission and distribution systems. The indirect supply mode (at higher transmission temperatures) has the possible advantage of lower costs through reduced transmission piping costs, which normally represent the largest capital cost of such a system. In the analysis of electrical capacity derate from retrofitted turbines, the direct transmission mode at lower temperatures was given first priority over the indirect mode requiring high temperatures, because available resources did not allow a complete and detailed analysis of both power plant extraction cycles.

The plant production costs of thermal energy were estimated by NSP to include only the readily identifiable cost elements for hypothetical, modified plant operation for cogeneration. There are many areas where the economic implications of retrofitting an existing nuclear plant to cogeneration operation would require more detailed information of electric system load characteristics and plant outage scheduling considerations than could be reliably projected ten or more years into the future. These areas are discussed in Sect. 2 and Appendix D but were not included in the analysis of plant production costs for this study.

## 1.4 DESCRIPTION OF THE STUDY

To determine the cost of producing hot water for district heating in the Twin Cities from retrofitted turbines at Prairie Island, a first-level conceptual design was performed for the turbine retrofit, plant modifications, and the hot water transmission system. Then, total project costs were developed for the required equipment and modifications and translated to unit energy costs on the basis of an assumed annual utilization factor of the load center. The approach used and assumptions made for technical and economic analysis are summarized in this section.

### 1.4.1 Turbine and Plant Modifications

In the turbine retrofit study, Westinghouse Electric Corporation employed in-house, computerized performance models to calculate the reduction in electric output, or derate, for increasing steam extraction flows. Turbine blade stresses and steam velocity levels were bases for screening out undesirable extraction flow conditions.

The plant modification study continued the analysis of acceptable steam flow conditions in ancillary equipment to the turbine unit, such as cross-under piping and moisture separator reheaters (MSRs). Space requirements for piping modifications were also considered in the development of a conceptual system design for retrofitting the plant to cogeneration operation. New equipment units—heat exchangers, back-pressure turbines, and MSRs—were specified on the basis of thermodynamic requirements and practical sizes available. For both turbines retrofitted, the conceptual design was based on complete isolation between steam and condensate flow from the individual turbines. Therefore, each retrofitted turbine would be supplied with an identical set of equipment from the steam/condensate perspective.

The conceptual system design was then used for developing a capital and installed cost estimated for all equipment and turbine modifications required to heat the recirculating water in the transmission system. Cost estimates were made by Westinghouse for turbine modification equipment, back-pressure turbines, MSR units, and associated piping and controls. Fluor developed a conceptual facility layout and supplied cost estimates for the heat exchangers, drain tanks, pumps, piping and valves, and building and structure required for the extracted steam-to-hot water production facility. All cost estimates were made in mid-1982 dollars and included allowances for indirect costs (engineering services) and contingencies.

On the basis of the turbine and plant modification studies, Westinghouse suggested several areas which would require more detailed engineering analysis should a nuclear plant turbine such as that installed at the Prairie Island plant ever be retrofitted to cogeneration operation.

### 1.4.2 Transmission System

The transmission system would represent the largest part of the capital investment required for hot water district heating supplied to the Twin Cities from the Prairie Island plant. For the transmission cost to have a realistic basis rather than a "rule-of-thumb" basis, a Transmission Route Option task was developed. This task was conducted by Metcalf & Eddy, Inc., of Boston, Massachusetts, with assistance from FVB of Västerås, Sweden. The overall objectives of this task were (1) to determine, on a first-level basis, the conceptual design and transmission route of the most economically feasible hot water transmission pipeline between Prairie Island and the NSP High Bridge plant in St. Paul; and (2) to develop a cost estimate for construction of the transmission system and the direct operating costs. The design capacities for the system were as specified in Table 1.1.

The first phase of this task was a general survey of geotechnical characteristics along several possible rights-of-way. On the basis of general cost comparison between several pipeline construction techniques—deep tunnels, shallow-buried culverts, or above ground on stanchions—several routes were selected for detailed analysis. Specific pipeline designs were then developed for the temperature and flow requirements of the direct and indirect modes of operation. Finally, construction costs were developed for each major segment of the route, and an overall optimum route and construction costs were determined. Construction costs included pipeline, valves, pumps, tanks for filling and emptying, pressure controls, instrumentation, and terminal heat exchangers (for the indirect mode). All costs were based on mid-1982 conditions, as was used for the turbine and plant modification cost estimates.

### 1.4.3 Price of Thermal Energy from Prairie Island

An important element in determining the overall or total cost of thermal energy from a private-utility-owned cogeneration plant is the "price" of thermal energy from the electric utility. There is a fundamental difference to be appreciated between the "cost" and "price" of thermal energy from the perspective of the utility that would be the seller to another utility distributing the final product. The discussion that follows briefly outlines the difference between these approaches to setting an economic value on cogenerated thermal energy for an existing base-loaded nuclear power plant such as Prairie Island.

In most studies of this type, the cogenerated cost of thermal energy is developed for a new power plant from two factors: first, the cost of energy production, considering fixed or capital-related costs and variable or production-related costs such as fuel; second, a cost allocation that divides the total cost between the two products, electricity and thermal energy. Procedures for calculating total plant costs are well established and documented from standard engineering economic practice. Methods of cost allocation between two cogenerated products can include wide variations in the distribution of the cogeneration advantage or cost reduction depending on the philosophy employed.[3]

However, the cost of thermal energy from a power plant should include additional cost elements to fully represent costs from the electric utility perspective. Such costs are "system-related" costs that derive from situations when the district heating system demand does not coincide with the optimal electric load dispatch determined by overall electrical system capacity and load characteristics. System-related costs are classified as either "replacement" costs (when the cogeneration unit electricity output is decreased by cogeneration and must be replaced by higher-production-cost electricity) or as "excess energy" costs (when the cogeneration unit produces more electricity than would be specified by an optimal economic dispatch of the electric system). For a base-loaded power plant—that is, a plant operated at its rated or licensed capacity at essentially all times that it is available—replacement energy costs will always apply, whereas excess energy costs will seldom apply to the production cost.

In this study, the cost of thermal energy to a purchasing distribution utility was based on a "pricing" methodology developed by NSP. The NSP pricing methodology, discussed in detail in Appendix D, was developed from analyses of future (1992–1997) replacement energy costs for the Prairie Island plant within the NSP electrical generating system and other incremental costs attributable to cogeneration operation of an existing power plant. Replacement energy costs were analyzed for two supply scenarios: "firm" service, which implies interruption of thermal energy production only for electrical system emergencies; and "oil-interruptible" service, for which thermal energy would be interrupted whenever oil-fueled peaking plants would be called upon in the normal economic dispatch of NSP system's generating plants. All cost results were de-escalated from future dollars to mid-1982 dollars for consistency with estimates of construction costs for the transmission system and plant modifications.

### 1.4.4 Economic Analysis Methodology

The primary purpose of this study is to provide a meaningful estimate of the total unit cost ($\$/10^6$ Btu) of thermal energy delivered to a metropolitan load center from an existing, remotely sited nuclear power plant. As was stated earlier, the primary purpose for developing these results is to allow a comparison with the cost of thermal energy from other potential heat sources. The time frame for the earliest possible application of a large transmission system is assumed to be at least ten years into the future, and possibly much longer, because of the time required to develop a large, hot water, metropolitan-scale district heating system.

To provide meaningful results for the cost of thermal energy, two conditions must be met. First, the cost estimates for the overall system must have a sound technical and economic basis. Second, the economic analyses used to calculate the unit cost of thermal energy must be understandable and easily related to current economic conditions. The first condition has been met in this study by using highly qualified subcontractors in their respective areas of expertise. For the second condition, the economic results for a future project are being presented in terms of mid-1982 dollar values and economic conditions rather than inflating and escalating 1982 cost estimates 10 to 15 years into the future. The basic reason for using this approach is that the extreme volatility of inflation rates in the 1980–1983 time period makes long-term projections of inflation and escalation rates highly speculative. Also, economic results presented in highly inflated dollars are difficult to relate to current economic values. However, "allowance-for-funds-during-construction" (AFDC) financing costs are included, using 1982 estimates of long-term inflation and construction cost escalation rates. Section 5 describes the economic assessment methods and assumptions used in this study.

The remainder of this report describes the results of the technical and economic analyses described above. The detailed subcontractor reports, *Plant and Turbine Modifications*, by Westinghouse Electric Corporation; *Transmission Line Route Options*, by Metcalf & Eddy, Inc.; and *Price of Thermal Energy*, by NSP, are given as Appendixes B, C, and D, respectively.

## REFERENCES FOR SECTION 1

1. A. Rubin, M. A. Karnitz, and J. O. Kolb, "The Economics of Long Distance Thermal Energy Transport for District Heating Applications," *Proceedings of the Third Miami International Conference on Alternative Energy Sources*, Bal Harbour, Fla., Dec. 15–17, 1980.

2. Peter Margen et. al., *Overall Feasibility and Economic Viability for a District Heating/New Cogeneration System in Minneapolis-St. Paul*, ORNL/TM-6830/P3, Oak Ridge National Laboratory, Oak Ridge, Tenn., October 1979.

3. G. F. Pavlenco and G. A. Englesson, *Allocation Methods for the Separation of Electrical and Thermal Cogeneration Costs*, ORNL/TM-6830/P12, Oak Ridge National Laboratory, Oak Ridge, Tenn., October 1980.

# 2. LOAD CENTER REQUIREMENTS AND COGENERATION HEAT SOURCE CHARACTERISTICS

In a 1978 study assessing the overall feasibility of hot water district heating for the Twin Cities,[1] the future district heating load for Minneapolis-St. Paul was estimated to be between 2600 and 4000 MW(t). Analyses from this study will be used as a basis for considering the characteristics of the thermal load to be met by nuclear units such as the Prairie Island plant. The relationships between the thermal load and the nuclear plant heat source will be considered further in this section.

## 2.1 ROLE OF NUCLEAR COGENERATION

In general, the dominant cost in using the remote power plants as heat sources for a metropolitan area such as the Twin Cities would be the cost of the hot water transmission system to the urban load center. A corollary of this observation is that base-load cogeneration sources for a district heating system would be selected largely on the basis of the distance from the load, and the type of fuel used would be of very little importance. Nevertheless, nuclear plants are well-suited to cogeneration. They are usually the lowest-fuel-cost plants in any system and are therefore operated with the highest capacity factors that the operators can achieve. A high-capacity factor is favorable for the amortization of the hot water transmission system.

There appear to be two primary technical requirements imposed by the use of a nuclear plant as opposed to any other type of base-load unit:

1. The steam system must be isolated from the hot-water transmission system by a heat exchanger.
2. The hot-water side of the heat exchanger must be operated at a higher pressure than the steam side.

The above requirements result from the need to prevent any radioactivity that might be present in the reactor steam system from reaching the transmission line. (Various monitors and block valves may be used in addition.) Most designs for cogeneration systems impose a heat exchanger between the hot water and the steam, mainly to maintain the high quality required for boiler feedwater. Therefore, the only requirement unique to the nuclear system is that the hot water side pressure always be maintained greater than the steam side.

It appears, then, that nuclear plants have no large advantage or disadvantage relative to other base-load plants, and the selection of plants for retrofit to cogeneration would be based mainly on the economics of the transport of the heat to the load. In general, larger sources closer to the load center would be favored.

## 2.2 POTENTIAL HEAT SOURCES FOR THE TWIN CITIES SYSTEM

A hypothetical supply-and-demand scenario (Scenario "A" from ref. 1) for district heating load growth in Minneapolis–St. Paul, shown in Fig. 2.1, was chosen as the

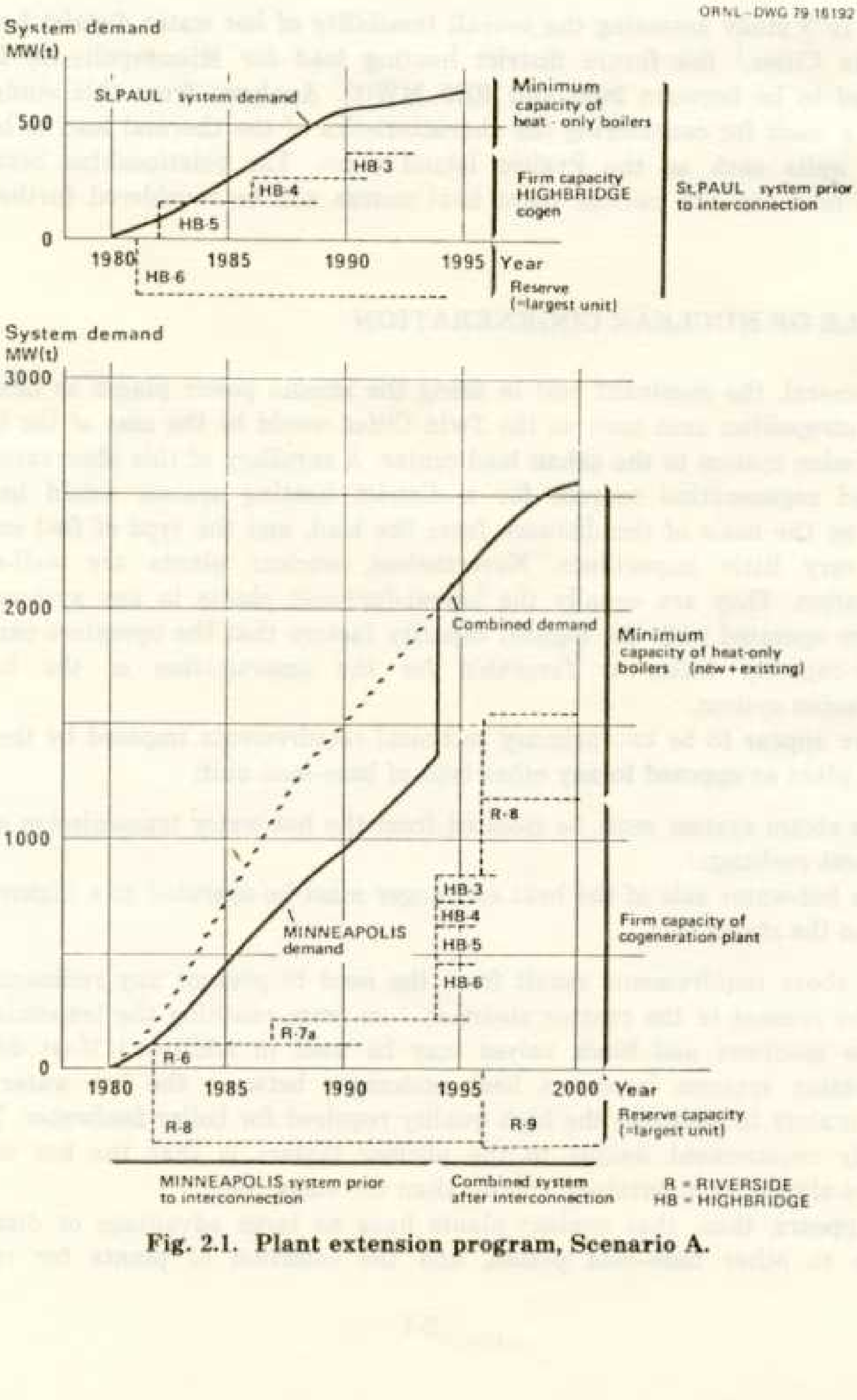


**Fig. 2.1. Plant extension program, Scenario A.**

basis for this study. The total system demand was projected to reach 2600 MW(t) by the year 2000. A thermal demand of this magnitude would be sufficient to justify considering the conversion of a nuclear plant to cogeneration.

It should be realized that the rate of demand growth in Fig. 2.1 was assumed in ref. 1 on the basis of an aggressive development of district heating in the Twin Cities. The institutional and financial support for such rapid development is critical, so that the probability of developing a large district heating market is somewhat speculative.

The Studsvik study suggested that, if district heating demand grew to the 2600-MW(t) size, a new coal-fired cogeneration plant could be built at the King site with a transmission distance to the load center of about 17 miles (10.6 km). Also, an extensive conceptual design and cost study was conducted by United Engineers & Constructors, Inc., for ORNL and NSP for a new coal-fueled cogeneration unit at the High Bridge Plant in St. Paul or a new site at Coon Rapids, 11.6 miles (18.6 km) northeast of Minneapolis.[2] This study considered a 250-MW(e)/350-MW(t)-sized cogeneration turbine at High Bridge and a 400-MW(e)/700-MW(t)-sized turbine at Coon Rapids. If such a new plant were not to be built near the cities, it would then be logical to consider conversion of Prairie Island to cogeneration operation. For economic reasons, the conversion of Prairie Island would not be considered until all closer-in sources were utilized, because of the dominant role of the cost of transmission mains in the economic balance.

## 2.3 THERMAL LOAD CHARACTERISTICS

The assumed annual heat load duration curve for the hypothetical district heating system is shown in Fig. 2.2. For about two months during the year, the full thermal capacity of the cogeneration system is required to meet the load; for the rest of the year, only part of the thermal capacity is needed. Table 2.1 presents the monthly average demand and energy for 600- and 1200-MW(t) capacity systems. The total annual energy for the load duration in Fig. 2.2 is calculated to be $20.57 \times 10^{12}$ Btu ($21.7 \times 10^{6}$ GJ or 251,164 MWd).

During the part-load periods, some of the nuclear cogeneration heat supply could be reduced, and the electrical production would be increased. However, the nuclear reactor and the thermal transmission system would be operated as a base-load unit except for situations discussed below. Cogeneration could supply about 46 or 23% of the winter peak thermal load with 1200- or 600-MW(t) capacity transmission systems. Heat-only boilers or other cogeneration plants would be used to supply the remainder of the thermal load on a peaking or intermediate basis.

Since electricity production would have priority over thermal production for emergency demands (see Appendix B), there would be periods when all the electrical capacity of a nuclear plant is required, and thermal cogeneration must be interrupted. As electricity demand has been summer-peaking in this region, such interruption of cogeneration would not often be necessary in the winter season.

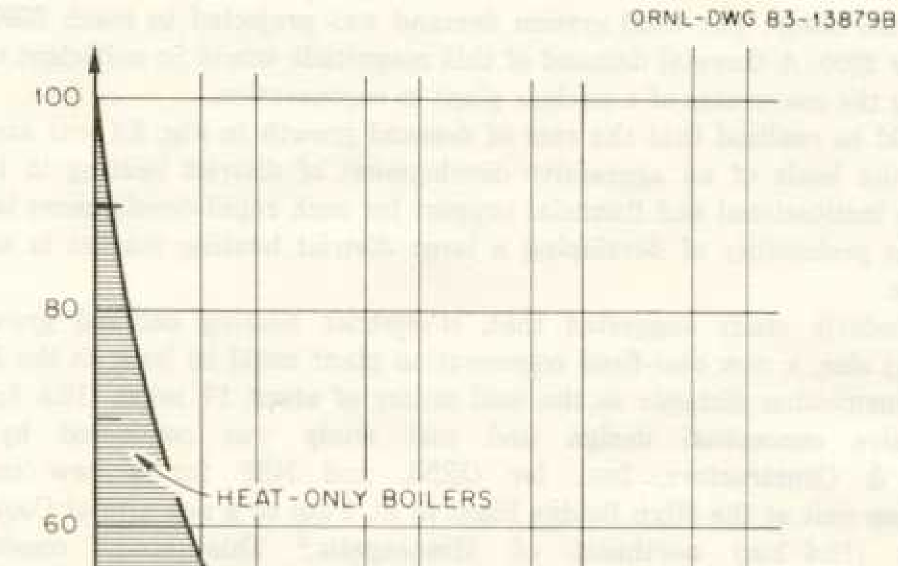


**Fig. 2.2. Heat load duration curve and load split.**

Basically, there are two situations which would reduce the thermal energy delivered from base-load power plants such as Prairie Island. The first situation is an assumed (NSP, Appendix B) 20% reduction in energy delivery to account for scheduled and unscheduled plant outages, both for the nuclear cogeneration units and also for other units causing a capacity short-fall. This availability reduction is shown in Fig. 2.3 as a uniform reduction in available time over the range of heat load. Such a uniform reduction favors plant availability during the winter heating season compared to the summer, high electricity demand season. The resulting load duration curve represents "firm" service to the cogeneration production of thermal energy.

**Table 2.1. Cogeneration energy demand for 600- and 1200-MW(t)-capacity systems**

| Month | 1200 MW(t) Average demand [MW(t)] | 1200 MW(t) Average energy [MWD(t)] | 600 MW(t) Average demand [MW(t)] | 600 MW(t) Average energy [MWD(t)] |
|---|---|---|---|---|
| 1 | 1,200 | 36,500 | 600 | 18,250 |
| 2 | 1,200 | 36,500 | 600 | 18,250 |
| 3 | 1,200 | 36,500 | 600 | 18,250 |
| 4 | 1,070 | 32,528 | 600 | 18,250 |
| 5 | 800 | 24,320 | 600 | 18,250 |
| 6 | 600 | 18,240 | 600 | 18,240 |
| 7 | 400 | 12,160 | 400 | 12,160 |
| 8 | 340 | 10,336 | 340 | 10,336 |
| 9 | 230 | 6,992 | 230 | 6,992 |
| 10 | 170 | 5,162 | 170 | 5,162 |
| 11 | 130 | 3,952 | 130 | 3,952 |
| 12 | 70 | 2,128 | 70 | 2,128 |
| Total energy | | 225,318 | | 150,130 |

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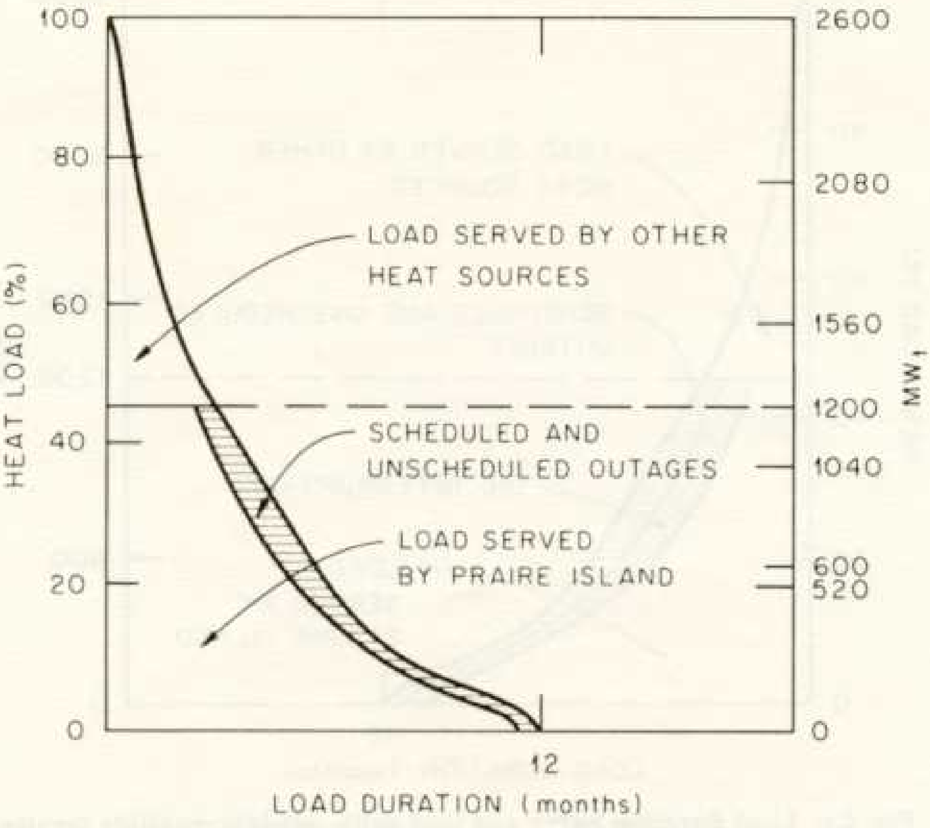


**Fig. 2.3. Load duration curve and load split—firm service.**

Firm service also represents the highest priced thermal energy from the cogeneration plant because of the higher "replacement" energy costs associated with the sacrificing of low-cost electricity production to higher-cost production plants. Therefore, NSP also provided estimates of the thermal energy cost and annual production level for "oil-interruptible" service from the Prairie Island units (see Appendix B). For this situation, the cogeneration thermal production was estimated to be 85% of the "firm" supply, as shown in Fig. 2.4. This estimate was based on NSP's analysis of electric system demand and capacity characteristics projected out to the 1992–1997 time frame and an estimated 15% coincidence factor for thermal and electric system peak demands. From both firm and oil-interruptible thermal energy costs from the power plant, this study can assess the trade-off between the lower cost of thermal energy for oil-interruptible service with the higher unit transmission cost from a lower utilization of the transmission system capacity.

## 2.4 THERMAL TRANSMISSION LAG EFFECTS

An additional consideration for supplying the base district heating load from a normally base-loaded power plant is the effect of thermal transmission lag on the

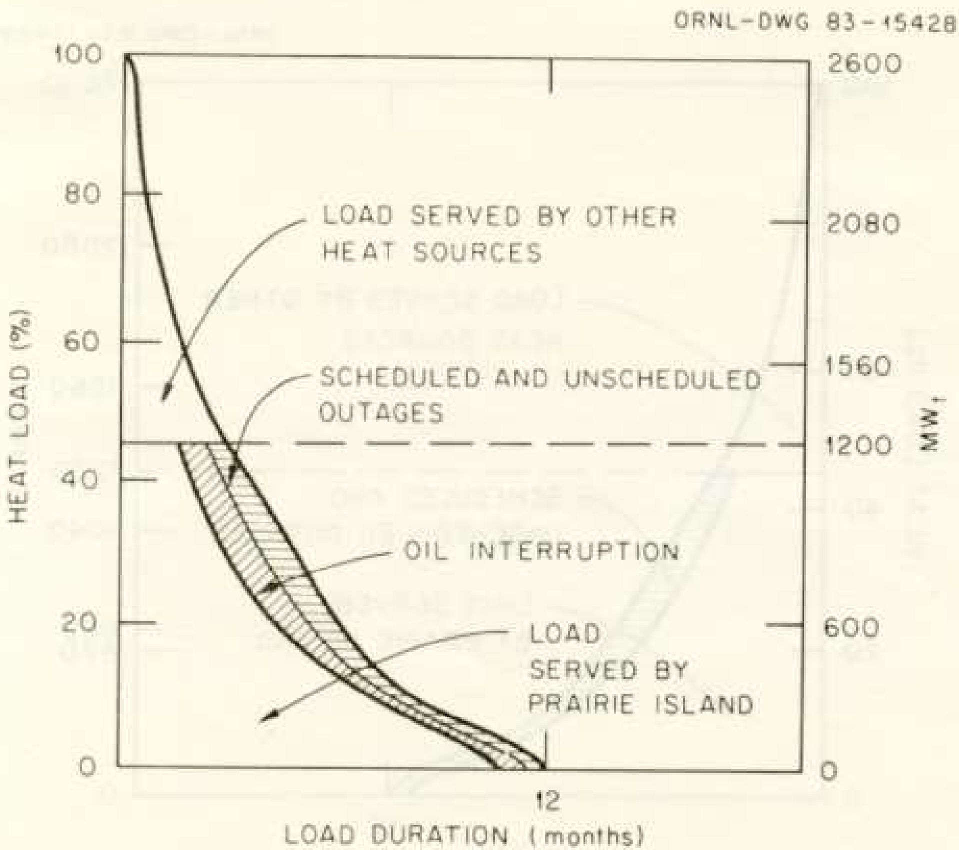


**Fig. 2.4. Load duration curve and load split—oil-interruptible service.**

coincidence of electric and district heating system loads. The transport time in the transmission pipeline between the Prairie Island plant and the load center at High Bridge in St. Paul would be on the order of 8 h at full load. The effects of this thermal lag are largely beneficial, especially for the oil-interruptible supply mode, because of the daily cycle in electric and thermal loads.

Hourly load curves for electrical load of the NSP system and for thermal load for the St. Paul system are shown for a typical January day in Fig. 2.5. The electrical demand is high from 6 a.m. until 10 p.m., with a peak at 5 p.m. The thermal load is high from 6 a.m. to 4 p.m., with a peak at 7 a.m. The basic advantage of an 8-h thermal lag is that the peak thermal energy between 6 a.m. and 6 p.m. can be produced at the cogeneration plant between 10 p.m. and 10 a.m. If plant outages in the electric generating system required high-cost, oil-fueled plants to operate during the electric peak period of 8 a.m. to 8 p.m., then

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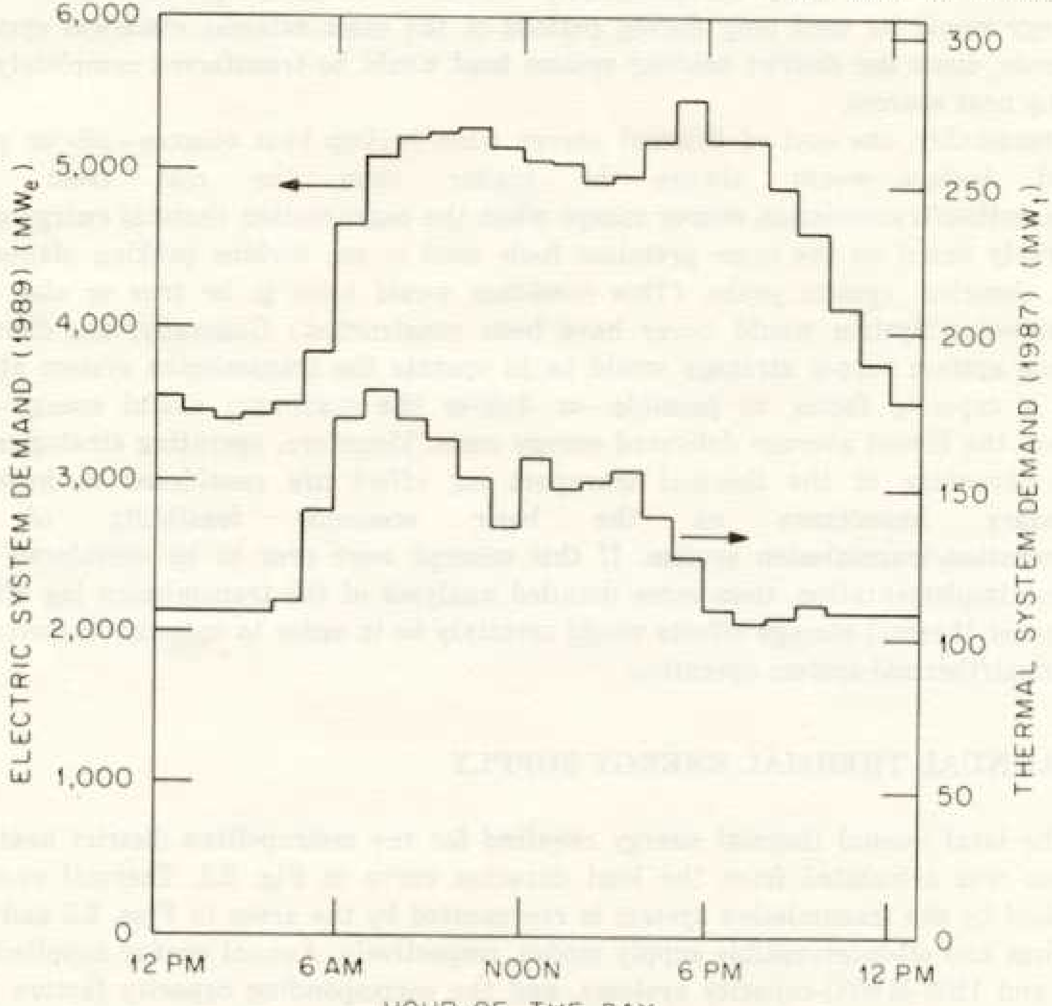


**Fig. 2.5. Anticipated electrical and thermal loads for a typical January weekday.**

electricity production at the cogeneration plant could be partially or fully restored. The reduced thermal output could be accommodated into the district heating system production strategy in several ways.

One way to accommodate reduced cogenerated thermal production would be to reduce the supply-water temperature while maintaining the transmission flow rate during the day. For a large part of the year, the relatively low system load during the night could be supplied from reduced supply temperatures (see ref. 2). This approach could tailor the transmission system supply temperature to the diurnal variation in the heating load without resorting to input from backup heat sources. The net effect of this approach would increase the cogenerated thermal energy and reduce the backup supplied energy for the oil-interruptible mode shown in Fig. 2.3.

Another way to increase nuclear electrical production to the greatest extent, if required by the electrical system load, would be to reduce the transmission system flow to a minimum with a corresponding reduction in extraction steam flow. This would decrease the electrical system load by the pumping electrical load, plus restore the nuclear plant to essentially maximum electrical production. Such a strategy would be used only during periods of the most extreme electrical system demands, since the district heating system load would be transferred completely to backup heat sources.

Presumably, the cost of thermal energy from backup heat sources—oil- or gas-fueled boilers—would always be higher than the cost from the cogeneration/transmission source except when the cogeneration thermal energy cost is largely based on the same premium fuels used in gas turbine peaking plants to meet electrical system peaks. (This condition would have to be true or else the transmission system would never have been constructed.) Generally, the district heating system supply strategy would be to operate the transmission system at as high a capacity factor as possible—or deliver the maximum useful energy—to achieve the lowest average delivered energy costs. Therefore, operating strategies to take advantage of the thermal transport lag effect are considered to have a secondary importance on the basic economic feasibility of a cogeneration/transmission system. If this concept were ever to be considered for serious implementation, then more detailed analyses of the transmission lag effect and other thermal storage effects would certainly be in order to optimize the overall electrical/thermal system operation.

## 2.5 ANNUAL THERMAL ENERGY SUPPLY

The total annual thermal energy required for the metropolitan district heating system was calculated from the load duration curve in Fig. 2.2. Thermal energy supplied by the transmission system is represented by the areas in Figs. 2.3 and 2.4 for firm and oil-interruptible supply modes, respectively. Annual energy supplied by 600- and 1200-MW(t)-capacity systems, and the corresponding capacity factors are presented in Table 2.2 for the two supply modes. Values for the annual energy supplied are the basis for the unit energy costs ($\$/10^6$ Btu or $10^6$ GJ) calculated in Sect. 5.

**Table 2.2. Annual transmission system capacity utilization[a]**

| | Firm service | | Oil-interruptible service | |
|---|---|---|---|---|
| | 600 MW(t) | 1200 MW(t) | 600 MW(t) | 1200 MW(t) |
| Annual energy supplied | | | | |
| $10^{12}$ Btu | 9.84 | 14.76 | 8.36 | 12.55 |
| $10^{6}$ GJ | 10.38 | 15.57 | 8.82 | 13.24 |
| Percentage of total energy required | 47.8 | 71.8 | 40.6 | 61.0 |
| Transmission line capacity factor, % | 54.8 | 41.2 | 46.6 | 35.0 |

[a]Assumed total annual energy requirement is $20.57 \times 10^{12}$ Btu ($21.7 \times 10^{6}$ GJ).

For long thermal transmission pipelines as being considered here, heat loss through the pipe wall and insulation would occur. However, the amount of heat loss from reasonably well-insulated pipelines will generally be balanced by the pumping energy input to the system.[2] Therefore, the energy delivered to the load center is assumed to be the same as energy input from heat exchangers at the power plant heat source.

## REFERENCES FOR SECTION 2


1. Peter Margen et al., *Overall Feasibility and Economic Viability for a District Heating/New Cogeneration System in Minneapolis-St. Paul*, ORNL/TM-6830/P3, Oak Ridge National Laboratory, Oak Ridge, Tenn., October 1979.

2. G. A. Englesson et al., *Economic Comparison of New Coal-Fueled, Cogeneration Power Plants for District Heating and Electric-Only and Heat-Only Power Plants*, ORNL/TM-6830/P8, Oak Ridge National Laboratory, Oak Ridge, Tenn., May 1982.

# 3. TURBINE AND PLANT MODIFICATIONS

## 3.1 INTRODUCTION

One of the key areas in this study has been the analysis and evaluation of an existing LWR nuclear turbine-generator retrofitted to extraction cogeneration operation to supply a hot water transmission system. Since the Prairie Island plant has two 560-MW(e)-rated Westinghouse turbine generator units, the Westinghouse Electric Corporation was selected to perform two tasks—Turbine Modification (Task 2) and Plant Modification (Task 3). Westinghouse was assisted by Fluor Power Service, Inc., the A-E firm for Prairie Island, in the conceptual design and cost estimate for the hot water production facility.

This section presents a summary of the Westinghouse Task 3 report, which appears in Appendix B.

### 3.1.1 Performance Requirements

The performance requirements of the Westinghouse turbine and plant modification studies were to analyze a retrofit design to provide a minimum of 300 MW(t) and a maximum of 600 MW(t) of thermal energy per turbine unit. Two temperature levels were established for the water side of steam-to-water district heating heat exchangers (DHHX) at maximum heat transfer: (1) a low-temperature transmission supply/return of 250/160°F for direct transfer at the load center to the distribution network and (2) a high-temperature transmission supply/return of 350/180°F for indirect transfer through water-to-water heat exchangers at the load center. The direct transfer mode at 250/160°F, to be supplied by back-pressure (BP) turbines (Cycle I) or from extracted steam directly (Cycle II-B), was to have first priority over the indirect transfer mode at 350/180°F from directly extracted steam (Cycle II-A). These performance requirements interface with the design conditions of the hot water transmission system described in Sect. 4.

### 3.1.2 Objectives

The primary objectives of this study were as follows:

1. Analyze the turbine-generator cycle from the standpoint of equipment pressure and flow limitations to establish the minimum electrical derate—or maximum thermodynamic efficiency—up to the specified thermal capacity of 600 MW(t).

2. Develop a conceptual design for the turbine extraction modification and the thermal production facility based on an analysis of interfacing and installation problems of extraction piping, instrumentation, and DHHX location.
3. Develop a capital cost estimate in mid-1982 dollars for the turbine modifications, thermal production facility, and all ancillary equipment required—electric switch gear, etc.

In addition to these primary objectives, Westinghouse was to provide a first-level analysis of plant operational problems and safety and licensing implications from a retrofit to cogeneration operation of a nuclear steam-supplied, turbine-generator unit.

### 3.1.3 Existing Turbine-Generator Unit

In its existing configuration, each turbine-generator in the Prairie Island plant has a double-flow, high-pressure (HP) element and two double-flow, low-pressure (LP) elements with 40-in. last-row blades. The HP element has four control valves supplying a partial-arc admission first stage which is followed by five stages of reaction blading. Steam is extracted for feed heating from a midpoint in the HP element and from the crossunder pipe at the HP element exhaust. The HP element exhausts to a combined moisture separator reheater (MSR). The moisture separator section takes the HP exhaust steam with about 10% wetness and restores it to an essentially dry and saturated condition. The moisture is drained to a flash tank in the feed cycle. The separator outlet flow then enters the single-stage, steam-to-steam reheater using throttle steam as the reheating source. The reheater outlet steam enters the 11-stage LP element at about 480°F (249°C), which includes 115°F (64°C) superheat, and expands to condenser pressure. There are three stages of feed water heating supplied from the LP elements (see Fig. 3.1). The unit heat rate at 560 MW(e) is 10,062 Btu/kWh.

## 3.2 TURBINE-GENERATOR CYCLE ARRANGEMENTS

There are a number of possible cycle arrangements and modes of turbine operation, each having different effects on plant performance and the allowable quantity of extracted steam available for district heating. The discussion in this section concerns the primary cycles I and II-B to provide 250/160°F hot water from the DHHX.

### 3.2.1 Multiple- vs Single-Point Extractions

The most efficient cycle arrangements are those that incorporate multiple stages of heat exchange and extract steam at the lowest possible pressures. However, the choice of available locations is constrained by the existing turbine and its cycle. The

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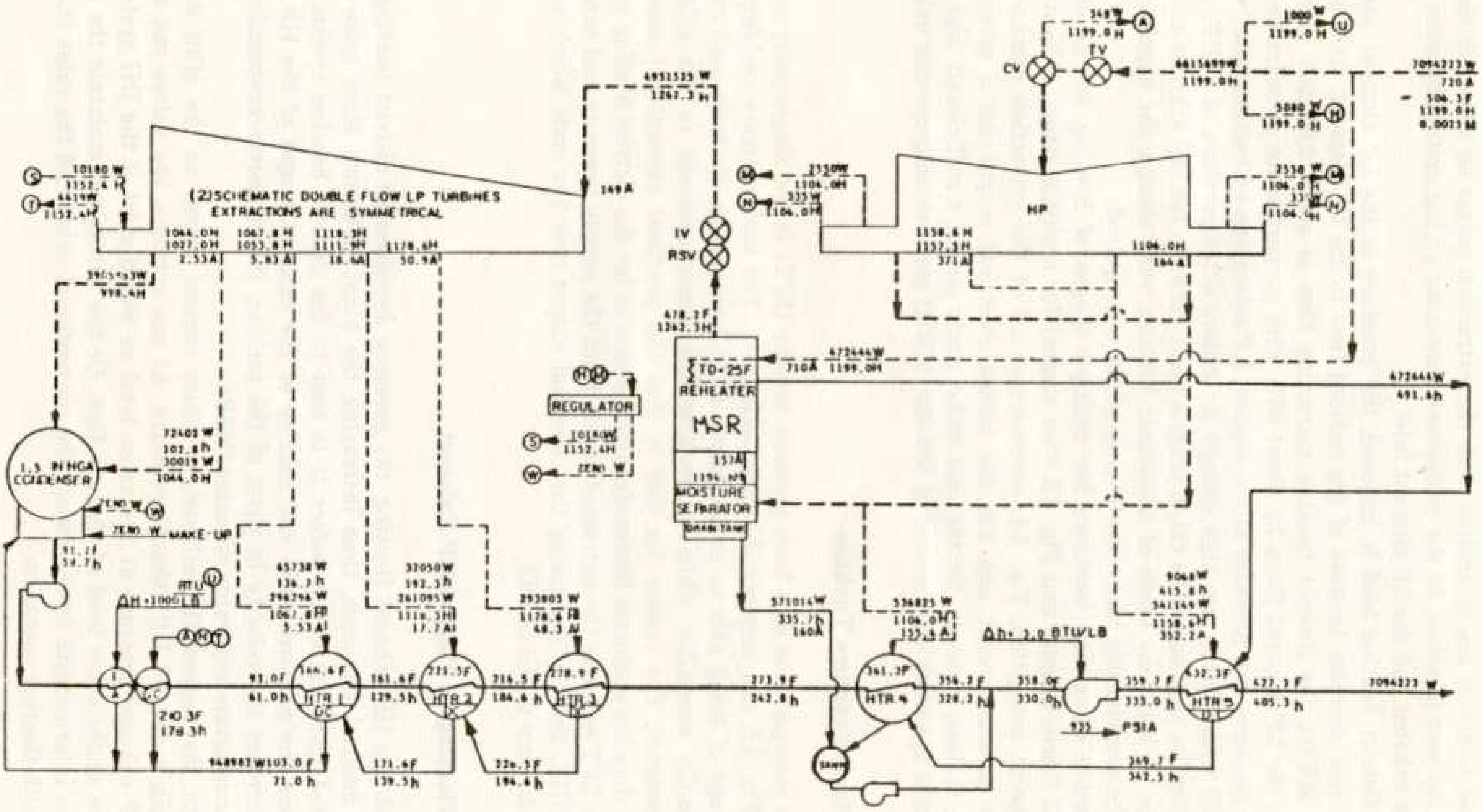


Fig. 3.1. Reference power plant. Heat balance: 559,592-kW turbine-generator unit.

available locations are coincident with the extraction points on the cycle illustrated in Fig. 3.1 and locations in the crossover/crossunder piping system between the HP element exhaust and the LP element inlet.

As district heating load is imposed, the pressure in the LP element and at the HP exhaust decrease because of the reduced flow to the LP element. At a heat load of 600 MW(t), the district heating extraction flow is about 2,000,000 lb/h, which reduces the LP element flows by about 40% with proportionate pressure reductions. In this instance, the pressure at the highest LP interstage extraction point would be about 30 psia. This is not high enough to achieve a temperature of 250°F. All such cycles require that steam be taken from a point between the HP exhaust and the LP inlet for at least one stage of additional heating, even though the supply pressure would be considerably above the 39-psia minimum required.

Feasible extraction locations for multiple stages of heating are as shown in Fig. 3.2 (three stages) and Fig. 3.3 (two stages) for multiple stages of heating. The single-point extraction, Fig. 3.4, concentrates all of the extraction demand at the highest pressure point and has the lowest electrical output for a given energy supply. It does, however, for the case under study permit a sufficiently high level of extraction to satisfy the demand of 300–600 MW(t) per turbine-generator unit.

### 3.2.2 Back-Pressure Turbine

The incorporation of a back-pressure turbine (BPT) in the single-point extraction cycle, Fig. 3.5, can improve the plant efficiency. The back-pressure turbine has the advantage of being able to accept steam from locations in the main unit cycle that are readily accessible while minimizing the losses inherent in the other cycle arrangements. The reason for this is that the practical extraction pressures are higher than the optimum thermodynamic pressures for the district heating system.

The BPT expands the extracted steam from the supply pressure and exhausts to the DHHX, thereby increasing the electrical output over the cycle which sends the steam directly to the DHHX.

### 3.2.3 Floating vs Pegged HP Exhaust

With the HP exhaust floating, the pressure decreases as district heating system steam demand increases, thus increasing the plant efficiency since more work is extracted from the steam before it is sent to the district heating system. It also increases the pressure ratio and loading of the exhaust stages of the HP element and increases the velocity in parts of the turbine, the crossover/crossunder piping, and the moisture separator reheater (MSR).

With the pegged HP exhaust, pressure control valves at the MSR discharge maintain a desired HP exhaust pressure. At one extreme, the valves can maintain the HP exhaust pressure at the same level as would exist if the DH system were inactive at the same level of throttle flow. Or the valves can maintain the pressure at some intermediate level between the conventional value and the value that would exist with floating operation.

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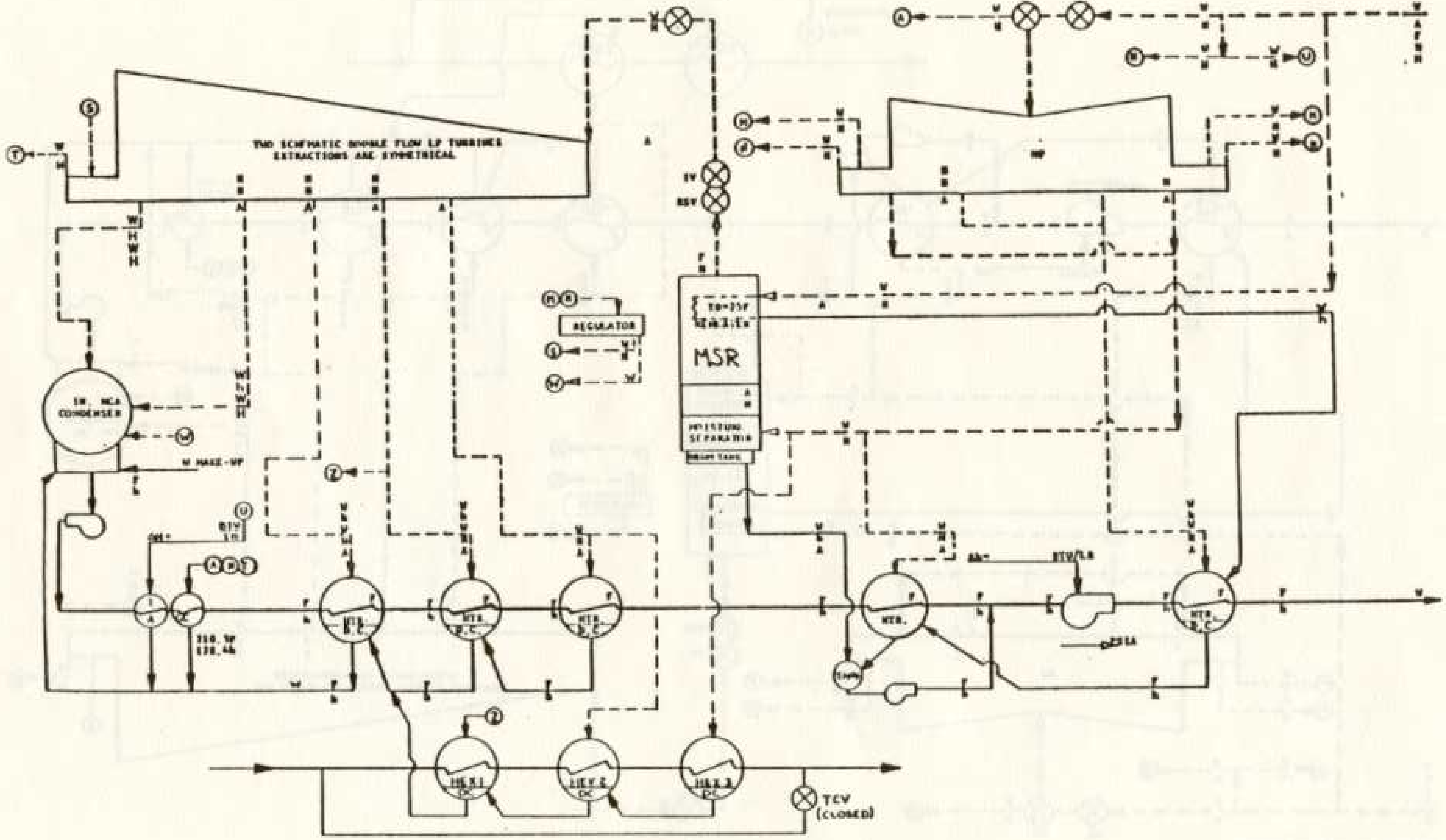


Fig. 3.2. Cycle schematic, three-point extraction.

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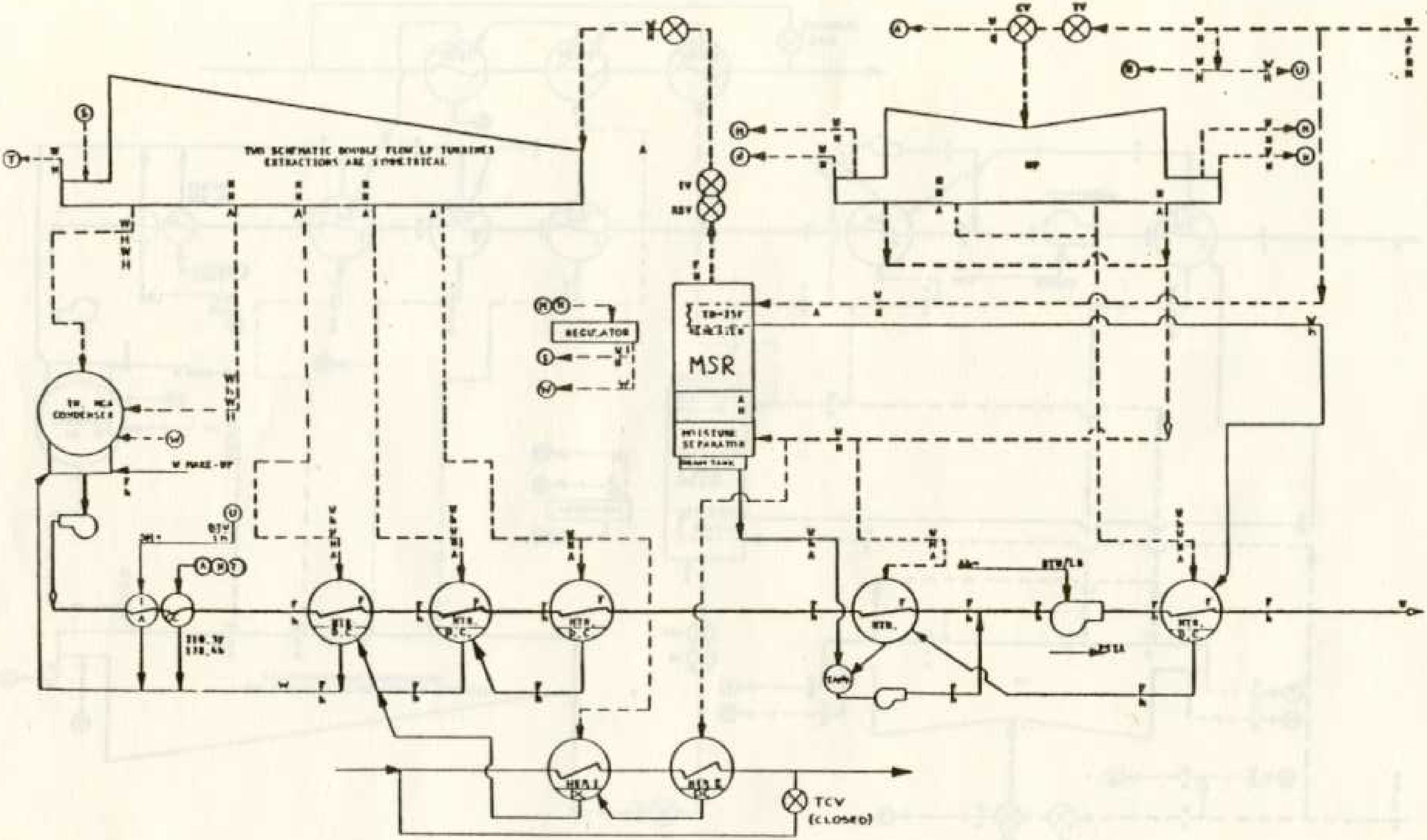


Fig. 3.3. Cycle schematic, two-point extraction.

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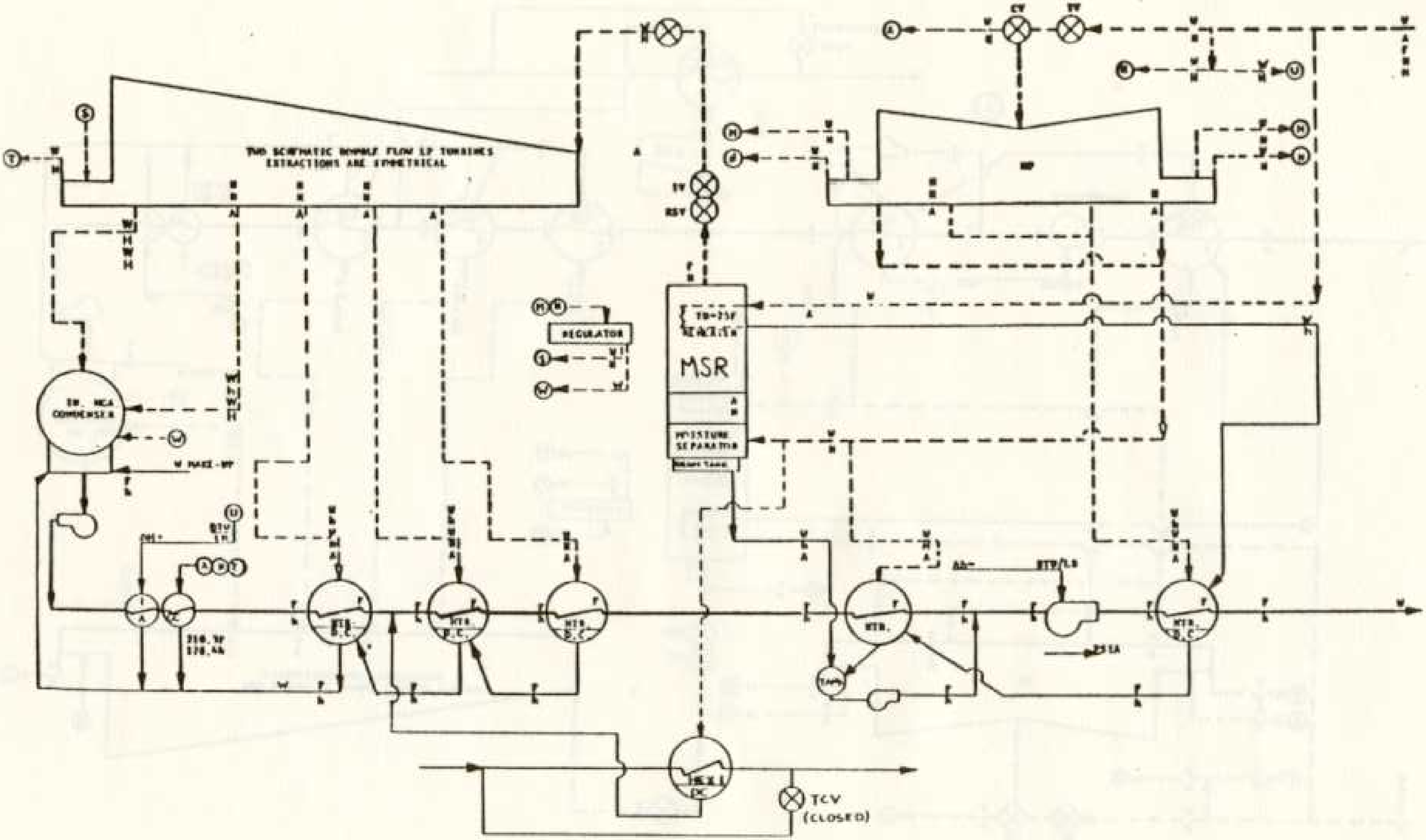


Fig. 3.4. Cycle schematic, single-point extraction.

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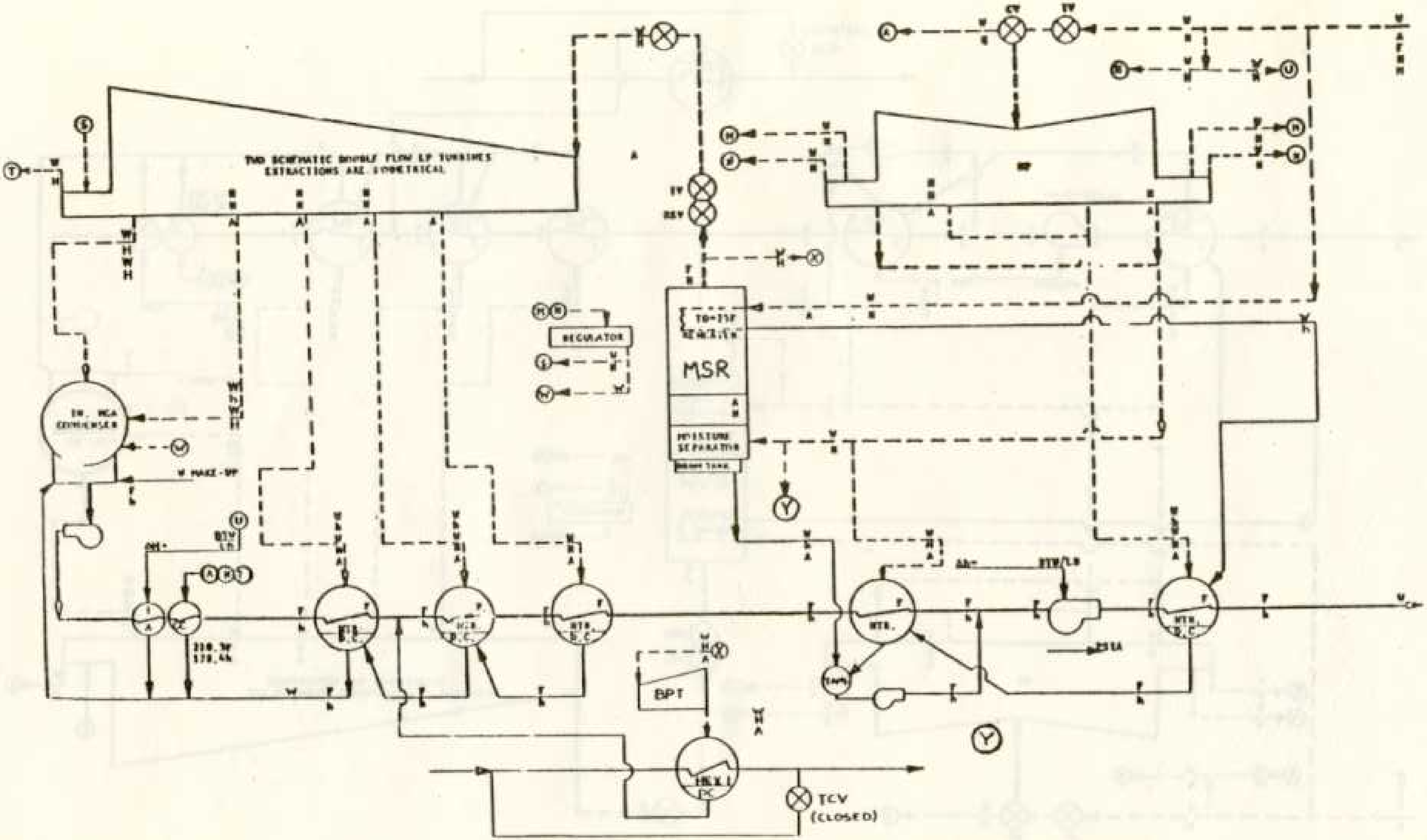


**Fig. 3.5. Cycle schematic, single-point extraction with back-pressure turbine.**

The minimum reasonable extraction pressure that would deliver hot water at the desired temperature is about 39 psia. With the HP exhaust pressure pegged at the normal full-load value of 165 psia, the crossover valves would throttle from 152 psia to 86 psia at a heat supply of about 600 MW(t). Even without valves in the crossover pipes (floating exhaust), the HP exhaust pressure at 600 MW(t) would be about 100 psia, considerably above the 39-psia minimum value.

The pegged cycle incurs throttling losses in the crossover valves as well as losses in the heat exchange process in the DHHX, because the steam supply pressure is higher than the ideal value.

## 3.3 TURBINE AND CYCLE CONSTRAINTS

Because the study is concerned with existing, standard-design turbine-generators which are normally applied for power generation only, there are a number of inherent constraints that would not be present in a custom-designed turbine and plant.

### 3.3.1 Blading

The extraction of additional flow for district heating beyond the normal cycle requirements changes the pressure distribution in the cycle. The turbine stage immediately preceding the extraction point is subjected to the greatest change in pressure ratio and blade loading.

Because of limitations on feasible extraction locations, all of the district heating flow was extracted between the HP exhaust and the LP inlet, which resulted in a substantial increase in loading of the last stage of the HP element. A series of blading checks was made with floating HP exhaust pressure. As a result, it appears that floating HP exhaust pressure operation is permissible at rated throttle flow for extraction demands up to 525 MW(t).

### 3.3.2 Velocity Limits

The Task 2 study and other similar studies reveal that extraction flow limitations, because of excessive velocities, are a major constraint.

#### 3.3.2.1 Extraction slots and piping

Velocity limits generally relate to the extraction slot, the extraction chamber (which is the annular collection area following the slot), and the extraction pipe snout. Excessive velocities can result in structural vibration, noise, and blade excitation.

It is easier to increase the steam flow at some locations in the cycle than at others. The turbine shells are complex, precision structures that interface with rotating parts. In contrast, the steam pipes connecting various turbine elements in the lower pressure parts of the cycle are simple structures that interface with stationary parts.

The pipes connecting the turbine elements and the MSRs are examples. Fabrication of a modified section of these pipes can be completed between normal plant outages, and the replacement section can be installed during an outage at reasonable cost. On the other hand, modifications to the turbine shells and blade path require additional or extended plant outages. The modification costs are comparatively high in relation to that of the pipes. Replacement power costs related to the outage must also be taken into account.

The extraction openings in the turbine shells used for feedwater heating are sized to limit steam velocities to values compatible with efficiency and reliability considerations on normal applications. The incorporation of district heating with appreciable steam flow extractions increases the mass flow considerably. The increased extraction flows result in lower extraction pressures, which further increases the extraction velocity.

The extraction of 300 MW(t) of heat energy from a single location in the turbine cycle would result in an extracted mass flow of about 1,000,000 lb/h. Note that this is at least four times the mass flow removed at any LP element extraction point (Fig. 3.1). With a heat demand of 600 MW(t), the extraction flow would be about 2,000,000 lb/h.

#### 3.3.2.2 HP turbine exhaust chamber and crossunder piping

Operation with floating HP exhaust pressure increases the velocity in the HP exhaust chamber, particularly at the exhaust snouts as well as in the piping between the exhaust snouts and the extraction line to the district heating system. The extraction pipe to heater 4 (Fig. 3.1) is not connected to the shell at the HP exhaust but to the crossunder piping below the HP element.

With floating HP exhaust pressure, the cycle which extracts steam from the HP exhaust and sends it directly to the district heat exchangers (Fig. 3.4) has acceptable velocities in the HP exhaust snouts and the crossunder piping at energy demands up to about 360 MW(t). To control the velocity at energy levels above 360 MW(t), valves in the crossover pipes between the MSR outlet and the LP inlet would partially close and peg the HP exhaust pressure at 144 psia with rated throttle flow.

Serious erosion has occurred on the piping and turning vanes of the crossunder piping system of some units. Since increasing velocities result in increased erosion, it would be prudent to limit the velocity increase and/or limit the length of piping exposed to higher velocities.

#### 3.3.2.3 Moisture separator reheater (MSR)

In the case of the BPT cycle of Fig. 3.5, there are additional constraints beyond those identified in the preceding discussion when the system operates with floating

HP exhaust pressure at high levels of extracted energy. The increase in MSR steam velocities would affect the moisture-removal effectiveness of the moisture-separator section. In the case of mesh-type separators, the allowable variation in velocity is very small for effective moisture removal. In addition, high velocity will cause deterioration of the mesh. With chevron separators, the latitude in allowable velocity is greater. However, the velocity would probably exceed the threshold where moisture carryover occurs. Finally, higher velocities with the attendant increase in pressure losses can result in vibration and structural collapse of the internal baffling of the MSR.

Crossover pressure control valves were also used on the BPT cycle to limit the velocities. Valve closure would begin at an extracted energy level of 330 MW(t) to maintain a HP exhaust pressure level of 124 psia with rated throttle flow. This additional pressure control strategy is called floating-pegged operation since it combines floating operation up to 330 MW(t) with pegged operation above that extraction level.

In the final configuration, the extraction location for the BPT cycle was changed from the MSR outlet to the HP exhaust (same as the cycle of Fig. 3.4). The extracted steam for the BPTs was sent to a separate MSR, adjacent to the BPTs, to reheat the steam. This reduced the length of crossunder piping exposed to higher velocities and kept the MSR velocities at normal levels.

The BPT MSRs are needed because of the high moisture content in the steam from the HP exhaust. The moisture would erode the BPT control valves and blading as well as reduce the BPT efficiency.

The cycle performance for the BPT cycle with a separate MSR is similar to the cycles where the BPTs receive steam from the exhaust of the main unit MSRs. The differences in performance occur in the operating range where the crossover valves throttle to limit the steam velocities. Cycle performance was recalculated for the BPT cycle and the cycle on Fig. 3.4 for the load points where the crossover valves are partially closed.

#### 3.3.2.4 Feed heaters

The extraction pipe velocities for the feed heaters supplied from the LP element increase when district heating is incorporated in a cycle, in contrast to a conventional plant in which extraction flow decreases as throttle flow decreases. In the latter case, the extraction volumetric flow and, consequently, the pipe velocities are practically constant for all loads with the exception of the lowest pressure extraction. However, when district heating is included in the cycle, the pressure (at all locations downstream of the highest pressure extraction point with district heating) decreases as the district heating energy demand increases. In addition, the condensate flow in the feed train is at the high value corresponding to normal cycle operation without district heating. The combination of lower flow in the turbine blading (consequently, lowered extraction pressures) and high flow in the condensate system results in high extraction line volumetric flows and velocities.

In order to limit the pipe velocities at these locations, flow was bypassed around these two heaters, as shown in Fig. 3.6. The flow is regulated by the bypass control valve (BPCV).

No bypass was required for an energy supply up to about 210 MW(t) at rated throttle flow. The bypass quantity at all higher energy levels was varied between 0 and 30% of the approaching condensate flow. The required level of bypass is shown in Fig. 3.7.

### 3.3.3 Final Feedwater Temperature

When the main unit is operated with a floating HP exhaust pressure, the final feed water temperature decreases as more and more steam is extracted for the district heating system. If the change is large enough, it can affect reactor operation.

With a floating HP exhaust pressure, the temperature decrease is greatest and was about 12°F at 600 MW(t). With a pegged HP exhaust pressure, the final feed temperature increased by about 1°F at 600 MW(t). The corresponding decreases for the cycle of Fig. 3.6 and the BPT cycle are 5 and 8°F, respectively, at 600 MW(t) when the system operates in the combined floating-pegged mode. These latter values appear to be in the range where reactor operation is not affected.

## 3.4 PERFORMANCE RESULTS FOR SELECTED CYCLES

Three similar cycle arrangements are described, one with back-pressure turbines (Cycle I) and two without back-pressure turbines (Cycles II-A and II-B). These cycles would be applicable to a number of other plants with both larger or smaller ratings than the reference plant. The cycles for 250°F hot water production (I and II-B) are described first since the analysis of these cycles was the primary emphasis. Cycle II-A for 350°F hot water production without back-pressure turbines was treated as a perturbation of Cycle II-B.

### 3.4.1 Cycles for 250°F Hot Water Production

The detailed performance investigations of Task 2 indicated that a single-extraction-point cycle was the only feasible alternative. Cycles with a greater number of extraction points were extremely limited in the allowable extraction energy level, less than 75 MW(t), because of velocity limitations at the extraction points in the LP element.

The loss in electrical output, or unit derate, calculated for single-point extraction from the HP element exhaust is shown in Fig. 3.8 for floating and pegged HP element exhaust pressure. The cycles that allow the HP element exhaust to float have performance superior to those in which the HP exhaust was pegged. However, at extraction energy levels between 300 and 400 MW(t), the velocities in the HP

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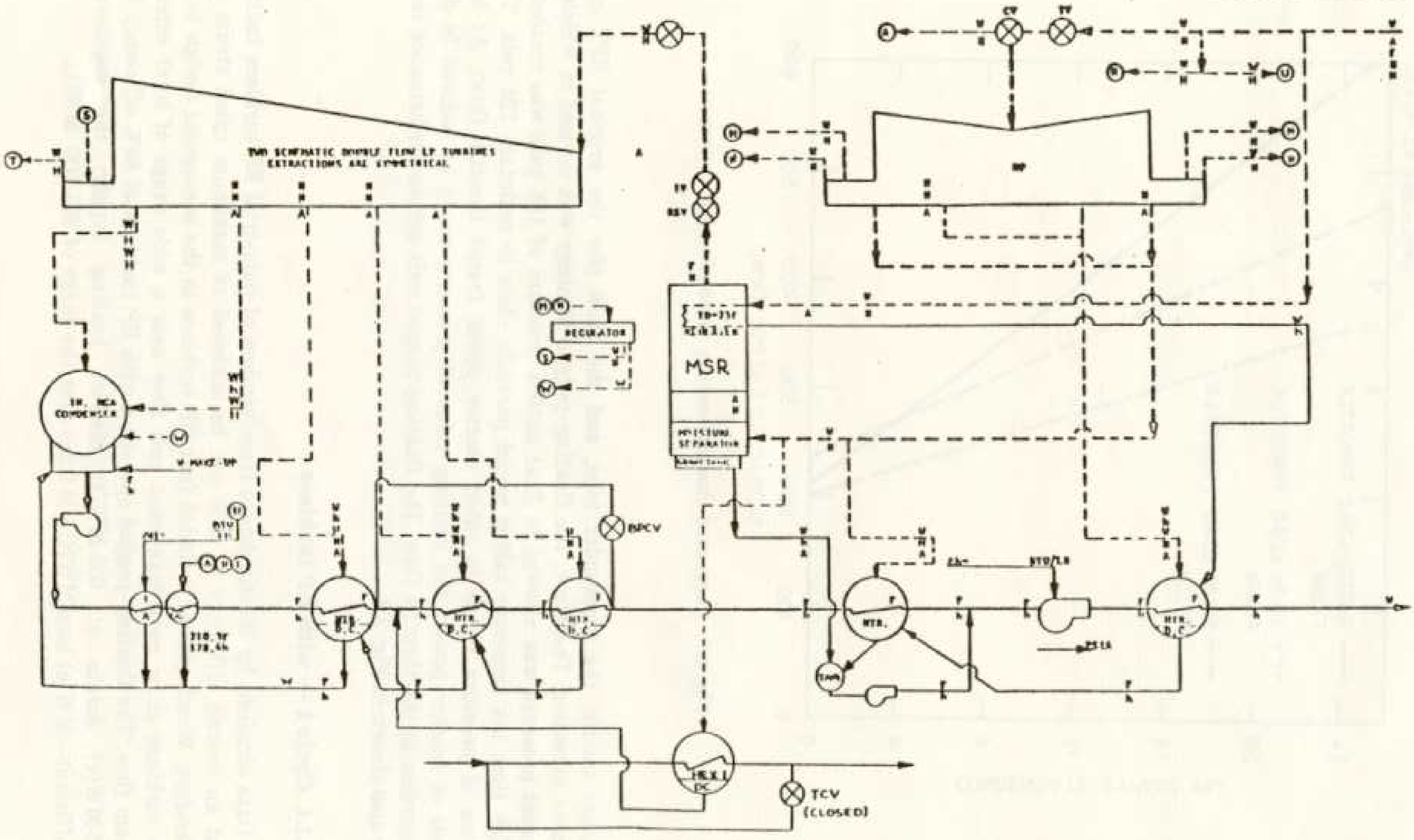


**Fig. 3.6. Cycle schematic, single-point extraction with heater bypass.**

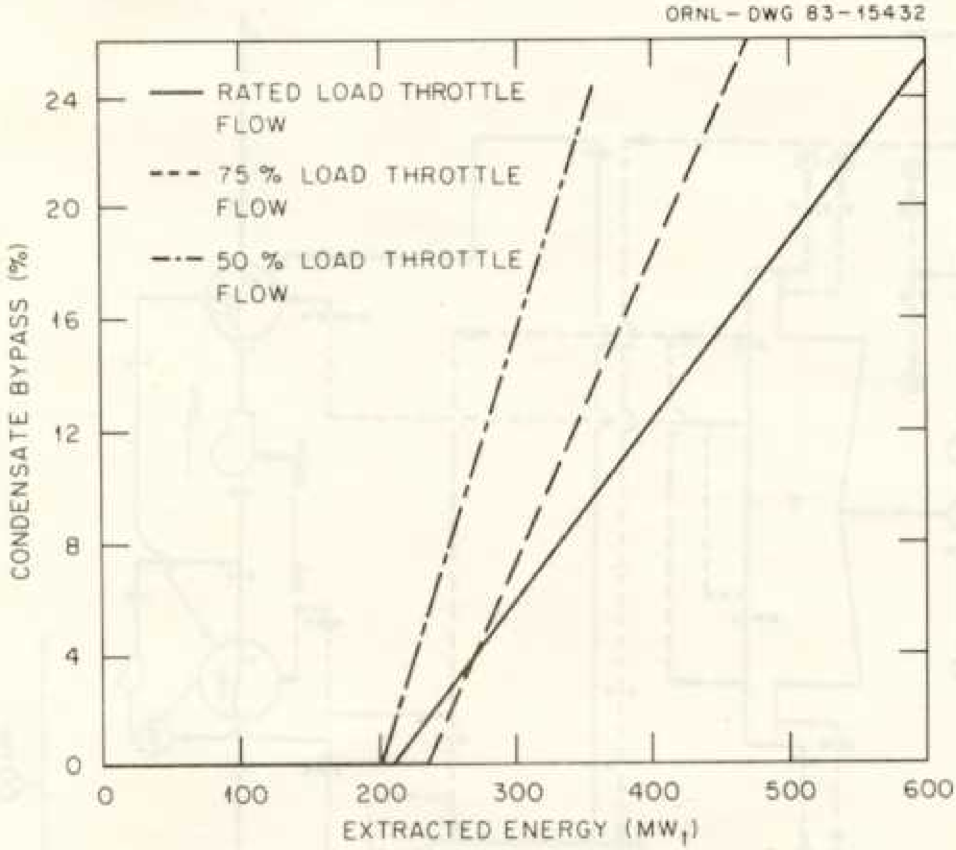


**Fig. 3.7. Extracted thermal energy vs condensate bypass.**

exhaust snouts, the crossunder pipe, and the MSR (for the original BPT cycle) became excessive. Therefore, the floating-pegged strategy was devised in which HP exhaust pressure was allowed to float until a pressure of 124 psia was reached, at which time the crossover valves would partially close to maintain 124 psia. These values of pressure relate to rated reactor power (rated throttle flow). At lower levels of reactor power, the limiting value of pressure would be reduced in direct proportion to the throttle flow. The floating-pegged unit derate performance results are also shown in Fig. 3.8.

### 3.4.1.1 Cycle I — with BP turbines

Data obtained by Westinghouse from vendors of industrial BP turbines indicated that an overall efficiency of 80% can be achieved at maximum rated steam flow. Therefore, Westinghouse included four BP turbines in the conceptual design to keep the turbines at or near maximum unit flow over a wide range of total extracted steam flow. The floating-pegged operation with BP turbines at 80% efficiency has a 112-MW(e) derate at 600-MW(t) district heating supply; the cogeneration coefficient—MW(e) loss/MW(t)—is 0.285 over the range of 350–600 MW(t).

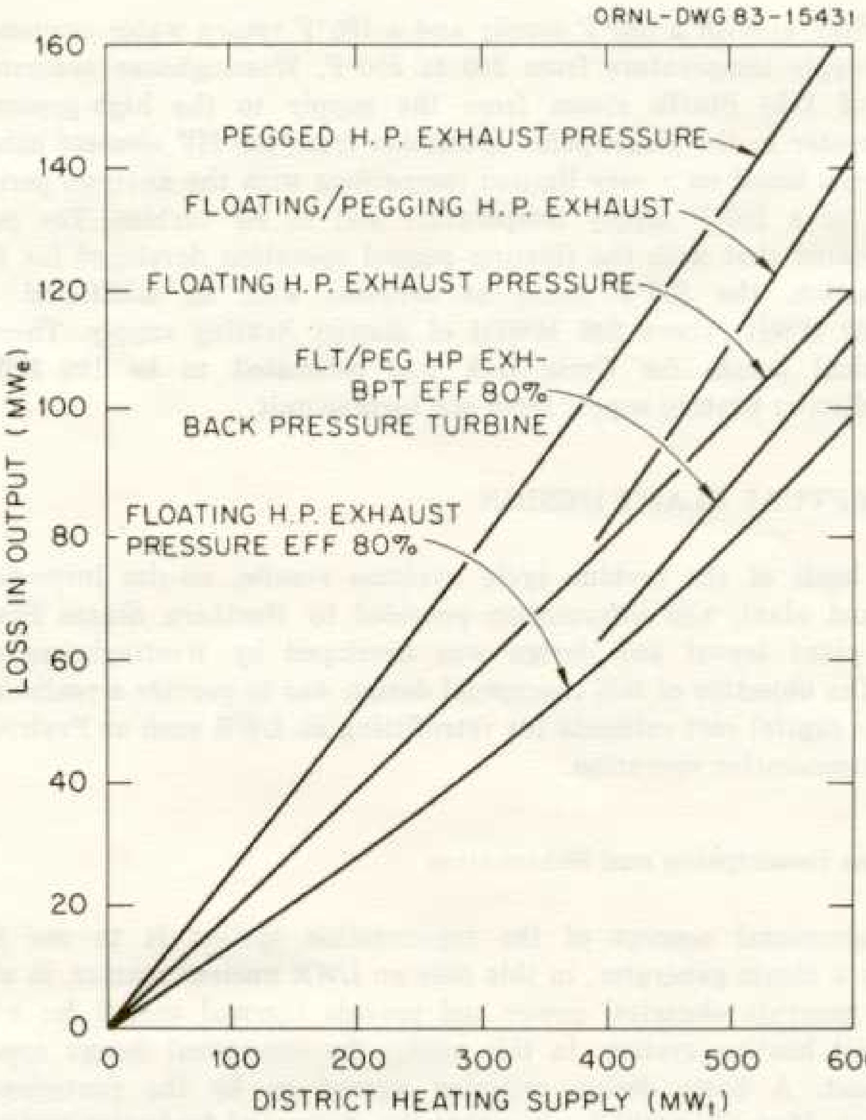


**Fig. 3.8. Loss in electrical output vs district heating load—various high-pressure exhaust extraction cycles.**

### 3.4.1.2 Cycle II-B — without BP turbines

In this cycle, steam extracted from the HP element exhaust is sent directly to the DHHXs. The floating-pegged operation results in a 142-MW(e) derate at 600 MW(t) with a cogeneration coefficient of 0.30-MW(e) loss/MW(t) over the range of 375-600 MW(t).

### 3.4.2 Cycle for 350°F Hot Water Production

So that the effect of higher hot water transmission temperatures on the overall economics could be evaluated, Cycle II-A was defined for the transmission route study (see Sect. 4) with a 350°F supply and a 180°F return water temperatures. To boost the supply temperature from 250 to 350°F, Westinghouse evaluated adding extraction of 1159 Btu/lb steam from the supply to the high-pressure No. 5 feedwater heater to the single-point extraction from the HP element exhaust. This evaluation was based on a very limited comparison with the analysis performed for Cycle II-A for a 250°F supply temperature and no BP turbine. The preliminary results indicated that with the floating-pegged operation developed for the single-point extraction, the 350°F could be supplied with an additional derate of 5 MW(e)/100 MW(t) above 240 MW(t) of district heating supply. Therefore, the total electrical derate for Cycle II-A was estimated to be 160 MW(e) at a 600-MW(t) district heating supply from one turbine unit.

## 3.5 CONCEPTUAL PLANT DESIGN

On the basis of the turbine cycle analyses results, on-site inspection of the Prairie Island plant, and information provided by Northern States Power Co., a conceptual plant layout and design was developed by Westinghouse and Fluor personnel. The objective of this conceptual design was to provide a realistic basis for developing a capital cost estimate for retrofitting an LWR such as Prairie Island to hot water cogeneration operation.

### 3.5.1 System Description and Schematics

The fundamental concept of the cogeneration system is to use the steam produced by a steam generator, in this case an LWR nuclear reactor, in such a way as to both generate electrical power and provide thermal energy for a large hot water district heating system. In this study, the conceptual design applies to an existing plant. A basic design criterion agreed on by the participants—NSP, Westinghouse, Fluor, and ORNL—was that the steam and feedwater systems for the two Prairie Island PWR units should be kept isolated from each other. This criterion meant that the district heating equipment would be duplicated for each unit for a two-unit retrofit capable of supplying up to 1200 MW(t).

Figure 3.9 shows the overall site plan as modified for district heating. The "Enlarged Plan" on the left is a section below the main floor of the turbine hall and shows the principal steam supply and condensate return lines required for the modification. Steam is extracted from the crossunder piping as shown physically in Figs. 3.10 and 3.11 and as shown schematically in Fig. 3.12. An additional major modification to main plant hardware is the replacement of the four reheat stop valves in the crossover piping with four combined reheat stop and pressure control valves.

As shown on the "Location Plan" in Fig. 3.9, the lines from and to each main power plant unit are led in elevated pipe racks outside the turbine building and across the road to a new DHHX station, one station for each main power plant unit. The new DHHX station is shown schematically in Figs. 3.13 and 3.14 for the two

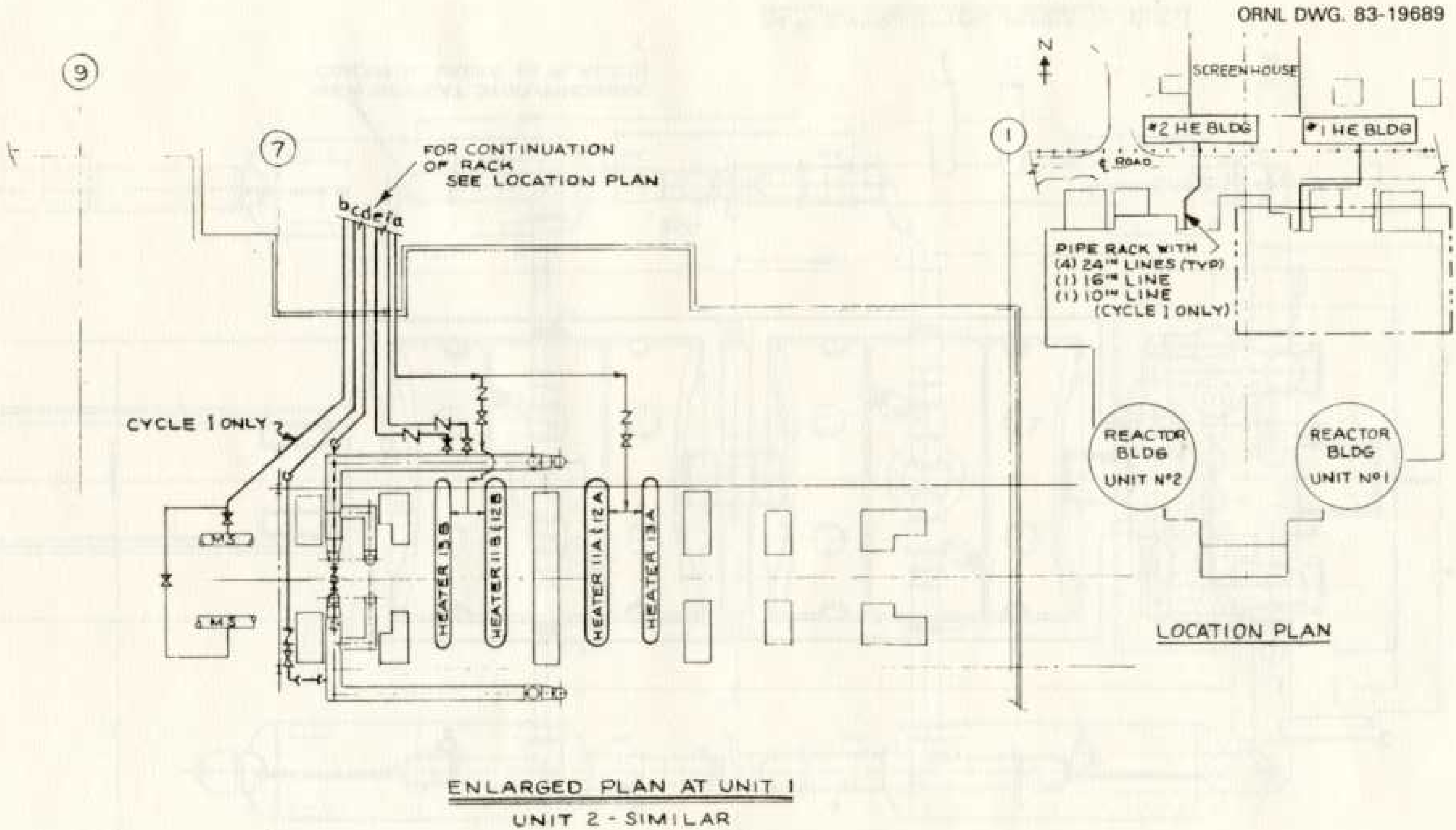


**Fig. 3.9. Piping arrangement, site and main turbine building.**

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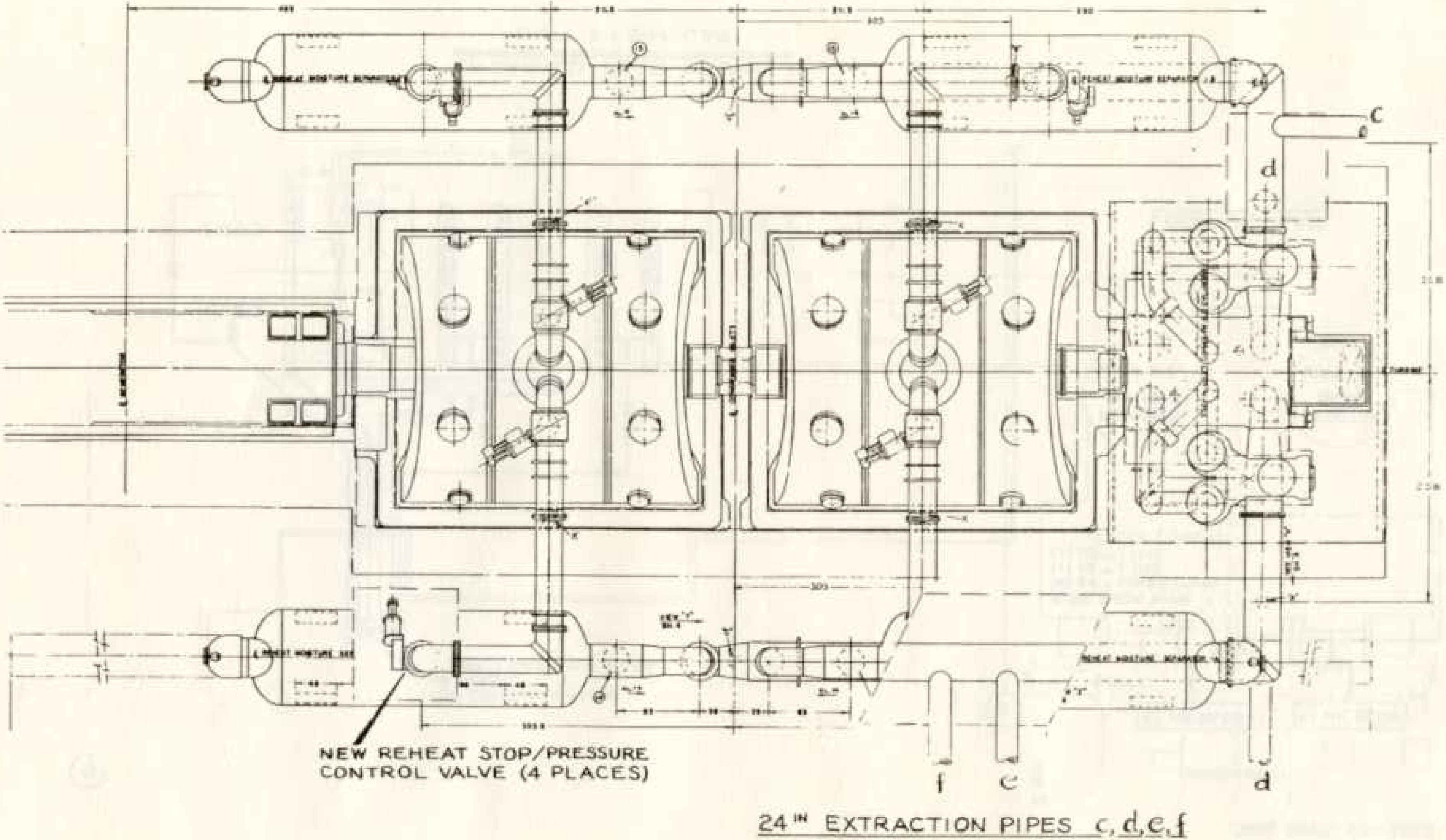


**Fig. 3.10. Plan view, main turbine modifications.**

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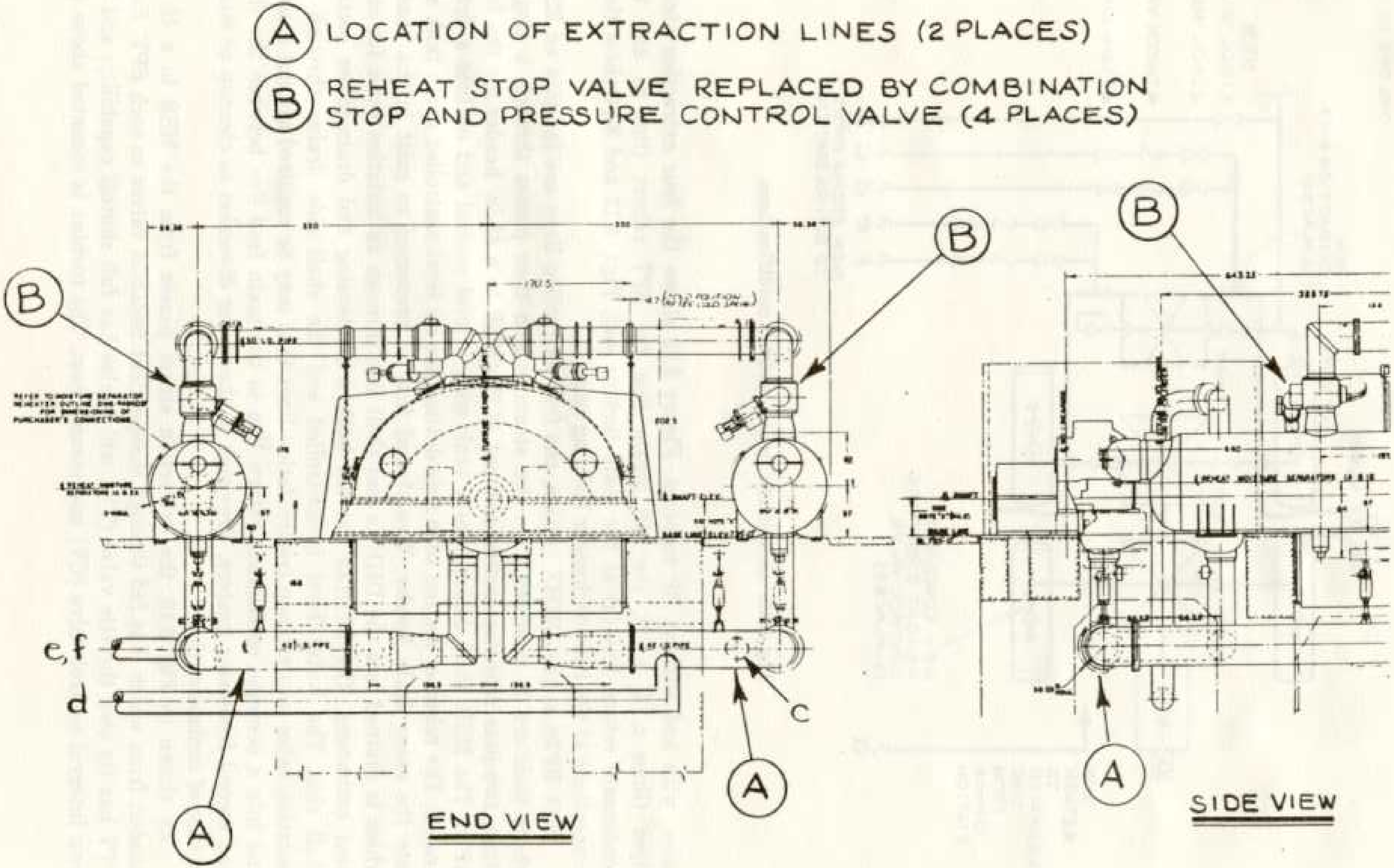


Fig. 3.11. End and side views, main turbine modifications.

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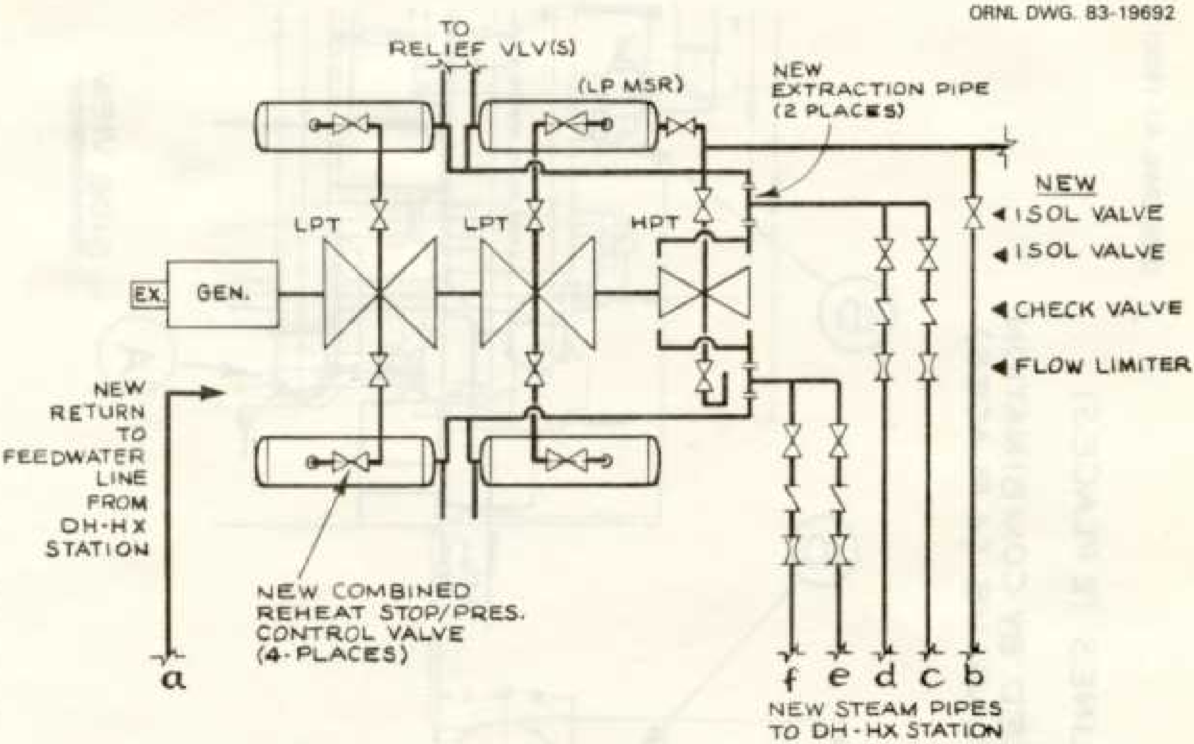


**Fig. 3.12. Schematic, main turbine modifications.**

cases, with and without BP turbines. Figure 3.12 shows the four extraction steam pipes (lines c, d, e, f), the HP steam line for BPT reheat (line b) and the condensate return line (a) at the main turbine end; Figs. 3.13 and 3.14 show their connections at the heat exchanger building end.

With BPTs at the DHHX station, the four extraction lines are fed into an MSR where moisture is removed from the steam, which then passes through a single-stage, two-pass reheater section and out of the MSR to a 42-in. header for the four BPTs. The MSR shell drains into a tank with a level control and on into a drain header. The reheater section tube side drains into its level-controlled drain tank and into the same drain header. Two-phased flow is presumed to exist in the header which is drained into the DHHXs and shell side through an isolation valve for each heat exchanger. The DHHXs contain both a condensing and drain-cooler section shell side. The water level is controlled and the shell side drains through an isolation valve into a pump (recirculation lines that may be required are not shown) and into a common condensate return line to the main feed line between the first and second feedwater heaters. Provision is shown for diversion to cleanup or waste in case of contamination.

As shown in Fig. 3.13, the extraction steam passes from the MSR to a 42-in. header, from which it is fed through independent isolation valves to each BPT. Each BPT has its own throttle valve (TV), which also has full shutoff capability, and its own integral control valve (CV) and steam chest. The turbine is mounted above the

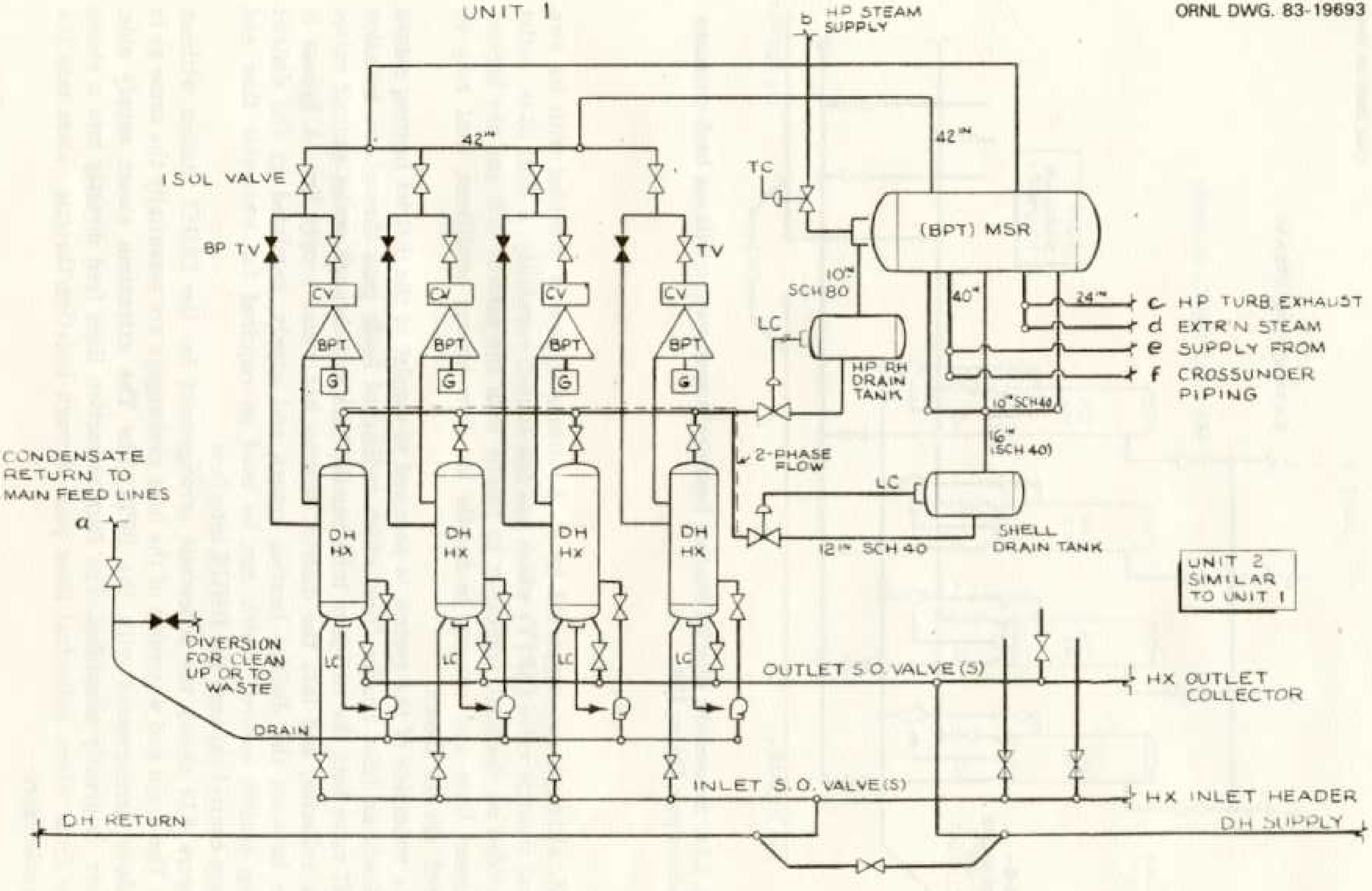


**Fig. 3.13. Schematic, district heating exchanger station with back-pressure turbines—Cycle I.**

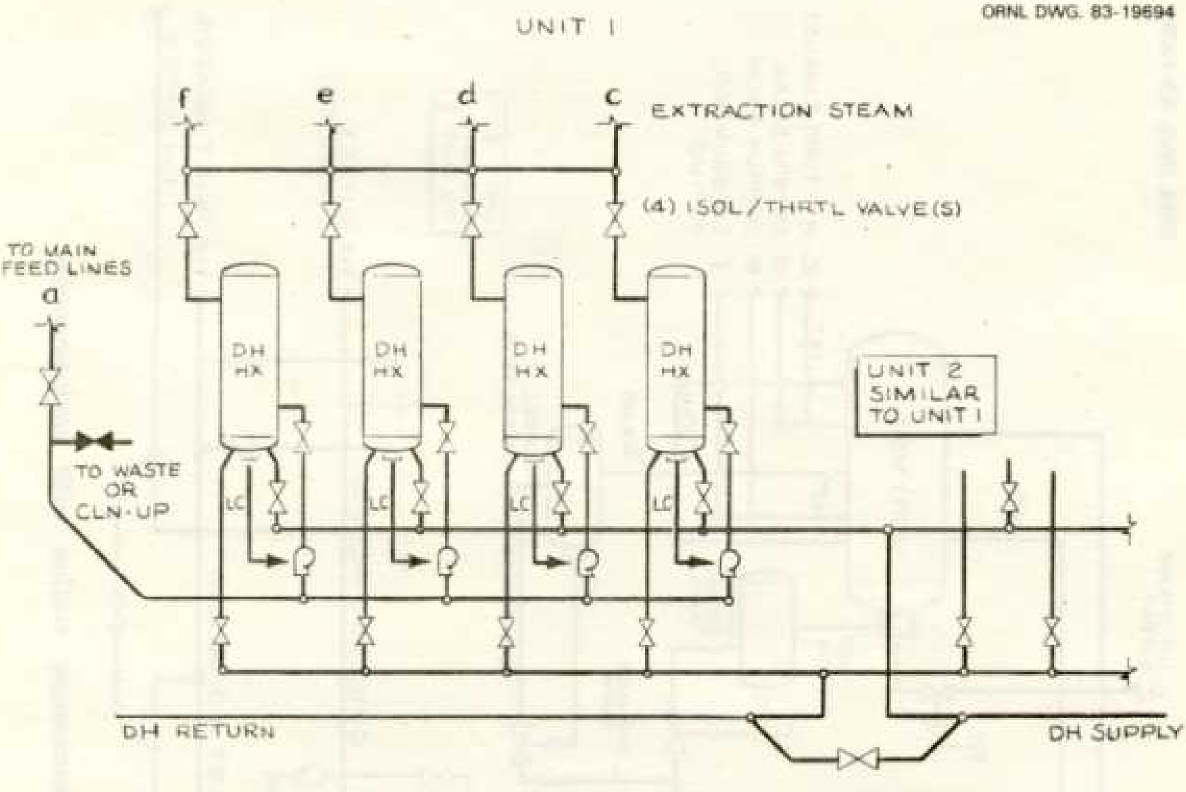


**Fig. 3.14. Schematic, district heating heat exchanger station without back-pressure turbines—Cycle IIA or IIB.**

DHHX, although not supported by it. A turbine bypass is provided with its own integral throttle valve (BPTV) which has full shutoff capability. A dual 36-in. outlet is provided on the heat exchanger to handle both the turbine exit and the bypass. Additional lines and valving to handle low-flow bypass conditions, that may be required, are not shown.

The waterside of the system is assumed to consist of the district heating return that feeds an inlet header from which individual feeds pass through an isolation shutoff valve into the two-pass tube bundle, back out through outlet shutoff valves into a collector, and into the district heating hot water supply line. A bypass is shown between the district heating return and supply, regulated by the district heating system control, which can be used as required for waterside flow and pressure control through the DHHX interface.

Figure 3.14 shows the schematic arrangement for the DHHX station without BPTs. The drain and waterside of the heat exchangers are essentially the same as in the above arrangement with the BPTs. The The extraction steam supply side, however, is greatly simplified. The four extraction lines feed directly into a steam header from which individual lines pass through isolation/throttle valves into the heat exchangers.

### 3.5.2 Basic Power Plant Changes and Additions

The following paragraphs describe the equipment and structure changes and additions needed to accommodate the proposed main plant modifications for cogeneration. The design level of this study is not such as to permit detail definitions but such that contingency factors are established in the cost estimates to cover such undeveloped items and unknowns.

#### 3.5.2.1 Reheat/stop control valves

This unit has two LP turbines, each fed through a pair of crossover pipes. Each pipe has one reheat stop valve and one interceptor valve in the line. Both of these are butterfly-type valves and are mechanically similar but receive different control logic. Neither of these valves is well suited to controlling flow in a midstroke valve position.

A design feature is that the crossover piping has a set of turning vanes at all sharp-angle turns. This Westinghouse design practice, in part based on experience and model testing, specifies that valves in the crossover lines should be three pipe diameters away from any turning vanes, if possible. This reduces the excitation on both the turning vanes and on the valves.

Since there is limited space to meet the above criteria, each of the existing four reheat stop valves would be replaced with a new butterfly-type valve to serve as the reheat stop valve as well as the steam extraction control valve. This may also require minor modification to the crossover pipe to accommodate installation of the new valves. A control-type butterfly valve would be custom designed for this application by a subcontractor noted for making this type of valve.

#### 3.5.2.2 Crossunder piping

The crossunder piping will be modified to add special tees; two crossunder lines would each have two extractions. The special tees will be designed to minimize flow disturbance and pressure drop. The result is four extraction lines, each approximately 24 in. in diameter.

#### 3.5.2.3 Extraction piping, valves, and limiters

As noted above, the steam is extracted at four locations, two on each of the crossunder pipes connecting the high-pressure turbine exhaust and the moisture separators. Each of the four 24-in. steam lines per unit would be provided with a motor-operated shutoff valve and a power-assisted check valve, which would be controlled to protect the turbine from water induction, overspeed, and loss of pressure. Flow-limiting nozzles would also be provided to limit the effects on the turbine which could result from a misoperation of the steam conversion system or a rupture of a steam supply line.

Steam piping would be routed in the turbine building on the mezzanine level and would exit the plant through the north wall approximately 30 ft above grade. All piping outside the turbine building would be supported by a structural steel pipe rack, which will span the access road and railroad tracks that run along the north end of the turbine building.

#### 3.5.2.4 Other piping

For all configurations, the drains or condensate returns from the DHHXs are pumped back to the feedwater lines after their exit from the lowest level extraction heaters. The return from the heat exchanger building is by a single 16-in. line.

For the back-pressure turbine configuration only, a high-pressure 10-in. steam line with suitable valving is run from the main turbine MSR heater supply lines to supply heating steam to the BPT MSRs in the DHHX building.

All of the above lines are carried from building to building in the same steel pipe rack used for the extraction piping.

#### 3.5.2.5 Control room/AEH and instrumentation

The main turbine-generator unit features a Westinghouse Analog Electro-Hydraulic (AEH) control system. This system would have to be modified in that the new reheat stop valve will require a more complex control logic which will permit the valves to operate at any opening position. This logic will have to include an interface with the district heating control system as well as incorporate several new data parameters which will be selected to protect the main unit turbine-generator. This will require the addition of supervisory instrumentation which may measure such points as LP turbine pressures and temperatures and HP turbine exhaust pressures and parameters at the district heating exchangers and back-pressure turbines.

### 3.5.3 DHHX and BPT Station

As shown in Fig. 3.9, the steam conversion equipment would be located north of the plant in structures erected between the railroad tracks and the screenhouse. The buildings would be steel frame construction with insulated steel panels. Two buildings would be provided, each housing the equipment for a single unit.

Four heat exchangers would occupy the grade elevation of the building. The hot water would be piped to the transmission system through openings in the building floor. Isolation valves will be provided on the inlet and outlet nozzles of each heat exchanger.

For the direct condensing configuration, Cycles II-A and B, the steam supply lines would be routed directly into the top of the heat exchangers. Isolation valves would be provided on the steam lines to the heat exchangers.

For a configuration utilizing back-pressure turbine-generators, Cycle I, a second floor would be provided in the heat exchanger building. A turbine generator would be mounted directly above each heat exchanger and would exhaust downward into the heat exchanger. A moisture separator reheater would be located at the turbine level through which the extraction steam would be passed before being routed to the back-pressure turbines.

### 3.5.3.1 BPT-generator sets

The turbine proposed for the study by Westinghouse Canada is a 10,180-kW(e) Westinghouse D136 condensing three-stage turbine with a grid valve control. The steam rate is 46.7 lb/kWh at rated conditions. A Woodward 43027 control system is specified, including a Woodward EG10P electro/hydraulic actuator. The control system is largely electronic. The generator is a totally enclosed water-to-air-cooled synchronous generator rated at 0.85 PF, 10,180 kW(e), three phases, 60 Hz, 13,000 volts, 3600 RPM.

The turbine inlet diameter is 30 in., and the exhaust diameter is 36 in. The turbine is located above its specific DHHX, which serves as a condenser. A flexible connection leads down from the turbine exhaust to the DHHX. Typical overall size allowance for the turbine would be 14 ft long, 12 ft wide, and 10 ft high, and it would weigh about 26 tons. The generator, including exciter, would be about 17 ft long and 9 ft wide, extend 8 ft above datum, and weigh approximately 50 tons.

### 3.5.3.2 MSR and crossover piping

The MSR is required at the DHHX and BPT station to properly condition the steam for the back-pressure turbine. Extraction steam from the main power plant enters the building and is led into the MSR, where chevrons are used to remove moisture. A heater bundle of 900 U-tubes, each approximately 90 ft long, provides a single stage of reheat. For this configuration, main extraction steam entry is at the end opposite the tube bundle header. The MSR is a pressure vessel around 10 ft long and 12 ft in diameter designed to the same code specification as the main turbine MSRs. Main extraction steam exits from the top at two locations. Alternate configurations could include a double unit with half-length heater tube bundles at each end and main extraction steam inlet at the bottom, both standard Westinghouse configurations.

Steam exits from the MSR through crossover pipes into a 41-in. header from which four pipes lead, each through an isolation valve, to each of the four turbines and turbine bypass piping. Design of this piping is in accord with the same standards used for main turbine crossover piping.

High-pressure, high-temperature steam supply to the reheater bundle will be taken from the plant main steam line and will be routed in a 10-in. pipeline along the same path as the 24-in. pipelines. Moisture-separator shell and tube-side condensate will be collected in drain tanks located at the district heating heat exchange level to be discharged to the DHHXs. The level will be controlled in the drain tanks by control valves on the tank drain lines.

#### 3.5.3.3 Turbine and bypass throttle valves

The valving for the BPT would include a combined throttle-stop valve and a set of integral control valves for each back-pressure turbine. These valves would be similar to what the industry has frequently used for small steam turbines. The turbines would also feature a bypass valve which would be a semicommercial valve designed to control the pressure going to the heat exchanger. For protection, each combined bypass valve and BPT would have an isolation valve, expected to be commercially available.

#### 3.5.3.4 DHHX with BPT

The DHHXs are similar to feedwater heaters, and the same optimazation techniques (in this case, a proprietary Westinghouse computer program) has been used in their specification. Design conditions for the case with a BPT are as follows.

- Thermo design:

  Water flow = $22.6 \times 10^6$ lb/h (total for four heat exchangers)
  $T_{in}$ = 160°F—inlet water temperature
  $T_{out}$ = 250°F—outlet water temperature
  $T_{sat}$ = 263°F—steam saturation temperature
  $T_{dr}$ = 170°F—drain cooling section outlet temperature
  TTD = 13°F—terminal temperature difference in
  condensing section
  Steam flow = 1,880,000 lb/hr at 37.2 psia and H = 1191 Btu/lb (total for four)

- Mechanical design:

  Temperature = 400°F
  Pressure = 200 psig (test at 300 psig)
  Tube metal = stainless steel

The resultant DHHX has a total tube area of 17,676 ft$^2$, uses two tube passes, has a shell inside diameter of 64 in., and has a total length of 43 ft.

#### 3.5.3.5 Steam, drain, and condensate systems

Drains from the MSR drain tanks feed into the DHHX and become mixed with the condensate.

The condensate will be returned to the main turbine cycle by condensate pumps located below the heat exchanger building in the district heating piping tunnel.

To avoid detrimental effects of possible contamination of the return condensate caused by DHHX leakage from the water side into the shell, there would be continuous monitoring of the condensate from each DHHX. If the contamination level so indicates, the offending DHHX would be immediately isolated and the contaminated water diverted to waste or to a special clean-up station as appropriate.

#### 3.5.3.6 District heating waterside interface

The interface with the district heating water system is at the DHHX nozzles and does not include any waterside piping, valves, instrumentation, control, or installation, including the DHHX bypass. The cogeneration system would accept thermal load demand from the district heating system control center and attempt to satisfy it by steam extraction ultimately fed to the DHHX. Meeting electrical demand could have priority, along with any limit conditions. The district heating system, not the Prairie Island complex, is responsible for water pressures, temperatures, and flows.

#### 3.5.3.7 Control systems and instrumentation

An independent control station would be developed for monitoring and controlling the heat exchanger and back-pressure turbines. An electronic integrated control system would consist of the four separate controls for each of the BPTs and a master control for the entire heat exchanger and BPT complex. This system would be interconnected with the district heating station and the main turbine-generator AEH control system. Protective supervisory instruments on the BPTs and heat exchangers would also input to this control system.

#### 3.5.3.8 Electrical power systems

Each turbine includes an automatic voltage regulator, limiters, controls, etc. Additional electrical equipment, including transformers, breakers, alarms, controls, busses, instrumentation, etc., are supplied to connect the BPT-generators independently into the network.

### 3.5.4 DHHX Station Without BPT

The basic configuration concept for Cycles II-A and II-B is the same as for the back-pressure turbine except for the elimination of these turbines, the second floor for housing them, and the associated steam-handling and support equipment.

Eliminated would be the turbines, MSR, MSR heating and drain system, related piping, etc. (see Fig. 3.14).

### 3.5.4.1 Inlet piping and control valves

The piping and valving for the heat exchangers without BPTs present would consist of a set of isolation and throttle valves for each heat exchanger. These valves would be similar to those on the BPTs. Inlet piping is to a header with separate leads to each isolation valve, throttle valve, and DHHX.

### 3.5.4.2 DHHX

The DHHXs are similar to feedwater heaters, and the same optimization techniques (in this case, a proprietary Westinghouse computer program) have been used in their specification. Two cases have been evaluated for Cycle II-B, which include heat exchangers of different sizes. The first case had a low (10°F) terminal temperature difference (TTD) and produced a hot water return temperature as high as 329°F, while the second one had a smaller heat exchanger and produced a maximum supply temperature of 250°F with a high (89.3°F) terminal temperature difference. Design conditions at 600 MW(t) for the high TTD case, which had the lowest installed cost, are as follows.

- Thermo design:

  Water flow = $22.6 \times 10^6$ lb/h (total for four)
  $T_{in} = 160°F$
  $T_{out} = 250°F$
  $T_{sat} = 339.3°F$
  $T_{dr} = 175°F$
  TTD = 89.3°F
  Steam flow = 2,080,000 lb/hr at 117 psia and H = 1089 Btu/lb (total for four)

- Mechanical design:

  Temperature = 400°F
  Pressure = 200 psig (test at 300 psig)
  Tube metal = stainless steel

- Resultant configuration:

  Total tube area = 7917 $ft^2$
  Shell inside diameter = 65 in., length = 22 ft

For Cycle II-A with supply/return temperatures of 350°/180°F, the DHHX thermo design was not determined. However, the required water flow for 600 MW(t) would be $11.8 \times 10^6$ lb/h.

## 3.6 COST ESTIMATES

Costing in this study assumes a commercial environment in which the buyer's cost is normally determined by fixed-price proposals, with escalation, in competition with a buyer's bid specification. A fixed-price contract assumes that the equipment supplier is willing to take certain risks to achieve his expected profit, both matters of proprietary concern in a free market. The pricing exercise in this case is based on a scenario in which Fluor acts as the A&E for the buyer, as well as the construction and installation contractor. Westinghouse ST-G Division acts in their traditional role of supplier of new equipment, modifications, engineering and services for steam turbine, generators, and certain ancillaries such as MSRs, hydraulics, regulators, controls, etc. Westinghouse could also be a competitive supplier of installation services through its Power Generation Services Division, but for purposes of this scenario it was assumed for convenience that Fluor won the total installation contract.

The results of the pricing exercise in mid-1982 dollars are shown in Table 3.1 for the Westinghouse-supplied equipment and in Tables 3.2 and 3.3 for the Fluor-supplied equipment and services. Fluor pricing is for a single system and should be doubled for twin systems. The prices for Cycle II in Table 3.3 were developed for Cycle II-B with DHHXs sized for 250°F/160°F supply/return hot water temperatures. For Cycle II-A, the higher supply/return temperatures would require larger and more costly DHHXs. However, this increased cost would be a small fraction of the total price estimated in Table 3.3; therefore, the Fluor prices in Table 3.3 were used for both Cycles II-A and II-B.

**Table 3.1. Westinghouse-supplied equipment pricing (mid-1982 dollars)**

| | | Cycle I | Cycle II-A or II-B |
|---|---|---|---|
| Main power plant equipment— Valves and fittings, piping modifications, control system changes, and engineering and support | Single system[a] | 1,423,000 | 1,423,000 |
| | Twin system | 2,846,000 | 2,846,000 |
| DH station equipment[b]— Back-pressure T-G set (complete); moisture separator reheater; piping, valves, and fittings; DH station supervisory control; and engineering and support | Single system | 15,215,000 | 155,000 |
| | Twin system | 30,431,000 | 311,000 |
| Total | Single system | 16,638,000 | 1,578,000 |
| | Twin system | 33,277,000 | 3,157,000 |

[a]Single system is for either Prairie Island Unit 1 or 2; twin system is for both.
[b]DH station equipment for Cycles II-A and II-B is DH station supervisory control only.

Table 3.2. Fluor-supplied equipment and services pricing—Cycle I (mid-1982 dollars)

| | Field labor | Field material | Manufacturing equipment | Total cost |
|---|---|---|---|---|
| Building | 1,224,000 | 1,099,000 | | 2,323,000 |
| Equipment | 52,000 | | 1,058,000 | 1,110,000 |
| Piping and insulation | 620,000 | 981,000 | | 1,601,000 |
| Electrical and instrumentation and control | 166,000 | 86,000 | 80,000 | 332,000 |
| Subtotal | 2,062,000 | 2,166,000 | 1,138,000 | 5,366,000 |
| Field indirects[a] | | | | 4,124,000 |
| Engineering and home office services | | | | 2,372,000 |
| Contingency at 25% | | | | 2,966,000 |
| Project total (mid-1982 dollars) | | | | 14,828,000 |
| Cycle I equipment: four condensate return pumps, two drain tanks, and four heat exchangers; see Table 3.1 for Westinghouse equipment not shown here. | | | | |

[a]Field indirects include insurance and taxes, union fund, temporary construction and facilities, supervision, equipment rental, small tools, and quality assurance and control.

## 3.7 SUMMARY AND CONCLUSIONS

The turbine and plant modification study, conducted by Westinghouse Electric Corporation, provides a first-level definition of the necessary equipment, facilities, modification, installation, and operational changes involved in retrofitting an existing PWR plant to cogeneration operation. Basic performance, arrangement, and pricing are described for several cogeneration systems in which steam is extracted from the crossunder pipes between the HP turbine exhausts and the moisture separator reheaters of the main turbine. Cycle I feeds the extracted steam into four back-pressure turbines which exhaust into a separate district heating heat exchanger and heat district heating water to 250°F. An alternate Cycle II feeds the extracted steam directly into the four heat exchangers, in which Cycle II-A heats the water to a temperature of 350°F; Cycle II-B, with a smaller heat exchanger, heats the water to a maximum value of 250°F. Temperatures lower than the maximum are always obtainable by flow adjustment and bypassing.

No feasibility problems were discovered in the study. There is no apparent effect on reactor operations, although a certain amount of increased complication would result in power plant operations. The quantitative effect of the cogenerator modifications on main power plant availability was not determined. Availability

**Table 3.3. Fluor-supplied equipment and services pricing—Cycle II (mid-1982 dollars)**

| | Field labor | Field material | Manufacturing equipment | Total cost |
|---|---|---|---|---|
| Building | 909,000 | 784,000 | | 1,693,000 |
| Equipment | 50,000 | | 760,000 | 810,000 |
| Piping and insulation | 514,000 | 797,000 | | 1,311,000 |
| Electrical and instrumentation and control | 150,000 | 70,000 | 80,000 | 300,000 |
| Subtotal | 1,623,000 | 1,651,000 | 840,000 | 4,114,000 |
| Field indirects[a] | | | | 3,246,000 |
| Engineering and home office services | | | | 1,840,000 |
| Contingency at 25% | | | | 2,300,000 |
| Project total (mid-1982 dollars) | | | | 11,500,000 |
| Cycle II equipment: four condensate return pumps and four heat exchangers; see Table 3.1 for Westinghouse equipment not shown here. | | | | |

[a]Field indirects include insurance and taxes, union fund, temporary construction and facilities, supervision, equipment rental, small tools, and quality assurance and control.

should be further studied because greater design detail might develop in an actual application of the concept. Within the scope of the study, there was no problem foreseen in respect to nuclear safety or licensing. Typical performance and price values in round numbers are summarized in Table 3.4 for a single main turbine unit system at the maximum thermal energy supply of 600 MW(t).

In effect, the performance advantage of Cycle I with BPT over Cycle II without BPT is priced at about $600 per kW(e). The marginal heat rate approaches 3412 Btu/kWh.

**Table 3.4 Summary of retrofit 560-MW(e) PWR performance and price results[a]**

| | Net MW(e) reduction | DH water supply/return temperatures (°F) | DH water flow ($10^6$ lb/h) | Project price per unit (mid-1982 dollars) |
|---|---|---|---|---|
| Cycle I | 112 | 250/160 | 22.6 | 31,466,000 |
| Cycle II-A | 160 | 350/180 | 11.8 | 13,462,000 |
| Cycle II-B | 142 | 250/160 | 22.6 | 13,462,000 |

[a]For 600-MW(t) district heating (DH) supply capacity.

# 4. TRANSMISSION SYSTEM

## 4.1 INTRODUCTION

For an existing, remotely sited electric power plant to be considered as a source of thermal energy via conversion to cogeneration operation, a hot water transmission system would be constructed and operated to transport thermal energy to the metropolitan district heating system load center. Although there are no hot water transmission systems currently operating in the U.S., there are large hot water transmission systems operating in Western Europe—see Appendix A—linking power plants with well-developed hot water district heating systems. Also there are many large-scale, fluid-transport systems operating over long distances in the U.S. such as the regional water transport system in California, interstate oil pipelines, and coal-water slurry pipelines. Therefore, the basic feasibility of construction and operation of a hot water transmission system has been established.

However, it has been recognized from the beginning of this study that the cost contribution from a transmission system 30 to 40 miles in length to link the Prairie Island plant with the Twin Cities would be significant. It was therefore decided that (1) a study of transmission route option be conducted to determine the most economical route and construction methods and (2) the study be conducted by engineering consultants experienced in design and construction of hot water district heating transport systems. The latter requirement was met by the engineering consulting firms of Metcalf & Eddy, Inc., Boston, Massachusetts, and FVB, Vasteras, Sweden, performing the Task 4 study—*Transmission Line Route Options*. These firms combine extensive design and construction management experience in water systems of Metcalf & Eddy with the similar experience of FVB, in European hot water district heating distribution and transmission systems.[1]

This section is a summary of the Metcalf & Eddy, Inc./FVB Task 4 report, which appears in Appendix C.

### 4.1.1 Performance Requirements

The performance requirements for the transmission system interface directly with the performance requirements of the turbine and plant modifications study in Sect. 3.1.1. Specifically, two capacity levels were established—600 and 1200 MW(t)—and two temperature levels at the maximum heat transfer rate: (1) low temperature supply/return of 250/160°F for "direct" transfer at the load center to the distribution network and (2) a high temperature supply/return of 350/180° for "indirect" transfer to 250/160°F water-to-water heat exchangers as the load center.

The 250/160°F supply/return temperatures were selected to conform with the design conditions of the new hot water district heating system in St. Paul, Minnesota.[2]

### 4.1.2 Objectives

The principal objectives of this study were as follows:

1. determine, on a first-level basis, the conceptual design of the most economical transmission route between the Prairie Island plant and the High Bridge plant in St. Paul; and
2. develop a cost estimate, in mid-1982 dollars, for construction of the transmission system and the direct operating costs.

## 4.2 DESCRIPTION OF THE TRANSMISSION SYSTEM

### 4.2.1 Overall System Description

Two alternative transmission system modes have been investigated. The first system is a direct system, whereby hot water from the transmission system is transmitted to the local district heating networks within the consumption areas, without the use of heat exchangers. The second system is an indirect system in which heat exchange must occur between the transmission system and the district heating network serving the customers. Two capacity levels, 600 MW(t) and 1200 MW(t), respectively, have been calculated for each mode.

The transmission system for this hypothetical study receives heat by steam extraction from the HP turbine exhaust of the nuclear power plant at Prairie Island. Heat transfer occurs in DHHX where the steam is cooled with incoming water from the transmission system. The incoming water—the hot water system's return water—is heated from about 160°F (70°C) to about 250°F (120°C) under direct supply mode, and from about 180°F (80°C) to about 350°F (175°C) in the indirect mode. Thermal energy transferred to the water in the transmission system in this manner is transported through the proposed transmission system to consumer areas where it is delivered to existing and planned district heating systems. In the direct heating system, the hot water is cooled to about 160°F, after which it is pumped, through return pipes, back to the DHHXs at Prairie Island for reheating.

The thermal energy that is transferred to the district heating systems is transported through them to individual consumers where, after one or more additional heat exchangers, it is delivered to radiators and other water appliances.

Thus, the overall system consists of a number of closed cycles, connected in series, in which heat transfer occurs from cycle to cycle in heat exchangers. Heat is exchanged in the heat exchangers without the heat-transport media coming in contact with each other. In this system, separation of the media plus the long transmission times and isolation features protect against any transfer of radioactivity from Prairie Island to the individual consumer heating systems.

The basic principles of the heat transport systems are illustrated by the diagrams in Fig. 4.1.

### 4.2.2 Transmission System Descriptions

This study deals with the hot water transmission system between the DHHXs at Prairie Island and the load center for the local district heating systems at the High Bridge plant in St. Paul.

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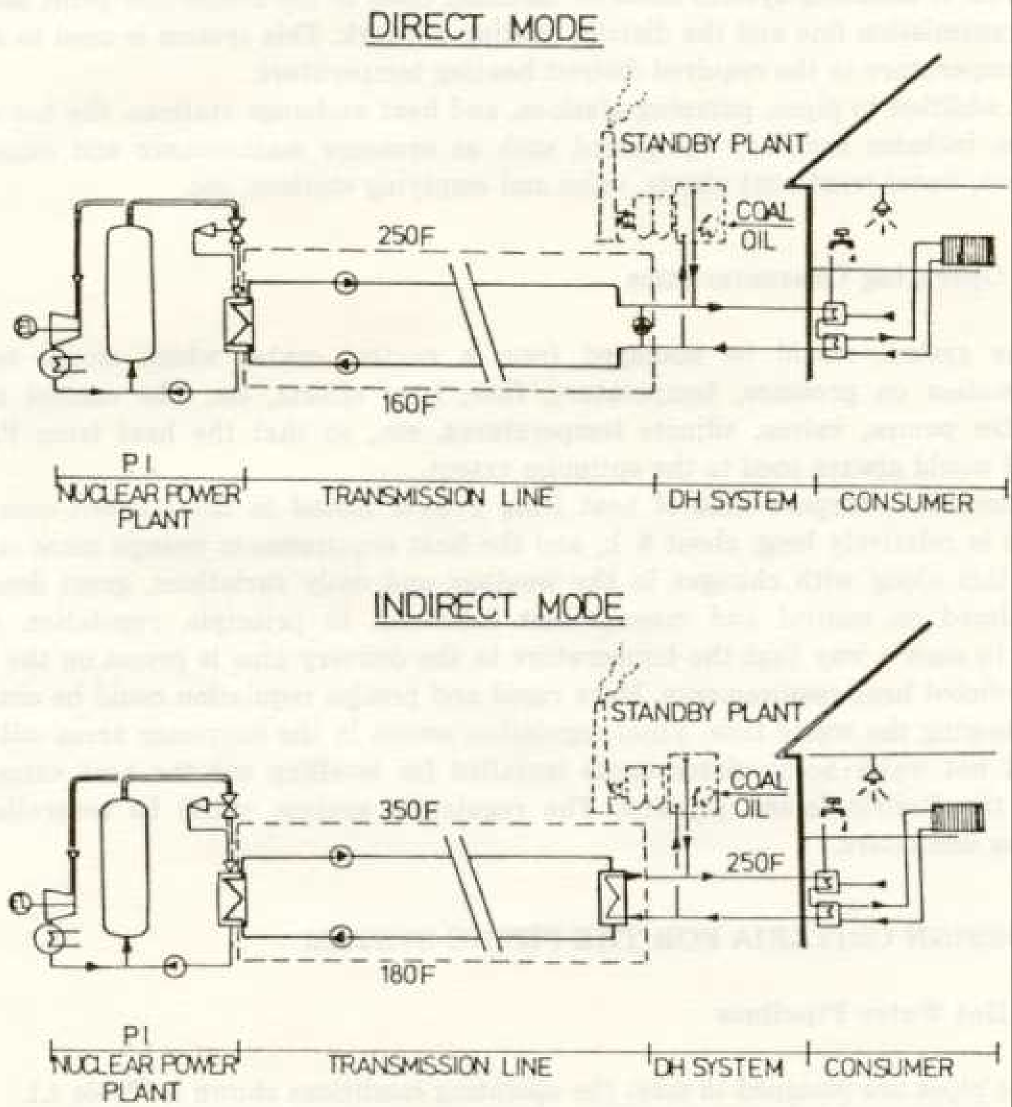


**Fig. 4.1. Schematic diagrams of heat transport system.**

The transmission system consists of insulated pairs of steel pipes, the supply pipe, and the return pipe. The pipes would be placed in buried culverts, in tunnels, or on low supports aboveground.

The pipeline from Prairie Island to the High Bridge plant in St. Paul has a total length of about 36 miles (60 km). The pipe diameters in the direct mode are about 66 in. (1600 mm) for a power level of 1200 MW(t) and 48 in. (1200 mm) for a power level of 600 MW(t). In the indirect mode, the pipe diameters are 48 in. (1299 mm) and 36 in. (900 mm).

In the direct mode, pipe is designed for a working pressure of 235 psi or 16 bar and a maximum temperature of 250°F (120°C). The comparative criteria for the indirect mode are 370 psi or 25 bar and 350°F (175°C). In both modes, piping is designed for 100% vacuum.

Four pumping stations along the line and one station at Prairie Island are required. A shunting system must be installed close to the connection point between the transmission line and the district heating network. This system is used to adjust the temperature to the required district heating temperature.

In addition to pipes, pumping stations, and heat exchange stations, the hot water system includes auxiliary equipment such as pressure maintenance and expansion systems, water treatment plants, valve and emptying stations, etc.

### 4.2.3 Operating Characteristics

The system would be managed from a control center which would receive information on pressure, temperature, flow, heat effects, etc. The control center operates pumps, valves, adjusts temperatures, etc., so that the heat from Prairie Island would always used to the optimum extent.

Since the transport time of heat from Prairie Island to the farthest consumer region is relatively long, about 5 h, and the heat requirements change more rapidly than this along with changes in the weather and daily variations, great demands are placed on control and management functions. In principle, regulation would occur in such a way that the temperature in the delivery pipe is preset on the basis of predicted heat requirements. More rapid and precise regulation could be obtained by changing the water flow. Final regulation occurs in the consumer areas with the use of hot water accumulator tanks installed for levelling out the heat extraction from the Prairie Island pipeline. The regulation system would be controlled by process computers.

## 4.3 DESIGN CRITERIA FOR THE PIPING SYSTEM

### 4.3.1 Hot Water Pipelines

The pipes are designed to meet the operating conditions shown in Table 4.1.

The pipeline dimensions for the various systems and capacities are shown in Table 4.2. The pipeline capacity was dimensioned for a favorable water velocity at

**Table 4.1. Pipe design parameters**

| | Indirect mode | Direct mode |
|---|---|---|
| Maximum pressure | 25 bar (370 psi) | 16 bar (235 psi) |
| Highest temperature | 350°F (175°C) | 250°F (120°C) |
| Vacuum | 100% | 100% |

**Table 4.2. Pipeline sizes and flow rates**

| | Direct mode | | Indirect mode | |
|---|---|---|---|---|
| | 600 MW(t) | 1200 MW(t) | 600 MW(t) | 1200 MW(t) |
| Pipe diameter, in. | 48 | 66 | 36 | 48 |
| Pipe diameter, mm | 1200 | 1600 | 900 | 1200 |
| Total flow rate | | | | |
| $10^6$ lb/hr | 22.6 | 45.2 | 11.8 | 23.6 |
| kg/s | 2850 | 5700 | 1490 | 2980 |

a given flow rate. The maximum water velocity will be 10 ft/sec, or 7 miles per hour (3 m/s, about 10 km/h).

To absorb pipe expansion due to heat differential, U-, Z-, and L-shaped expansion elements will be placed symmetrically between fixed points. Basically, U-shaped elements would be used in tunnels and Z- and L-shaped elements would be used in concrete culverts. The radii of the pipe bends included in the expansion elements will be 1.5 times the outer diameter of pipe.

Expansion elements, shown in Fig. 4.2, were dimensioned for the expansion lengths given in Table 4.3.

The distance between support points is as given in Table 4.4.

In calculating the pipe thicknesses, in addition to the considerations above, steel quality SS141430* was assumed for pipes and bends in the direct system, and steel quality SS142101 in the indirect system. Consideration has also been given to surge pressures resulting from pressure changes due to sudden shutdown of a pump or similar operating situations. The calculated thicknesses in Table 4.5 include 1 mm to allow for corrosion.

These dimensions are in accordance with European standards. Preliminary calculations based on Power Pipe Code ANSI/ASME B31.1 indicate that pipe walls would be about 15% thicker in the United States. The appropriate steel quality would be a A-155 type CMS-H80. Cost estimates are based on U.S. criteria.

*Swedish material standards.

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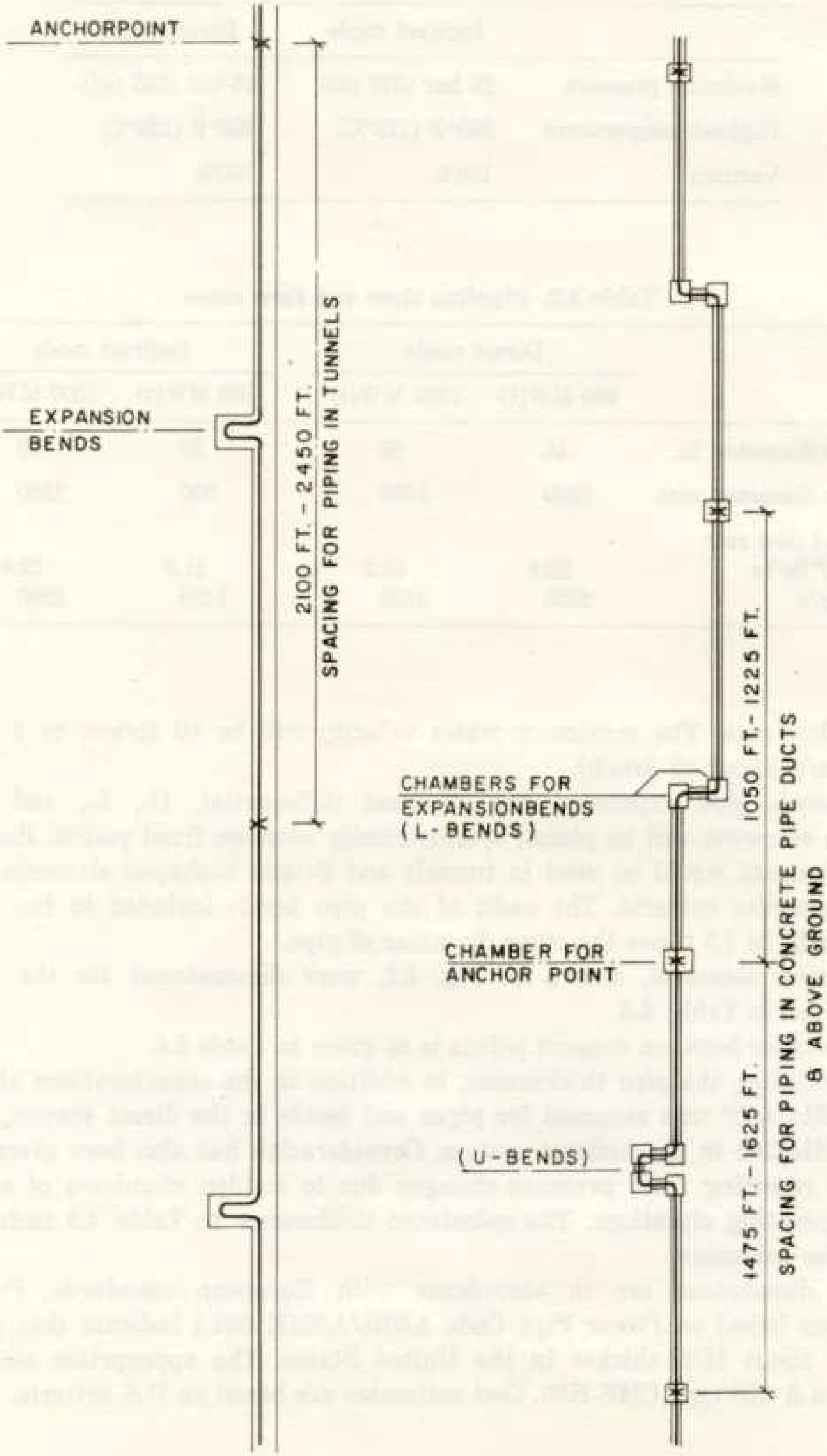


**Fig. 4.2. Prairie Island hot water feasibility study piping schematic of spacing requirements for expansion.**

**Table 4.3. Expansion elements**

| Pipe diameter | | Distance between expansion elements | | | |
|---|---|---|---|---|---|
| | | U-element | | Z-element | |
| in. | mm | ft | m | ft | m |
| 36 | 900 | 1475–2125 | 450–650 | 1050 | 325 |
| 48 | 1200 | 1475–2350 | 450–750 | 1050 | 375 |
| 66 | 1600 | 1475–2350 | 450–750 | 1225 | 375 |

**Table 4.4. Spacing of supports**

| Pipe diameter | | Distance between supports | |
|---|---|---|---|
| in. | mm | ft | m |
| 36 | 900 | 40 | 12 |
| 48 | 1200 | 60 | 18 |
| 66 | 1600 | 80 | 24 |

**Table 4.5. Pipe wall thickness**

| Nominal diameter (mm) | Outer diameter (mm) | Pipe wall thickness (mm) |
|---|---|---|
| 1600 | 1625.6 | 11.5 |
| 1200[a] | 1219.2 | 9.5 |
| 1200[b] | 1219.2 | 9.5 |
| 900 | 914.4 | 9.0 |

[a]Direct mode.
[b]Indirect mode.

### 4.3.2 Thermal Insulation

All pipes and fittings in tunnels and culverts would be thermally insulated with mineral wool or polyurethane foam insulation. In tunnels, the insulation would be made of mineral wood pads with a density of about 2.5 lb/ft$^3$ (40 kg/m$^3$), or polyurethane foam would be used.

Preliminary economic evaluations have been made for the insulation thickness. The present calculations show that the optimum range is broad and that changes in insulation thicknesses have only a marginal effect on costs. The cost calculations have been based on the insulation thicknesses in Table 4.6.

With the planned insulation, heat losses would be limited to about 3% of the quantity of heat transported. Approximately this same quantity of thermal energy would be imparted to the water during pumping.

**Table 4.6. Pipe insulation**

| Pipe diameter | | Insulation thicknesses | | | |
|---|---|---|---|---|---|
| | | Supply pipe | | Return pipe | |
| in. | mm | in. | mm | in. | mm |
| 66 | 1600 | 7 | 180 | 4 | 100 |
| 48 | 1200 | 6 | 150 | 4 | 100 |
| 48 | 1200 | 5 | 120 | 3.5 | 90 |
| 36 | 900 | 4 | 100 | 3 | 80 |

## 4.4 CONSTRUCTION TECHNIQUES

### 4.4.1 General

In Sweden and Scandinavia, pipes for carrying thermal energy in the form of hot water over considerable distances have been studied for a number of projects. Experience acquired over the years from the development of local distribution systems in cities and suburbs has been used as the basis for this study. The development of such systems has been in progress in Sweden for more than 25 years.

Experience to date has shown unanimously that;

1. longitudinally or helically welded steel pipes with adequate wall thickness to withstand loads of both internal and external pressure should be used;
2. the longitudinal expansion of the pipes due to temperature fluctuation should not be absorbed by means of axial temperature equilizers but by the use of expansion-absorbers in the form of U-elements or Z-elements;
3. thermal insulation should be made of mineral wool or polyurethane foam;
4. the pipes must be protected against mechanical damage and corrosive attack by laying them in concrete culverts or rock tunnels. They may be laid aboveground, but in that case both pipes and insulation must be protected by a water-tight but not diffusion-tight protective casing;
5. the media water must be kept under control and be of such quality that internal corrosion does not occur; and

6. material stresses arising due to transient flow (water hammer) can be kept within acceptable limits by special measures.

Four major methods of installing the hot water mains have been considered for the alternate routes. These methods are cut and cover with a pipe duct, tunnel in the bedrock, stanchions with the pipe above the ground, and pipe bridges for the major river crossings.

Brief descriptions of each method and its advantage are given below. Table 4.7 shows the average unit cost for construction for these methods for the anticipated geotechnical and general construction conditions. The general advantages and disadvantages of each viable construction alternative were weighed in making the recommendation for each section.

**Table 4.7. Unit cost of construction methods[a]**

| Method of construction | Pipe diameter (in.) | Unit cost (dollars per linear foot) | | |
|---|---|---|---|---|
| | | Labor | Materials[b] | Total |
| Pipe duct | 36 | 132 | 528 | 660 |
| | 48 | 151 | 604 | 755 |
| | 66 | 188 | 754 | 942 |
| | 36 | 249 | 250 | 499 |
| Stanchion | 48 | 290 | 291 | 581 |
| | 66 | 387 | 388 | 775 |
| | 36 | 1318 | 565 | 1883 |
| Tunnel (dolomite) | 48 | 1491 | 639 | 2130 |
| | 66 | 1564 | 670 | 2234 |
| | 36 | 433 | 649 | 1082 |
| Tunnel[c] (St. Peter sandstone) | 48 | 528 | 792 | 1320 |
| | 66 | 569 | 854 | 1423 |
| | 36 | 1309 | 7415 | 8724 |
| Bridge (25 ft wide) | 48 | 1318 | 7471 | 8789 |
| | 66 | 1334 | 7562 | 8896 |

[a]Mid-1982 dollars, Engineering News Record Index = 3100.

[b]See Fig. 4.3 for spacing of anchors, U-bends, and Z-bends. For above-ground pipe, 85% U-bends and 15% Z-bends was assumed; for pipe in ducts, 30% U-bends and 70% Z-bends was assumed; and 100% U-bends was assumed in tunnels.

[c]Does not include cost for dewatering.

### 4.4.2 Cut and Cover with Culverts

The cut-and-cover method is the preferred, typical method of installing pipelines. The pipelines are located in a concrete culvert duct to protect the insulation. The advantages of this method are that no special construction equipment is required and that a mass balance of excavated material and cover over the duct can be achieved. Also, by maintaining a shallow trench, the need for dewatering on a permanent basis can be eliminated. In general, the top of the culvert should be buried a minimum of 3 ft below the ground surface.

As shown in Fig. 4.3, the pipe duct can be either a precast section or a cast-in-place duct with waterstops. The waterstop in the cast-in-place duct will provide a watertight duct where it is exposed to high groundwater. The precast section, even with a rubber seal at the joint, is more subject to some leakage and, therefore, is less suitable in areas were high groundwater is anticipated. In either case, the pipe duct must be designed so that flooding of the culvert cannot occur. In addition, the duct must be able to withstand all stresses caused by ground pressures above and from traffic, including all types of agricultural and forestry machinery.

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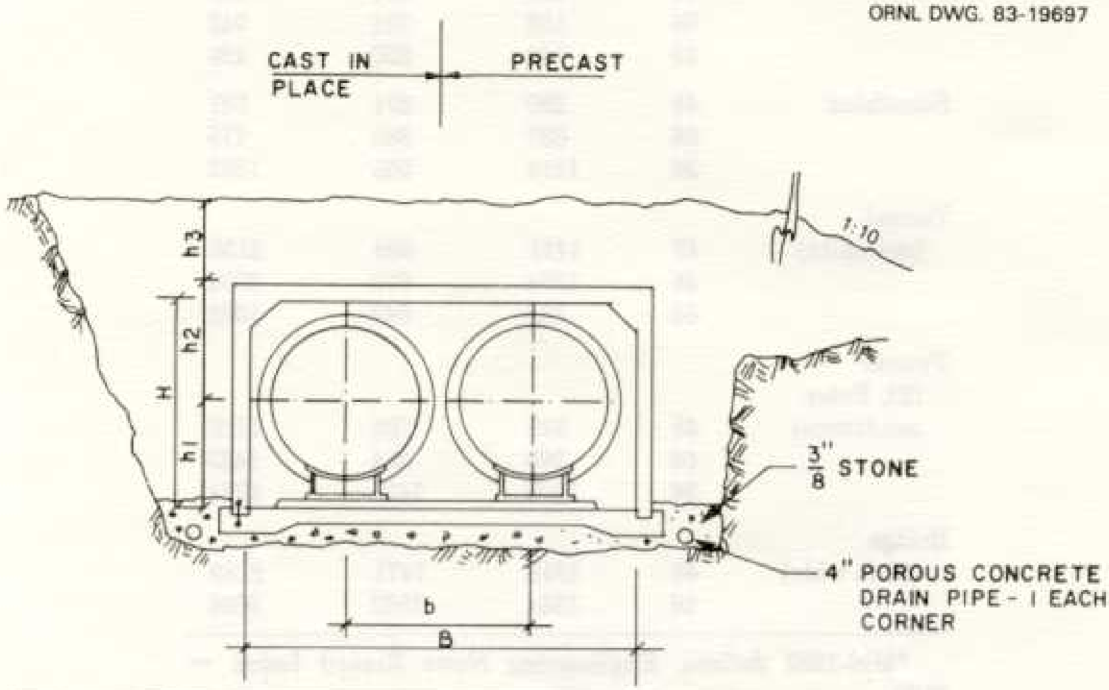



| PIPE DIAMETER | H | $h_1$ | $h_2$ | $h_3$ | B | b | SUPPORT SPAN |
|---|---|---|---|---|---|---|---|
| IN. | FT. | FT. | FT. | FT. | FT. | FT. | FT. |
| 66 | 8.0 | 4.0 | 4.0 | 2-3.5 | 14.0 | 7.0 | 75 |
| 48 | 5.5 | 2.75 | 2.75 | 2-3.5 | 11.0 | 5.5 | 55 |
| 36 | 4.5 | 2.25 | 2.25 | 2-3.5 | 8.5 | 4.5 | 35 |

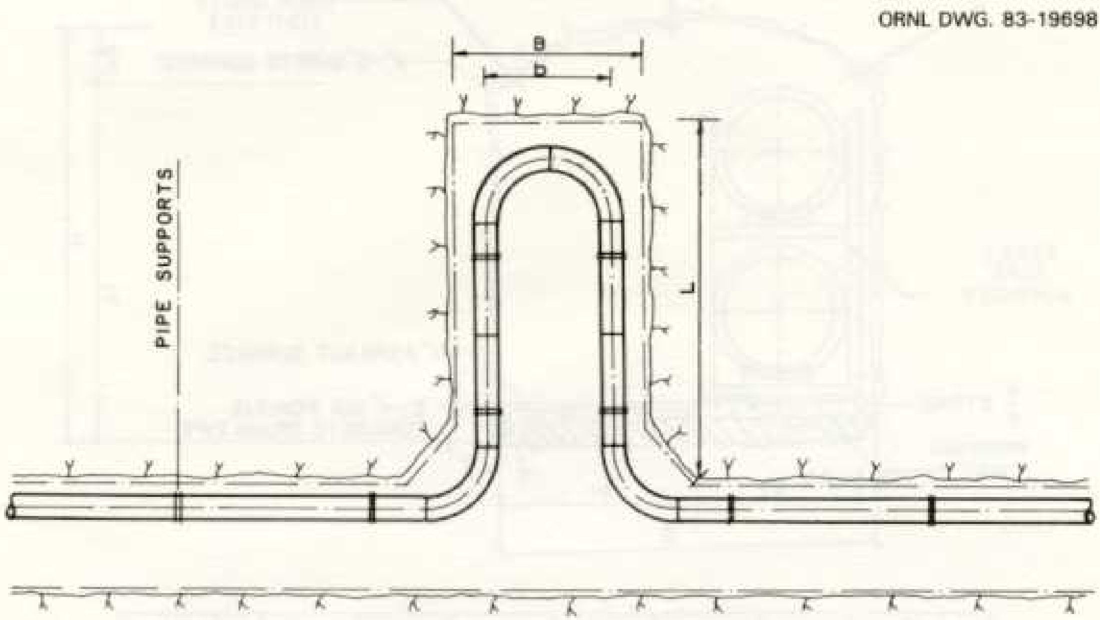
Fig. 4.3. Prairie Island hot water feasibility study—concrete pipe duct.

The precast section offers the additional advantage of allowing the contractor to stockpile material in areas of difficult access using all terrain vehicles, whereas the cast-in-place section would require an access road for concrete trucks.

Complete pipe systems with media, pipe, concrete culvert, and appurtenant structures can be installed at a rate of about 3 miles/year per workplace with a crew of about 40 men. At this rate, the concrete culvert pipelines for the project would take about 3 years, assuming that the work were to be carried out simultaneously along five stretches with a total of 200 men laying pipe.

### 4.4.3 Tunnel

A tunnel has two advantages over the cut-and-cover method: minimal disturbance to the residential and commercial area and a straight-line distance between two points. In rock tunneling, the rock with gunite lining will suffice as the pipe duct. Also, because the climatic conditions in the tunnel are uniform throughout the year, the spacing of expansion bends can be increased as shown in Fig. 4.4. The higher cost per foot and the financial risk associated with tunneling require a closer look at the advantages before this alternative is selected.




| PIPE DIAMETER IN. | MAX DIST. BETWEEN CHAMBS FT | B ft. | b ft. | L ft. | VOL. OF CHAMBER C.Y. |
|---|---|---|---|---|---|
| 66 | 2450 | 21.5 | 16.0 | 75 | 1100 |
| 48 | 2450 | 15.0 | 11.0 | 65 | 500 |
| 36 | 2100 | 12.0 | 8.0 | 55 | 300 |

**Fig. 4.4. Prairie Island hot water feasibility study—expansion chamber in tunnels.**

Tunneling conditions in the St. Peter sandstone are unique because of the favorable engineering properties of this rock. The rock is friable and easily excavated using a hydraulic lance. The unsupported rock stabilizes in a cathedral shape. The standup time is sufficient to place a permanent lining when required for the particular use of the tunnel. Also, the upper zone of the rock has been dewatered and has a limestone and shale cap which acts as a seal from the saturated overburden. These characteristics result in a less expensive per-foot tunneling cost than the average tunnel, while all the advantages of a tunnel are maintained.

Tunnel piping systems require a technique and labor planning entirely different from culvert systems, partly because work in the rock must be completed over a considerable distance before any pipe-laying or insulation work can be started. A suitable distance for each section is about 1.5 miles.

Tunnel sections in dolomite and in St. Peter sandstone, the two geological formations encountered in the area examined, are shown in Figs. 4.5 and 4.6.

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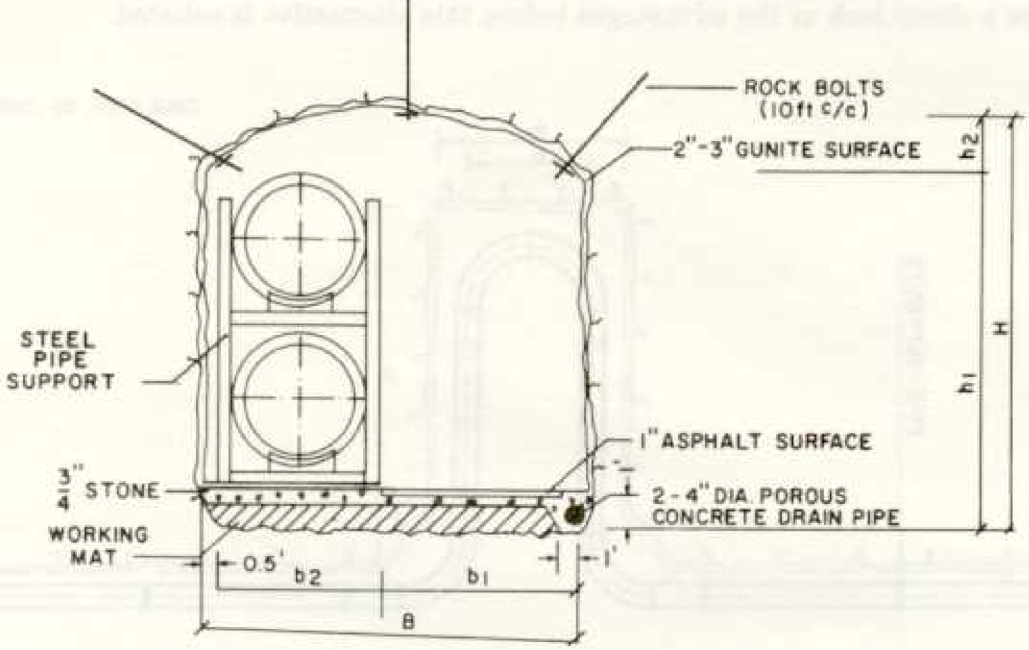
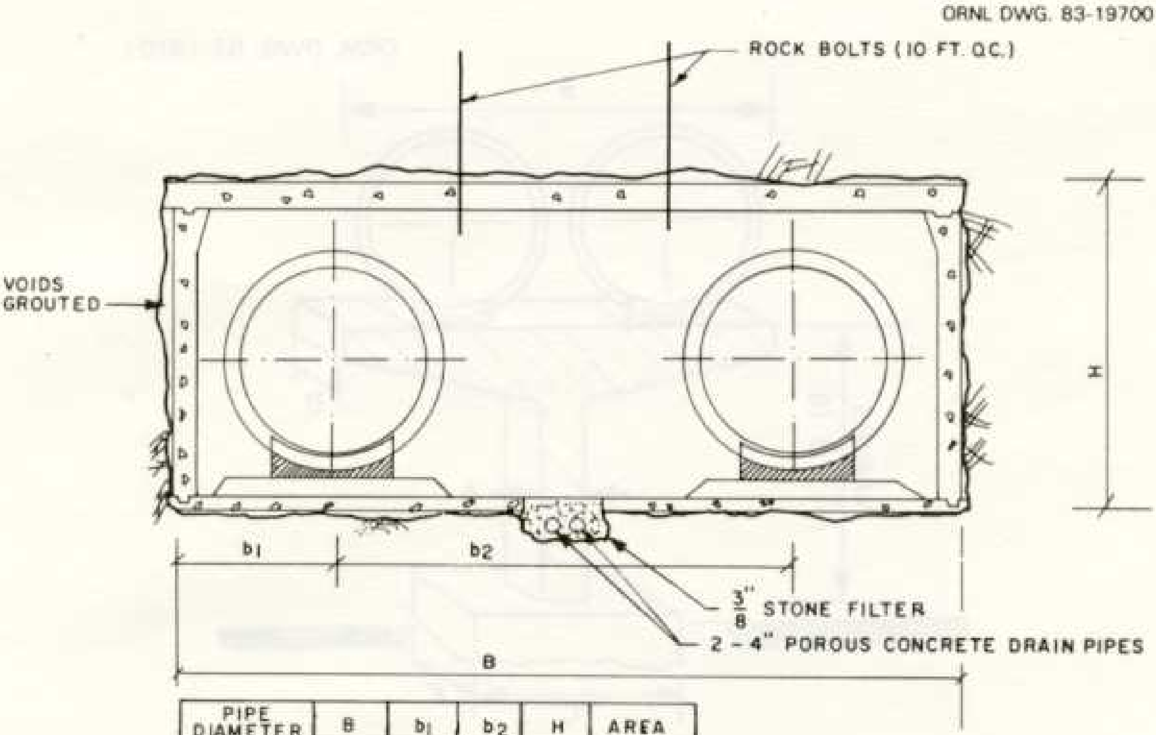



| PIPE DIAMETER | B | b1 | b2 | H | h1 | h2 | EXCAVATED AREA |
|---|---|---|---|---|---|---|---|
| IN. | FT. | FT. | FT. | FT. | FT. | FT. | SQ.FT. |
| 66 | 15.5 | 7.0 | 7.5 | 17.5 | 15.0 | 2.5 | 218 |
| 48 | 14.0 | 7.0 | 6.0 | 14.5 | 12.5 | 2.0 | 202 |
| 36 | 13.0 | 7.0 | 5.0 | 11.5 | 10.0 | 1.5 | 151 |

**Fig. 4.5. Prairie Island hot water feasibility study— tunnel section in dolomite.**




| PIPE DIAMETER | B | $b_1$ | $b_2$ | H | AREA |
|---|---|---|---|---|---|
| IN. | FT. | FT. | FT. | FT. | SQ. FT. |
| 66 | 16.5 | 4.0 | 12.5 | 8.0 | 132 |
| 48 | 14.25 | 3.25 | 11.0 | 8.0 | 114 |
| 36 | 12.75 | 2.75 | 10.0 | 7.0 | 90 |

**Fig. 4.6. Prairie Island hot water feasibility study—tunnel section in St. Peter sandstone.**

### 4.4.4 Stanchions

Where the ground is saturated or subject to frequent flooding, the pipe must be supported above the ground on stanchions, as shown in Fig. 4.7. The major disadvantage to this construction is that the pipeline is visible and exposed. The allowable pipe span is in direct relation to the pipe size. Spans of 35, 55, and 75 ft were used for the 36-, 48-, and 66-in. diameter pipes, respectively.

Piping systems of this type can be laid at approximately the same rate as culvert piping, about 3 miles/year.

### 4.4.5 Bridge

The bridge is proposed to cross the major rivers as an alternative to a very deep tunnel.

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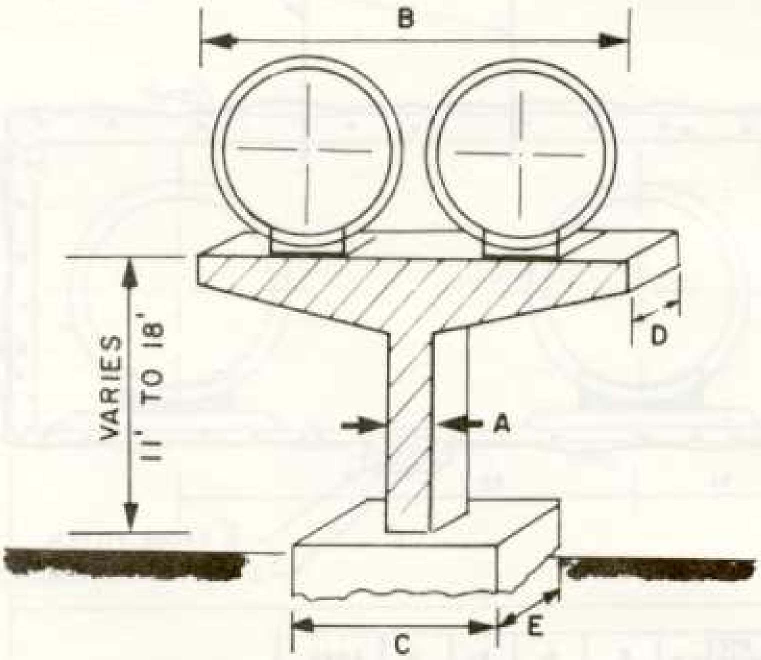


| PIPE DIAMETER | A | B | C | D | E |
|---|---|---|---|---|---|
| IN. | FT. | FT. | FT. | FT. | FT. |
| 66 | 13.0 | 13.0 | 14.0 | 2.0 | 8.0 |
| 48 | 6.0 | 11.0 | 13.0 | 2.0 | 8.0 |
| 36 | 4.0 | 9.0 | 10.0 | 1.5 | 7.5 |

**Fig. 4.7. Prairie Island hot water feasibility study—stanchion for pipe support aboveground.**

## 4.5 ROUTING OPTIONS

### 4.5.1 General

A feasibility study was undertaken for the purpose of selecting and evaluating the most economical route for the proposed hot water transmission line. Two general route options were selected for comparison. The routes were delineated on topographic maps and were further defined during a field reconnaissance conducted on the ground and by helicopter. The selection of the final route is dependent on several geotechnical factors, including:

1. topography,
2. geologic conditions,

3. groundwater conditions, and
4. construction access in remote areas as well as in highly developed urban areas.

These (geotechnical) considerations generally dictate the method of installation of the pipeline and will, therefore, affect the cost of the project.

The alternate routes selected for evaluation in the feasibility study are shown on Figs. 4.8 and 4.9. Each alternate route has been divided into sections for discussion

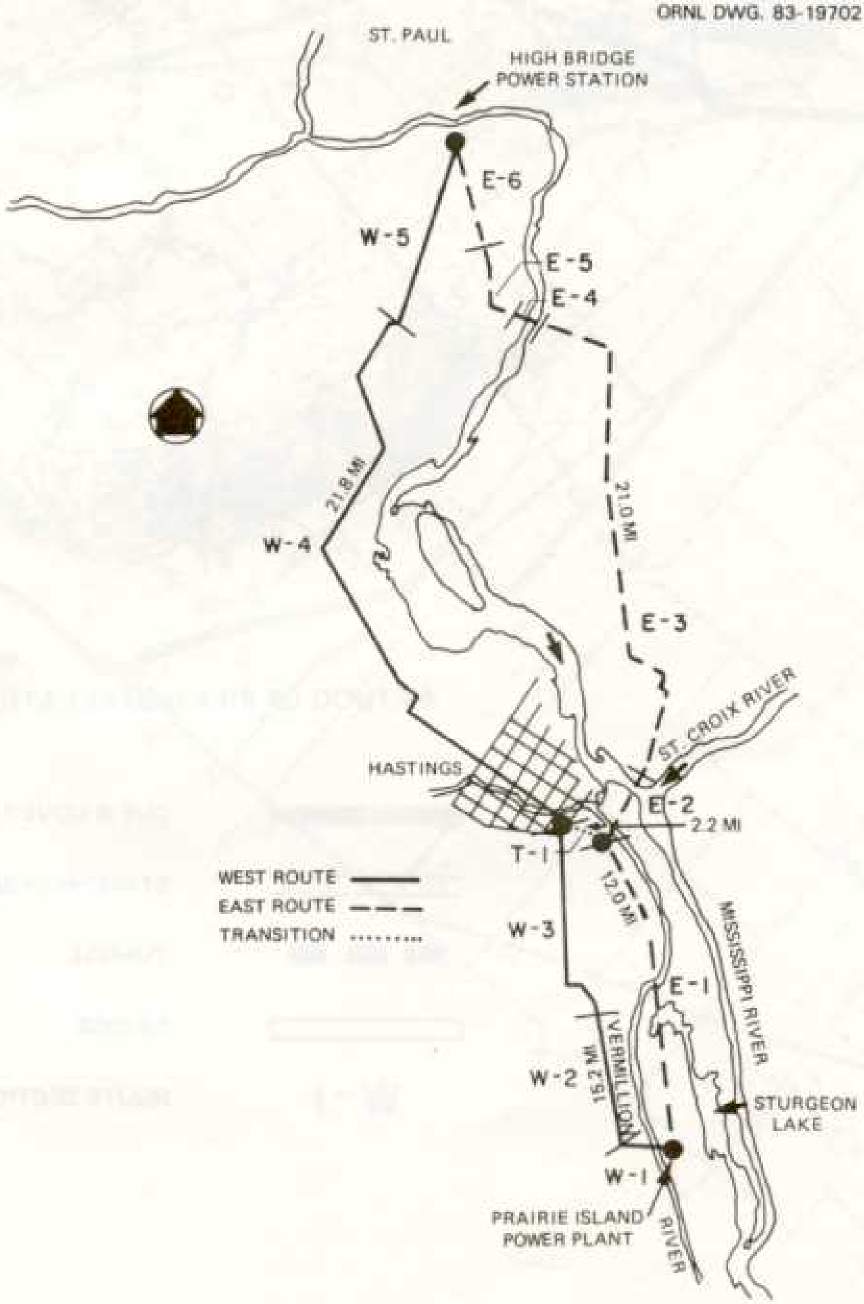


**Fig. 4.8. Prairie Island hot water feasibility study schematic plan—alternate routes.**

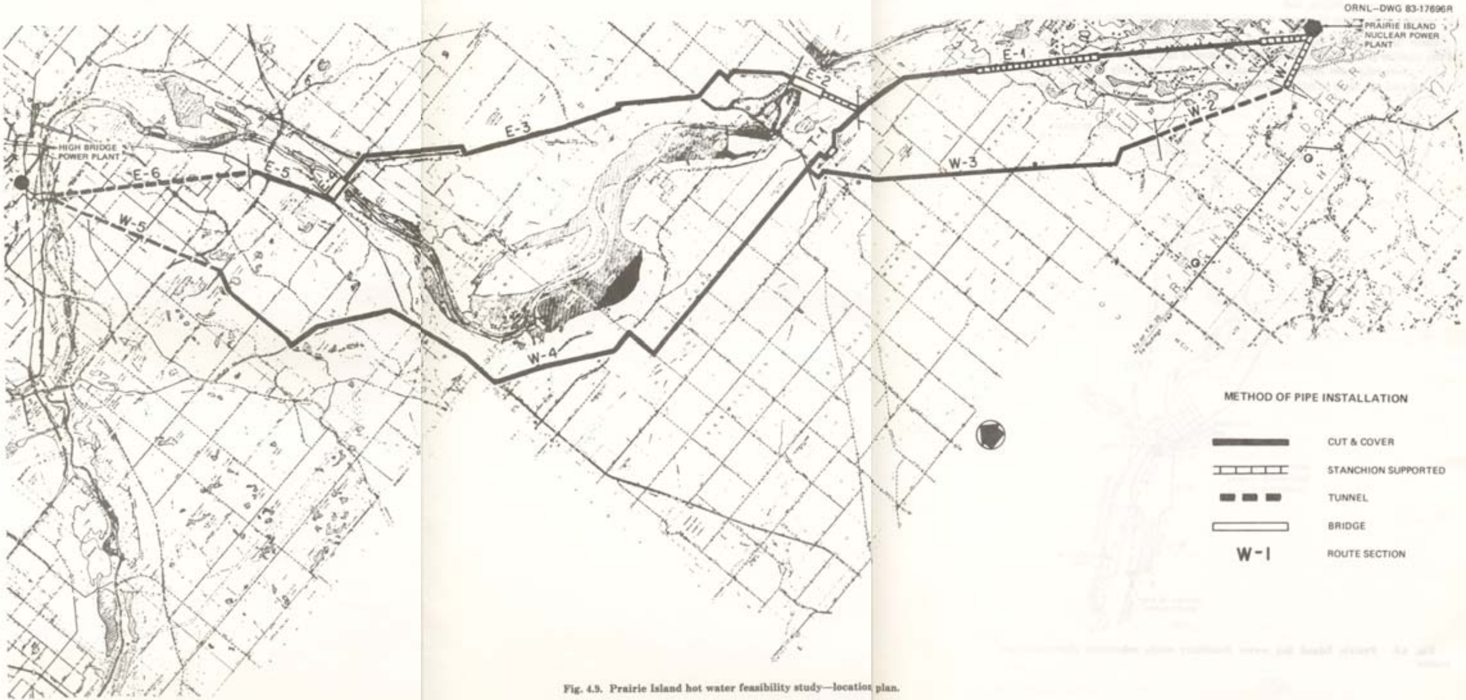


Fig. 4.9. Prairie Island hot water feasibility study—location plan.

purposes. For the most part, each section corresponds to a particular geologic or topographic condition and, therefore, a different method of installation.

The western route consists of Sects. W-1 through W-5. The route crosses the Mississippi River flood plain directly opposite the Prairie Island plant. From there the transmission line proceeds northwesterly to Hastings, west to Pine Bend, and finally north to the High Bridge. The route is approximately 37 miles long (see Fig. 4.8).

The east route, Sects. E-1 through E-6 on Fig. 4.9, also starts at the Prairie Island plant and proceeds northwesterly along the axis of Prairie Island, following the easement of the Chicago, Milwaukee, St. Paul, and Pacific Railroad. The pipeline route turns north, crossing the Vermillion and Mississippi rivers just east of Hastings. The route continues northwesterly through St. Paul Park and Cottage Grove and crosses the river a second time to Inver Grove Heights. The east route also ends at High Bridge and is a total of 33.1 miles long.

Section T-1, shown on Fig. 4.9 represents a transition between the east and west routes, through the city of Hastings. The transition was selected because it became evident during the course of the evaluation that a combination of the two routes might be the most economically feasible.

Information available for evaluating subsurface conditions along the proposed routes came from a variety of sources: published and unpublished maps and reports from the U.S. Geological Survey (USGS) and the Minnesota Geological Survey (MGS); well log data from both agencies; and soil boring logs and profiles from the Minnesota Department of Transportation (DOT) and Northern States Power (NSP). In addition, interviews were conducted with representatives of the agencies listed above and several other town and city agencies along the proposed route. (A complete listing of the sources is included in Appendix C.) Since no soil borings or other subsurface investigations were scheduled for the feasibility study, the information reviewed must be considered to be general in nature. The available data were particularly limited in the more remote areas of the alternate routes. However, even this information can be considered adequate for purposes for evaluating geotechnical conditions and estimating costs for comparison.

## 4.6 PUMPING STATIONS

### 4.6.1 Pressure and Power Requirements

To prevent intolerably high pressure in the system and to avoid steam formation, calculations have shown that five pumping stations will be required in the transmission pipeline.

The maximum allowable pressure, in the direct mode, was determined by the design parameters of the St. Paul distribution system into which the proposed transmission main would feed directly. This system will have pipe designed for 250 psig (16 bar). A minimum pressure of at least 35 psig (2 bar) must be maintained to prevent the water from vaporizing to steam.

In the indirect mode, because of the higher temperatures, the minimum allowable pressure must be correspondingly higher. To have a reasonable band between maximum and minimum pressures, the pipeline will be designed using a higher pressure pipe class for a normal working pressure of 400 psig (25 bar).

Figure 4.10 shows a typical system schematic diagram and pressure profile of a 600-MW(t) pipeline at design flow rate. The principal design criteria for the pumping stations are given in Tables 4.8 through 4.15.

To utilize components with which operational experience already has been gained, each pumping station is provided with three pumps in the delivery pipeline and three pumps in the return pipeline. The single exception to this general pump arrangement is Pumping Station No. 2, at 600 MW(t), where pumps are provided only on the supply main. Other advantages gained from distributing the flow over several pumps are increased accessibility and reduced impact of pump stoppage from water hammers.

### 4.6.2 Station Layout and Operational Considerations

A station layout (Fig. 4.11) was made for the purpose of developing a cost estimate.

The operating personnel at the Prairie Island plant must be able to rapidly stop flow to the heat exchangers under emergency conditions. This would be made possible by the installation of quick-closing isolation valves on both supply and return pipelines. These valves will be located in Pumping Station No. 1, just outside the fence at the Prairie Island plant. To protect the district heating pipelines from surges (water hammer) resulting from closure of the isolation valves, a shunt (bypass) pipe and valve will be installed between the supply and return pipes to automatically open upon closure of the isolation valves. Figure 4.11 schematically shows this piping arrangement.

All the pumps except those on the supply line at Prairie Island would be speed controlled. The latter pumps would operate at constant speed to reach the static pressure level. One of these pumps must operate at all times to maintain a static pressure in the line, even as flow approaches zero.

## 4.7 HEAT EXCHANGE STATIONS

The installation of heat exchangers in the indirect system has been proposed for the High Bridge station. Six heat exchangers of 200 MW each, for a capacity level of 1200 MW, and six heat exchangers of 100 MW each, for a capacity level of 600 MW, have been proposed. Layout proposals for the heat exchange stations have been made as a basis for cost estimates (see Fig. 4.12).

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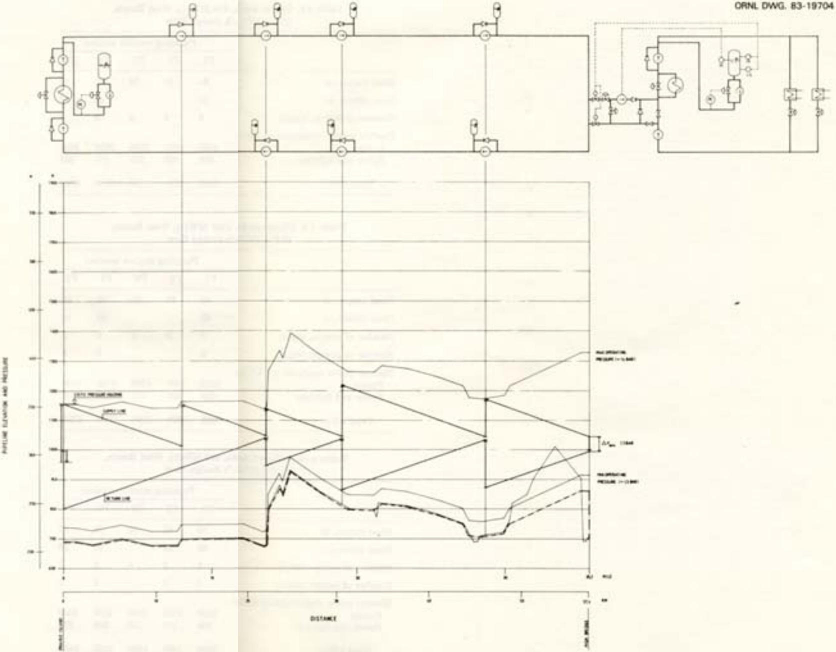



**Fig. 4.10. Prairie Island hot water feasibility study system diagram—East Route, direct mode 600 MW.**

**Table 4.8. Direct mode, 600 MW(t), West Route, 22.6 × $10^6$ lb/h design flow**

| | Pumping station number | | | | |
|---|---|---|---|---|---|
| | P1 | P2 | P3 | P4 | P5 |
| Head supply, m | 64 | 40 | 73 | 64 | 43 |
| Heat return, m | 64 | | | 42 | 43 |
| Number of pumps, supply | 3 | 3 | 3 | 3 | 3 |
| Electric power requirements, kW(e) | | | | | |
| Pumps | 4360 | 1370 | 2500 | 3600 | 2900 |
| Power and lighting | 300 | 200 | 200 | 300 | 300 |
| Total kW(e) | 4660 | 1570 | 2700 | 3900 | 3300 |

**Table 4.9. Direct mode, 1200 MW(t), West Route, 45.2 × $10^6$ lb/h design flow**

| | Pumping station number | | | | |
|---|---|---|---|---|---|
| | P1 | P2 | P3 | P4 | P5 |
| Head supply, m | 60 | 38 | 69 | 60 | 40 |
| Head return, m | 60 | | | 38 | 40 |
| Number of pumps, supply | 3 | 3 | 3 | 3 | 3 |
| Number of pumps, return | 3 | | | 3 | 3 |
| Electric power requirements kW(e) | | | | | |
| Pumps | 8200 | 2600 | 4700 | 6700 | 5400 |
| Power and lighting | 350 | 200 | 300 | 350 | 300 |
| Total kW(e) | 8550 | 2800 | 5000 | 7050 | 5700 |

**Table 4.10. Indirect mode, 600 MW(t), West Route, 11.8 × $10^6$ lb/h design flow**

| | Pumping station number | | | | |
|---|---|---|---|---|---|
| | P1 | P2 | P3 | P4 | P5 |
| Head supply, m | 65 | 62 | 90 | 63 | 63 |
| Head return, m | 65 | 36 | | 63 | 63 |
| Number of pumps, supply | 3 | 3 | 3 | 3 | 3 |
| Number of pumps, return | 3 | 3 | | 3 | 3 |
| Electric power requirements, kW(e) | | | | | |
| Pumps | 2300 | 1750 | 1800 | 2250 | 2250 |
| Power and lighting | 200 | 150 | 150 | 200 | 200 |
| Total kW(e) | 2500 | 1900 | 1950 | 2450 | 2450 |

**Table 4.11. Indirect mode 1200 MW(t), West Route, 23.6 lb/h design flow**

| | Pumping station number | | | | |
|---|---|---|---|---|---|
| | P1 | P2 | P3 | P4 | P5 |
| Head supply, m | 61 | 58 | 85 | 59 | 59 |
| Head return, m | 61 | 34 | | 59 | 58 |
| Number of pumps, supply | 3 | 3 | 3 | 3 | 3 |
| Number of pumps, return | 3 | 3 | | 3 | 3 |
| Electric power requirements, kW(e) | | | | | |
| Pumps | 7550 | 3300 | 3050 | 4200 | 4200 |
| Power and lighting | 300 | 250 | 250 | 250 | 250 |
| Total kW(e) | 7850 | 3550 | 3300 | 4450 | 4450 |

**Table 4.12. Direct mode 600 MW(t), East Route, 22.6 × $10^6$ lb/h design flow**

| | Pumping station number | | | | |
|---|---|---|---|---|---|
| | P1 | P2 | P3 | P4 | P5 |
| Head supply, m | 60 | 42 | 27 | 51 | 39 |
| Head return, m | 60 | | 27 | 51 | 48 |
| Number of pumps, supply | 3 | 3 | 3 | 3 | 3 |
| Number of pumps, return | 3 | | 3 | 3 | 3 |
| Electric power requirements, kW(e) | | | | | |
| Pumps | 4100 | 1450 | 1850 | 3500 | 3000 |
| Power and lighting | 300 | 150 | 150 | 300 | 300 |
| Total kW(e) | 4400 | 1600 | 2000 | 3800 | 3300 |

**Table 4.13. Direct mode 1200 MW(t), East Route, 45.2 × $10^6$ lb/h design flow**

| | Pumping station number | | | | |
|---|---|---|---|---|---|
| | P1 | P2 | P3 | P4 | P5 |
| Head supply, m | 55 | 39 | 25 | 47 | 36 |
| Head return, m | 55 | | 25 | 47 | 44 |
| Number of pumps, supply | 3 | 3 | 3 | 3 | 3 |
| Number of pumps, return | 3 | | 3 | 3 | 3 |
| Electric power requirements, kW(e) | | | | | |
| Pumps | 7500 | 2650 | 3400 | 6400 | 5450 |
| Power and lighting | 400 | 250 | 250 | 400 | 300 |
| Total kW(e) | 7900 | 2900 | 3650 | 6800 | 5750 |

**Table 4.14. Indirect mode, 600 MW(t), East Route, $11.8 \times 10^6$ lb/h design flow**

| | Pumping station number | | | | |
|---|---|---|---|---|---|
| | P1 | P2 | P3 | P4 | P5 |
| Head supply, m | 78 | 60 | 42 | 65 | 57 |
| Head return, m | 78 | | 42 | 65 | 57 |
| Number of pumps, supply | 3 | 3 | 3 | 3 | 3 |
| Number of pumps, return | 3 | | 3 | 3 | 3 |
| Electric power requirements, kW(e) | | | | | |
| Pumps | 2800 | 1050 | 1500 | 2300 | 1150 |
| Power and lighting | 200 | 100 | 150 | 200 | 150 |
| Total kW(e) | 3000 | 1150 | 1650 | 2500 | 1300 |

**Table 4.15. Indirect mode, 1200 MW(t), East Route, $23.6 \times 10^6$ lb/h design flow**

| | Pumping station number | | | | |
|---|---|---|---|---|---|
| | P1 | P2 | P3 | P4 | P5 |
| Head supply, m | 70 | 54 | 38 | 59 | 51 |
| Head return, m | 70 | | 38 | 59 | 51 |
| Number of pumps, supply | 3 | 3 | 3 | 3 | 3 |
| Number of pumps, return | 3 | | 3 | 3 | 3 |
| Electric power requirements, kW(e) | | | | | |
| Pumps | 5000 | 1900 | 2700 | 4200 | 3650 |
| Power and lighting | 300 | 150 | 200 | 300 | 250 |
| Total kW(e) | 5300 | 2050 | 2900 | 4500 | 3900 |

## 4.8 COST ESTIMATE AND FINAL ROUTE SELECTION

A cost analysis for the different methods of construction along the two routes has been completed. The costs are for mid-1982 prices at an Engineering News Record (ENR) Index of 3100. The cost breakdown for the different construction methods are shown in Table 4.16. These costs are exclusive of land and right-of-way acquisition.

A cost breakdown for the transportation of hot water from Prairie Island to High Bridge indicates that the least expensive alternative, although not the shortest (see Fig. 4.8), would be a combination of the two routes; that is E-1, T-1, W-4, and W-5.

ORNL DWG. 83-19705

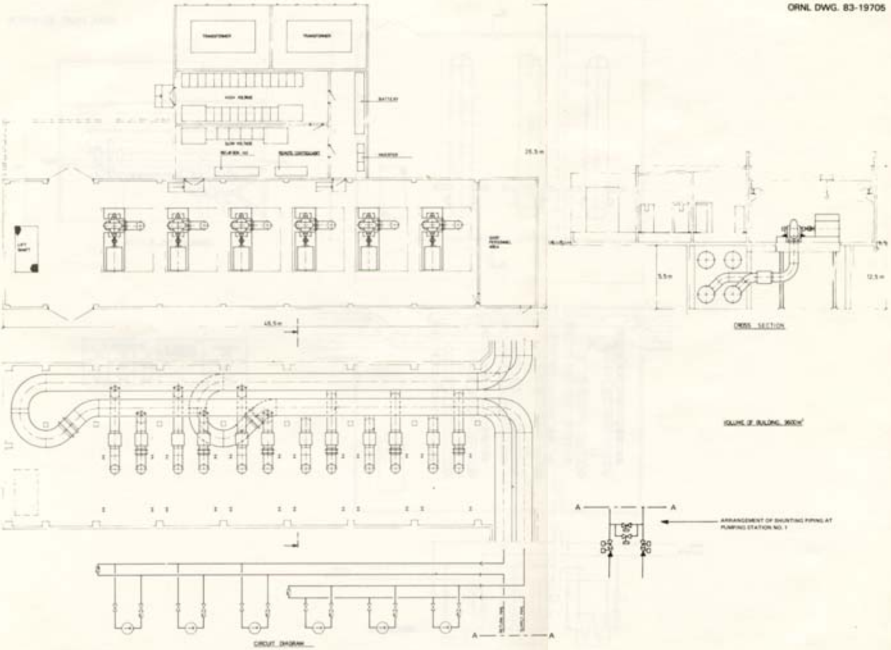
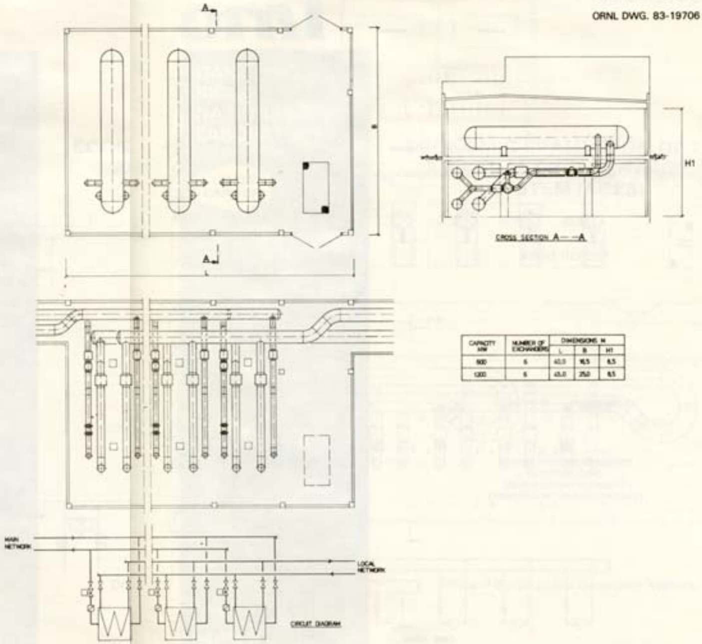



Fig. 4.11. Prairie Island hot water feasibility study—pumping station.

ORNL DWG. 83-19706




| CAPACITY MW | NUMBER OF EXCHANGERS | DIMENSIONS, M | | |
|---|---|---|---|---|
| | | L | B | H1 |
| 600 | 6 | 40.0 | 16.5 | 8.5 |
| 1200 | 6 | 45.0 | 25.0 | 8.5 |

**Fig. 4.12. Prairie Island hot water feasibility study—heat exchange station.**

**Table 4.16. Construction cost estimate[a]**

| System components | Direct mode | | Indirect mode | |
|---|---|---|---|---|
| | 600 MW | 1200 MW | 600 MW | 1200 MW |
| Pipelines | 156,000 | 193,000 | 136,000 | 160,000 |
| Valves and emptying stations | 7,000 | 8,000 | 5,000 | 7,000 |
| Pumping stations | 11,000 | 15,000 | 10,000 | 14,000 |
| Heat exchanger stations (excluding local pumps) | | | 6,000 | 11,000 |
| Storage tanks | 3,000 | 3,000 | 3,000 | 3,000 |
| Pressure-raising equipment | 2,000 | 2,000 | 2,000 | 2,000 |
| Water treatment plant | 2,000 | 2,000 | 2,000 | 2,000 |
| Instrumentation and control | 3,000 | 3,000 | 3,000 | 3,000 |
| Subtotal | 185,000 | 227,000 | 168,000 | 203,000 |
| Engineering, administrative, and contingencies | 28,000 | 34,000 | 25,000 | 30,000 |
| Total | 213,000 | 261,000 | 193,000 | 233,000 |

[a]Thousands of 1982 dollars; Engineering News Record Index = 3100.

This combination route also presents the least degree of risk from geotechnical considerations. Tunneling is a higher financial risk than cut-and-cover work, as reflected in the unit price of the work. The additional tunnel in Sect. W-2 is eliminated in this route. Also, the two Mississippi River crossings are eliminated and, therefore, the danger of damage to bridges due to natural elements and barge traffic on the river.

## REFERENCES FOR SECTION 4


1. M. H. Barnes et al., *St. Paul District Heating System Conceptual Design Study and Report*, ORNL/TM-6830/P10, Vol. II (January 1982).
2. *Hot Water Pipeline from Forsmark*, ORNL-tr-4801, Greater Stockholm Energy Co., translated from Swedish, report dated October 1981.

# 5. ECONOMIC ASSESSMENT

One of the primary purposes of this study is to develop estimates of the unit cost of thermal energy from a retrofitted nuclear power plant delivered to a load center by a hot water transmission system. This section describes the approach and methods used and discusses several of the most important factors for financing the capital investments required. These total project costs for retrofitting either one or two of the Prairie Island plant turbines to cogeneration and constructing a 600- or 1200-MW(t)-capacity transmission system have been presented in Sects. 3 and 4. The approach and methodology used for estimating the unit cost of thermal energy purchased from the electric utility-owned cogeneration plant, developed by NSP for the Prairie Island plant (see Appendix D), are summarized in Sect. 6.2.1.

## 5.1 BASIC APPROACH

As was mentioned in Sect. 1.4.4, the economic results from this study are presented in terms of constant mid-1982 dollars. The mid-1982 time was chosen because the cost estimates were developed at that time, so they represented current economic and monetary conditions. However, the most important reason for using mid-1982 as the "reference" year was to give the economic results in easily understood terms rather than in terms of highly inflated, and hence speculative, dollars of 10 to 15 years into the future. The reference year results can be inflated and escalated to any future period at whatever rate desired by the user to compare with other unit energy cost estimates.

It should also be noted that the unit cost of thermal energy delivered to the load center is based on estimates of costs to a utility company operating the transmission system. The unit cost results do not include any profit for the investment and operation of the transmission system. This approach is consistent with previous engineering cost estimates performed for the Twin Cities application studies.[1,2]

## 5.2 METHODS AND ASSUMPTIONS

The economic analysis followed the basic procedure outlined below; "reference" year refers to mid-1982.

1. Establish the total project cost (TPC) for the transmission system and plant modifications in reference year dollars. TPC includes all direct and indirect construction costs and contingencies.

2. Calculate AFDC funds required in reference year dollars assuming a 6% inflation rate, 8% escalation rate for construction costs,[3] a mid-1992 operation date, a four-year pipeline construction period, and a two-year plant modification construction period.
3. Levelized annual investment costs are based on an annual fixed charge rate (FCR) applied to the total investment cost, TPC + AFDC. A median FCR value of 18%/year is used with a range of 15 to 21% per year included as a parametric analysis to cover reasonable upper and lower levels.
4. Annual O&M costs for the transmission system, excluding electricity costs, are based on 1% per year of the TPC from Swedish district heating utility experience.[4]
5. Electricity costs for pumping and auxiliaries are based on 1982 NSP industrial power rates—including demand and energy charges. The annual pumping energy and auxiliary energy requirements were developed in the transmission system design study (see Sect. 7).
6. Annual thermal energy extracted from the power plant cycle and delivered to the load center is based on a load-duration curve developed for a 2600-MW(t) peak demand in the Twin Cities (Scenario "A" of ref. 4) and also an overall 80% annual availability factor for the cogeneration plant plus transmission system. This 80% availability factor was assumed to be determined by the availability of the nuclear power plant—that is, no unscheduled outage time added for the transmission system operation.
7. Purchased unit cost of thermal energy ($\$/10^6$ Btu) from the cogeneration plant owner for "firm" and "oil-interruptible" service is based on NSP's analyses of costs in the 1992–1997 time frame. Costs are adapted to the cogeneration derate appropriate for (1) a direct-cycle transmission system, with and without back-pressure turbines in the plant retrofit, and (2) an indirect-cycle transmission system without back-pressure turbines.
8. The total unit cost of thermal energy in reference year (1982) dollars is finally the summation of the unit investment costs, unit transmission O&M costs, unit electric energy cost, and the unit purchase price of thermal energy.

The most important parameter in this economic analysis is the fixed charge rate (FCR), since it includes the capital recovery factor for the large capital investment required. The next section discusses the financing options included in the range of FCR values used in this study.

## 5.3 CAPITAL INVESTMENT FINANCING OPTIONS

The fixed charge rate covers a large number of investment cost-related factors including the capital recovery factor, income and property taxes, interim replacement costs, and investment tax credits. The most important of these is the capital recovery factor, which is controlled by the effective interest rate (or "cost of money") and the project useful life over which the investment cost is amortized.

The values of FCR used in this study were based on a recent ORNL study of the data base for nuclear and fossil power generation cost analysis.[3] Recommended values for utility financing data were based on historical data plus long-run projections. Thus, recent perturbations of high inflation and bond interest rates experienced in the 1981-82 period were discounted. The recommended values in ref. 3 for several of the basic financial data are shown in Table 5.1, along with the resulting values for FCR of nuclear and fossil power plants.

**Table 5.1. Financial parameters**

| | |
|---|---|
| Plant economic life, years | 30 |
| Reference year | 1982 |
| Inflation rate, %/year | 6 (3-10)[a] |
| Escalation rate in excess of inflation rate for power plant construction, %/year | 2 (0-6)[a] |
| Capitalization, % | |
| Debt | 51 (55, 45)[a] |
| Preferred stock | 12 (15, 10)[a] |
| Equity | 37 (30, 45)[a] |
| Return on capitalization, %/year | |
| Debt interest | 10 (6-16)[a] |
| Preferred dividend | 10 (6-16)[a] |
| Equity return | 15 (10-18)[a] |
| Average cost of money, %/year | 11.9 (7-17)[a] |
| Federal income tax rate, %/year | 46 |
| State income tax rate, %/year | 4 |
| Tax adjusted cost of money, %/year | 9.4 (6-13)[a] |
| Local property tax rate,[b] %/year | 2 |
| Tax depreciation method | ACRS[c] |
| Tax depreciation life, years | |
| Nuclear | 10 |
| Fossil | 15 |
| Investment tax credit rate, % | 10 |
| Interim replacement/backfitting rate,[d] %/year | 1 |
| Decommissioning cost, millions of 1982 dollars | |
| Fossil | 0 |
| Nuclear | 120 (60-200)[a] |
| Interest rate of decommissioning fund, %/year | 8.5 (5-13)[a] |
| Fixed charge rates, %/year | |
| Fossil | 17.7 (12-25)[a] |
| Nuclear | 17.9 (12-25)[a] |

[a]Range of variation or uncertainty in parentheses.

[b]Based on initial investment with no escalation due to inflation or decrease due to depreciation.

[c]Accelerated capital recovery.

[d]Percentage of initial investment in constant dollars, escalating at general rate of inflation.

Source: H. I. Bowers et al., *Reference Data Base for Nuclear/Fossil Power Generation Cost Analysis*, ORNL/TM-8332, Oak Ridge National Laboratory, Oak Ridge, Tenn., June 1982.

Based on the FCR analysis in ref. 3, a median value for FCR = 18%/year was chosen as consistent with 12%/year average cost of money and all other parameters as shown in Table 5.1. To estimate upper and lower bounds on FCR, two sets of values were inserted for the project useful life, N, and the average cost of money, X. For the high value of FCR, N = 15 years and X = 12%/year yielded an FCR = 20.9%/year. For a low value of FCR, N = 40 years and X = 9%/year yielded an FCR = 14.9%/year. Therefore, a nominal range of FCR values from 15 to 21% per year were adopted for this study to represent annualization of the power plant and transmission system investment.

From the preceding brief discussion, it is evident that the project useful life, N, must be kept to at least 15 years or longer to prevent excessive FCR values to apply. If, for example, a pessimistic combination of N = 15 years and X = 15%/year were assumed, the FCR would increase to 22.7%/year. Thus, the remaining useful life of a nuclear power plant becomes an important consideration for a project involving a large capital investment for the transmission system.

The useful life is particularly important for an existing nuclear power plant because of the potential for a limit on pressure vessel life from accumulated neutron embrittlement. Strategies to cope with this potential problem have not been fully identified by nuclear utilities, so no definitive answer is available for the "useful life" question. However, it is possible that a cost-effective remedy for pressure vessel embrittlement will be developed to extend the life of nuclear steam systems because of the large investment by utilities in the balance of the plant.

## REFERENCES FOR SECTION 5

1. G. A. Englesson et al., *Economic Comparison of New Coal-Fueled, Cogeneration Power Plants for District Heating and Electric-Only and Heat-Only Power Plants*, ORNL/TM-6830/P8, Oak Ridge National Laboratory, Oak Ridge, Tenn., May 1982.

2. G. A. Englesson et al., *Modification of the Existing Units at the High Bridge Power Plant to Cogeneration for Hot Water District Heating*, ORNL/TM-6830/P9, Oak Ridge National Laboratory, Oak Ridge, Tenn., October 1980.

3. H. I. Bowers et al., *Reference Data Base for Nuclear/Fossil Power Generation Cost Analysis*, ORNL/TM-8332, Oak Ridge National Laboratory, Oak Ridge, Tenn., June 1982.

4. Peter Morgan et al., *Overall Feasibility and Economic Viability for a District Heating/New Cogeneration System in Minneapolis-St. Paul*, ORNL/TM-6830/P3, Oak Ridge National Laboratory, Oak Ridge, Tenn., October 1979.

# 6. COST OF COGENERATED HOT WATER AT THE PLANT GATE

## 6.1 INTRODUCTION

In this section, the unit cost of cogenerated hot water will be developed for a hypothetical utility company purchasing thermal energy from the electric utility that owns the nuclear plant for transmission to the district heating load center. (The transmission system could also be owned by the electric utility with essentially no change in the economic results.) The "plant gate" cost includes all of the necessary equipment and facilities to heat the circulating hot water in steam-to-hot water heat exchangers.

The unit cost of hot water thermal energy is composed of the plant retrofit investment cost and the thermal energy production cost, which includes all plant O&M and fuel costs and also all cogeneration system-related costs. These costs are developed for the three power plant extraction cycles analyzed by Westinghouse in Sect. 3: Cycle I, including back-pressure turbines, and Cycles II-A&B, without back-pressure turbines. Plant retrofit investment costs were developed by Westinghouse on the basis of an individual turbine retrofit to supply up to 600 MW(t). A 1200-MW(t)-capacity transmission system, requiring retrofit of both nuclear turbines, was assumed to double the investment cost, since each turbine retrofit is assumed to have completely separate and redundant equipment and operation. In practice, there could be some economies in the procurement and installation of the second of the two identical units.

## 6.2 PRODUCTION COST OF COGENERATED THERMAL ENERGY

The general considerations in developing the production cost of thermal energy for an existing, base-loaded power plant—fossil- or nuclear-fueled—have already been addressed in Sect. 1.4.3. In this section, the cost allocation method used by NSP is summarized and the results of NSP's anaysis of production costs are presented. Appendix D is the NSP report on the "price" of thermal energy for Prairie Island.

### 6.2.1 Cost Allocation Method

The method used by NSP to develop a "price"—to another "transmission utility"—of cogenerated thermal energy is based on the following principles:

1. The "price" includes all incremental O&M costs and system-related costs—replacement energy, specifically—from cogeneration operation.

2. The "price" does not include any investment cost of the plant; therefore, the electric utility retains the right to full condensing, or no cogeneration operation, for periods of emergencies within the regional power pool.
3. The "price" allocates most of the cogeneration advantage of reduced fuel consumption to the thermal energy customer. However, the electric utility does benefit from a service charge which guarantees that the production cost of electricity is not subsidizing the production cost of thermal energy. This service charge is included in a 20% addition to the replacement energy cost to calculate the price, with the exception of item 4.
4. The "price" includes an allowance for decommissioning costs of the nuclear plant, on the basis of the proportional share of energy production for thermal energy.

In NSP's report in Appendix D, the concept of "price" vs "cost" of thermal energy is emphasized because thermal energy produced by a base-loaded plant is valued higher than the production cost. The value of thermal energy produced by a base-loaded cogeneration plant already in the electric utility rate base is determined predominantly by savings in new base-load capacity costs rather than fuel costs. A corollary to this statement is that the annual average thermal energy "price" for a normally base-loaded plant is determined by (1) the derate characteristic—MW(e) capacity reduction/MW(t) of thermal capacity—of the retrofitted turbine-generator averaged over a yearly cycle and (2) the annual average "replacement" energy cost for the entire electric generating system. Therefore, NSP's primary task was to analyze the average replacement energy cost—or the average production cost from all higher production cost units such as intermediate-load coal units, No. 6 oil-fueled gas turbine peaking units, and No. 2 oil-fueled diesel peaking units.

The average replacement energy cost depends upon several factors such as (1) the maximum electric capacity derate from cogeneration operation, (2) the characteristics of the electric generating system relative to the distribution of electricity production costs and unit capacities, and (3) the reserve margin of electrical system on a time-dependent basis. The reserve margin includes the effects of system demand, scheduled plant maintenance outages, and block power sales or purchases with electrical system inter-ties. It is obvious that replacement energy costs can vary widely between different utilities because of significant differences in the factors mentioned above. The results of NSP's analysis of replacement energy cost, which is the basis for the price of thermal energy from Prairie Island, is based on NSP's best estimate of the system's characteristics in the 1992-1997 time frame. Although the replacement energy cost will therefore vary between utilities—and, in fact, from year to year within the same utility—the NSP results will be representative of a utility with similar coal-fueled, intermediate-load units and oil-fueled gas turbine peak-load units and a similar reserve margin.

Since it was realized that the replacement energy cost would be the dominant factor in determining the price of thermal energy, it was decided that two supply scenarios should be analyzed. The first scenario is for "firm" service, whereby thermal energy production from the cogeneration plant would be base loaded and only be interrupted for electrical system emergency demand. The second scenario is for "oil-interruptible" service, whereby thermal energy production would be interrupted at any time that the electrical system would require turning on an oil-fueled peaking plant to meet the next increment of increasing electric load. Thus, the highest cost contribution to replacement energy would be eliminated, as illustrated in the electrical production cost variation shown in Fig. 6.1. However, the annual hours of thermal production are reduced for this scenario, thus reducing the utilization factor for the transmission system and increasing the unit transmission cost. Using both the firm and oil-interruptible supply scenarios allows the economic trade-off between the price of thermal energy from the plant and the transmission cost to be evaluated.

On the basis of the consideration and principles discussed above, NSP used its PECOS cost production model to calculate replacement energy costs for the years 1992 through 1997 for firm and oil-interruptible supply scenarios at 140- and 113-MW(e) derate conditions. (These derate values were early estimates of the derate from Westinghouse analyses of plant cogeneration performance.) The PECOS program analyzes production costs and replacement energy costs on a monthly time period basis and, therefore, does not model any short-term effects such as diurnal load variations. NSP also emphasizes that the results of the PECOS analysis and the prices of thermal energy based thereon are for estimation purposes only. In an

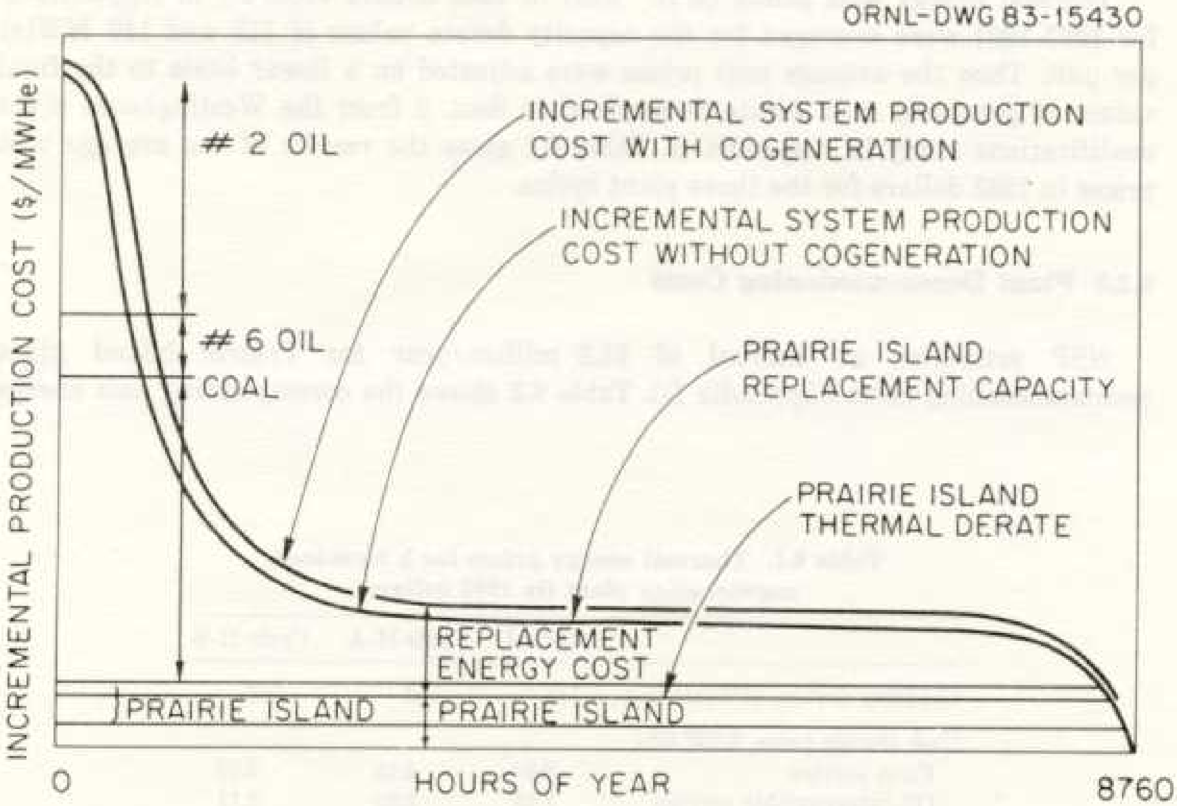



**Fig. 6.1. Sample incremental system production cost.**

actual negotiation between a "selling" utility and a "buying" utility, there is some risk to the buying utility because of uncertainties in future electric system loads and generating capacity that could affect the future price of thermal energy. However, considerations such as the value of reduced risk in future price variations, although valid, are beyond the scope of this study. The NSP analysis of replacement energy costs is taken to be a realistic basis for estimating the price of thermal energy from an existing base-loaded plant.

### 6.2.2 Price of Thermal Energy

Prices of thermal energy, not including plant decommissioning costs, were developed for the three levels of turbine-generator derate, identified by the three plant cycles in Sect. 3, and also for "firm" and "oil-interruptible" supply modes. The three plant cycles are as follows:

1. Cycle I—extraction from HP element exhaust to new back-pressure turbines to 250°F hot water heat exchangers for direct transmission.
2. Cycle II-A—extraction from HP element exhaust plus HP bleeds to 350°F hot water heat exchangers for indirect transmission.
3. Cycle II-B—extraction from HP element exhaust to 250°F hot water heat exchangers for direct transmission.

Thermal energy unit prices ($\$/10^6$ Btu) in 1982 dollars reported in Appendix D for 1992–1997 were averaged for the capacity derate values of 113 and 140 MW(e) per unit. Then the average unit prices were adjusted on a linear basis to the final values of generator unit derate determined in Sect. 3 from the Westinghouse plant modifications study in Appendix B. Table 6.1 gives the results of the average unit prices in 1982 dollars for the three plant cycles.

### 6.2.3 Plant Decommissioning Costs

NSP estimated an accrual of \$5.2 million/year for Prairie Island plant decommissioning costs (Appendix D). Table 6.2 shows the corresponding unit energy

**Table 6.1. Thermal energy prices for a base-load cogeneration plant (in 1982 dollars)**

| | Cycle I | Cycle II-A | Cycle II-B |
|---|---|---|---|
| Capacity derate, MW(e)/unit | 112 | 160 | 142 |
| Unit energy price, $\$/10^6$ Btu | | | |
| Firm service | 2.94 | 4.15 | 3.70 |
| Oil-interruptible service | 1.68 | 2.65 | 2.11 |

**Table 6.2. Plant decommissioning unit costs (in 1982 dollars/$10^6$ Btu)**

| | Cycle I | Cycle II-A | Cycle II-B |
|---|---|---|---|
| Firm service | 0.079 | 0.108 | 0.097 |
| Oil-interruptible service | 0.091 | 0.128 | 0.114 |

costs for the three plant cycles, based on partitioning the total cost between electrical and thermal production.

### 6.2.4 Total Unit Energy Prices

The total unit energy price, equivalent to the unit energy production cost, for nuclear steam extracted from the Prairie Island turbine cycle and delivered to district heating heat exchangers is obtained from the sum of the unit energy price and the unit decommissioning cost reported in the preceding sections. The resulting total unit energy purchase price to the transmission utility is shown in Table 6.3.

**Table 6.3. Total unit energy price (in 1982 dollars/$10^6$ Btu)**

| | Cycle I | Cycle II-A | Cycle II-B |
|---|---|---|---|
| Firm service | 3.021 | 4.258 | 3.797 |
| Oil-interruptible service | 1.771 | 2.778 | 2.225 |

## 6.3 COST OF PLANT RETROFIT TO COGENERATION

The total project costs—or TPC—have been developed by Westinghouse/Fluor (Appendix B) and reported in Sect. 3. These costs are the "overnight" construction costs in mid-1982 dollars for the three turbine retrofit cycles under consideration. For convenience, however, the TPCs for cycles II-A and B without back-pressure turbines are taken to be the same because of the small difference between them. In this section, results are given for the total investment cost, including allowance for funds during construction (AFDC); the annual investment cost based on an 18%/year fixed charge rate; and the unit investment cost ($/$10^6$ Btu) based on the annual thermal energy production from either a single-turbine retrofit or a two-turbine retrofit.

### 6.3.1 Total and Annual Investment Costs

The TPC from Sect. 3 is presented in Table 6.4 on a per-turbine unit basis. Retrofit of either turbine is taken to be identical in terms of equipment installed

**Table 6.4. Total and annual cost per turbine retrofit (in millions of 1982 dollars)**

| | Cycle I | Cycles II-A and B |
|---|---|---|
| TPC[a] | 31.47 | 13.46 |
| AFDC[b] | 0.686 | 0.293 |
| Total investment cost | 32.16 | 13.75 |
| Annual investment cost at 18% FCR[c] | 5.789 | 2.476 |

[a]Total project cost.
[b]Allowance for funds during construction.
[c]Fixed charge rate.

and hence construction cost, so retrofit of both turbines doubles the investment costs in Table 6.4.

Calculation of the AFDC cost component was based on the procedure in the 1981 EPRI *Technical Assessment Guide.*[1] Assuming a two-year construction period, inflation and construction cost escalation rates of 6 and 8% per year and a 1992 operation date, the AFDC cost is 2.18% of the TPC in 1982 dollars. The resulting AFDC cost, total investment cost, and annual investment cost for a 18%/year fixed charge rate are also given in Table 6.4.

### 6.3.2 Total Unit Investment Costs

The annual investment costs for an 18% fixed charge rate in Table 6.4 are converted to unit investment costs ($\$/10^6$ Btu) on the basis of the annual thermal energy production totals from Table 2.2. These unit costs are given in Table 6.5 for firm and oil-interruptible service.

**Table 6.5. Total unit investment costs for turbine retrofit (in 1982 dollars/$10^6$ Btu)**

| | Cycle I | Cycles II-A and B |
|---|---|---|
| One turbine | | |
| Firm service | 0.589 | 0.252 |
| Oil-interruptible service | 0.692 | 0.296 |
| Two turbines | | |
| Firm service | 0.784 | 0.336 |
| Oil-interruptible service | 0.923 | 0.395 |

## 6.4 TOTAL UNIT COST OF THERMAL ENERGY AT THE PLANT GATE

The total unit cost at the plant gate is the combined unit investment cost from Table 6.5 and the unit energy price from Table 6.3. The resulting total unit cost for an 18%/year fixed charge rate is shown in Table 6.6 for the three plant cycles, two modes of service, and one- or two-turbine retrofit cases.

**Table 6.6. Total unit cost of thermal energy at plant gate (in 1982 dollars/$10^6$ Btu)**

| | Cycle I | Cycle II-A | Cycle II-B |
|---|---|---|---|
| One turbine | | | |
| Firm service | 3.61 | 4.50 | 4.04 |
| Oil-interruptible service | 2.46 | 3.06 | 2.51 |
| Two turbines | | | |
| Firm service | 3.80 | 4.58 | 4.12 |
| Oil-interruptible service | 2.69 | 3.16 | 2.61 |

## 6.5 DISCUSSION OF RESULTS

At this point, two main observations can be made from the results in Table 6.6. First, the Cycle I unit costs are generally lower because of the lower energy price from the more efficient back-pressure turbine cycle. This situation is especially true for firm service, since the increased investment for the back-pressure turbine is used most extensively. The second observation is that Cycle II-B unit costs, the 250°F hot water direct extraction cycle, are very close to the Cycle I unit costs, especially for oil-interruptible service. Thus, the reduced investment cost almost compensates for the higher energy cost for Cycle II-B relative to Cycle I.

Unit costs of Cycle II-A, the 350°F hot water direct extraction cycle, are the highest costs of these three cycles, because it is the least efficient cycle with the highest energy prices. The final judgment on overall economic ranking among these cycles depends on the additional transmission costs reported in Sect. 7.

### REFERENCE FOR SECTION 6

1. Electric Power Research Institute, *Technical Assessment Guide*, 1981 ed., 1981.

# 7. COST OF HOT WATER TRANSMISSION TO THE LOAD CENTER

## 7.1 INTRODUCTION

In this section, the total annual cost and unit cost of the transmission system operation from the Prairie Island plant to the High Bridge plant are given. The basis for these costs is the most economical route selected in the Metcalf & Eddy *Transmission Line Route Options* study (see Appendix C) and described in Sect. 4. The selected route is 36.0 miles (57.9 km) in length and, although not the shortest, has the lowest construction costs of the routes considered.

The annual investment costs of the transmission system in this section are based on an 18%/year fixed charge rate (FCR) for the utility owning and operating the system. The effect of variations in the FCR on the unit cost of hot water transmission is considered in Sect. 8.

The transmission costs are reported in mid-1982 dollars for 600- and 1200-MW(t)-capacity systems that relate to a retrofit of either one or two turbines at Prairie Island and also "direct" transmission at a 250°F supply temperature (Cycles I and II-B) or "indirect" transmission at a 350°F supply temperature (Cycle II-A). These options are considered to evaluate the economic effect of transmission system capacity and transmission mode.

## 7.2 ANNUAL INVESTMENT COSTS

The total investment cost is composed of the total project cost (TPC) and the allowance for funds during construction (AFDC). Values for TPC from Sect. 4 in mid-1982 dollars are given in Table 7.1; the AFDC costs in Table 7.1 are 6.69% of the TPC. This result is based on a four-year construction period, inflation and construction cost escalation rates of 6 and 8%/year, respectively, and a 1992 operation date. The resulting annual investment cost for an 18%/year FCR is shown in Table 7.1.

## 7.3 ELECTRIC POWER COSTS

### 7.3.1 Electric Energy Required

The *Transmission Line Route Options* study (Appendix C) developed the electric energy and power required for the transmission system on the basis of both pumping and auxiliary electricity. These results are tabulated in Table 7.2. The

**Table 7.1. Transmission system total and annual investment cost (in millions of 1982 dollars)**

| | Cycles I and II-B | | Cycle II-A | |
|---|---|---|---|---|
| | 600 MW(t) | 1200 MW(t) | 600 MW(t) | 1200 MW(t) |
| TPC[a] | 212 | 260 | 192 | 232 |
| AFDC[b] | 14.2 | 17.4 | 12.8 | 15.5 |
| Total investment | 226.2 | 277.4 | 204.8 | 247.5 |
| Annual investment | 40.72 | 49.93 | 36.86 | 44.55 |

[a]Total project cost.
[b]Allowance for funds during construction.

**Table 7.2. Transmission system electrical requirements**

| | Cycles I and II-B | | Cycles II-A | |
|---|---|---|---|---|
| | 600 MW(t) | 1200 MW(t) | 600 MW(t) | 1200 MW(t) |
| Pump power, MW(e) | 14.8 | 27.6 | 12.6 | 22.3 |
| Auxiliary power, MW(e) | 1.3 | 1.5 | 0.9 | 1.3 |
| Equivalent full power, pump hours/year | 4,805 | 3,605 | 4,804 | 3,605 |
| Annual pump energy, MW(e) hours | 71,100 | 99,500 | 60,530 | 80,390 |
| Annual auxiliary energy, MW(e) hours | 11,390 | 13,140 | 7,880 | 11,390 |
| Total MW(e) hours | 82,490 | 112,640 | 68,410 | 91,780 |

annual pumping energy for "firm" and "oil-interruptible" service modes is assumed to be the same, even though the latter mode could reduce the pumping requirement by about 500  h/year.

### 7.3.2 Electricity Rates

The cost of electricity for the transmission system was based on the NSP industrial power rate of mid-1982. This rate has a demand charge and an energy charge as follows: An average demand charge is \$5.11/kW(e)·month up to 100 kW(e) and \$4.90/kW(e)·month above 100 kW(e); the energy charge is \$02.44/kW(e)·h. These rates are expected to escalate at the average inflation rate in the future.

### 7.3.3 Annual Electricity Costs

The annual electricity costs for auxiliary and pumping requirements were calculated for the power and energy levels in Table 7.2. The resulting annual costs are given in Table 7.3.

**Table 7.3. Transmission system annual electricity costs (in 1982 dollars)**

| | Cycles I and II-B | | Cycle II-A | |
|---|---|---|---|---|
| | 600 MW(t) | 1200 MW(t) | 600 MW(t) | 1200 MW(t) |
| **Auxiliary energy** | | | | |
| Demand charge | 76,692 | 88,452 | 53,172 | 76,692 |
| Energy charge | 277,916 | 320,616 | 192,272 | 277,916 |
| Total | 354,608 | 409,068 | 245,444 | 354,608 |
| **Pump energy** | | | | |
| Demand charge | 870,240 | 1,622,880 | 740,880 | 1,311,240 |
| Energy charge | 1,734,840 | 2,427,800 | 1,476,932 | 1,961,516 |
| Total | 2,605,080 | 4,050,680 | 2,217,812 | 3,272,756 |
| **Total cost** | | | | |
| Demand | 946,982 | 1,711,332 | 794,052 | 1,387,932 |
| Energy | 2,012,756 | 2,748,416 | 1,669,204 | 2,239,432 |
| Total | 2,959,738 | 4,459,748 | 2,463,256 | 3,627,364 |

## 7.4 ANNUAL MAINTENANCE COSTS

The annual maintenance cost for the transmission system was based on 1% per year of the total project cost (TPC) in Table 7.1. This approach is based on Swedish experience in operating hot water transmission lines.[1] The resulting annual maintenance costs are shown in Table 7.4.

**Table 7.4. Summary of annual costs of the transmission system (in millions of 1982 dollars)**

| | Cycles I and II-B | | Cycle II-A | |
|---|---|---|---|---|
| | 600 MW(t) | 1200 MW(t) | 600 MW(t) | 1200 MW(t) |
| Investment cost at 18%/year FCR[a] | 40.72 | 49.93 | 36.86 | 44.55 |
| Electricity cost | 2.96 | 4.46 | 2.46 | 3.63 |
| Maintenance cost | 2.12 | 2.60 | 1.92 | 2.32 |
| Total | 45.80 | 56.99 | 41.24 | 50.50 |

[a]Fixed charge rate.

## 7.5 TOTAL ANNUAL COST OF TRANSMISSION SYSTEM

The total annual cost of the transmission system is shown in Table 7.4, along with its basic cost components. As was expected, the investment cost represents the largest component, about 90% of the total cost in this analysis. These results are based on the median, 18%/year, value of the fixed charge rate.

## 7.6 UNIT COST OF HOT WATER TRANSMISSION

The unit cost of hot water transmission is based on the total annual cost in Table 7.4 and the annual thermal energy delivered to the load center from Table 2.2. As was noted in Sect. 2, the annual delivered energy is assumed to be the same as the annual production at the power plant, on the basis of an analysis of pipeline heat losses and pumping energy input in ref. 2. The resulting unit total costs and the major cost components for the hot water transmission system are given in Table 7.5 for the 600- and 1200-MW(t) capacities, two supply temperatures, and two service modes.

**Table 7.5. Summary of unit transmission costs in 1982 dollars/$10^6$ Btu**

| | Cycles I and II-B | | Cycle II-A | |
|---|---|---|---|---|
| | 600 MW(t) | 1200 MW(t) | 600 MW(t) | 1200 MW(t) |
| Investment | | | | |
| Firm | 4.140 | 3.383 | 3.748 | 3.018 |
| Oil-interruptible | 4.871 | 3.978 | 4.409 | 3.550 |
| Maintenance | | | | |
| Firm | 0.216 | 0.176 | 0.195 | 0.157 |
| Oil-interruptible | 0.254 | 0.207 | 0.230 | 0.185 |
| Electricity | | | | |
| Firm | 0.301 | 0.302 | 0.250 | 0.246 |
| Oil-interruptible | 0.354 | 0.355 | 0.295 | 0.289 |
| Total | | | | |
| Firm | 4.657 | 3.861 | 4.193 | 3.421 |
| Oil-interruptible | 5.479 | 4.540 | 4.934 | 4.024 |

## 7.7 DISCUSSION OF RESULTS

The calculated unit transmission costs in Table 7.5 quantify several important economic factors evaluated in this study. First, the unit cost of the firm service mode, with its higher utilization of the capital intensive transmission systems, is decreased approximately $0.8/$10^6$ Btu compared with oil-interruptible service. Second, the higher temperature (350°F) transmission design, Cycle II-A, decreases unit costs by $0.4–0.5/$10^6$ Btu compared with the lower temperature (250°F) design of Cycles I and II-B. Finally, the 1200-MW(t) capacity costs are $0.8–0.9/$10^6$ Btu less than the 600-MW(t)-capacity costs because of the economies of scale in constructing the larger system.

## REFERENCES FOR SECTION 7


1. J. V. Naraby et al., World Energy Conference, *Telethermics*, report of the Ad Hoc Committee on Large Scale Transportation of Heat Over Long Distances, R. R. Clark, Ltd., Edinburgh (1971).
2. G. A. Englesson et al., *Economic Comparison of New Coal-Fueled, Cogeneration Power Plants for District Heating and Electric-Only and Heat-Only Power Plants*, ORNL/TM-6830/P8, Oak Ridge National Laboratory, Oak Ridge, Tenn., May 1982.

# 8. RESULTS OF ECONOMIC EVALUATION OF UNIT COST OF DELIVERED THERMAL ENERGY

The total unit cost of hot water delivered to the load center at the High Bridge plant in St. Paul from the Prairie Island plant is given in this section from results in Sects. 6 and 7. The sensitivity of the total unit cost to several of the more important variables is also discussed. The unit cost of delivered thermal energy is projected for reduced transmission distance and compared with the cost of thermal energy from alternative sources.

## 8.1 BASE CASE RESULTS

The total unit cost results given here are based on the median value of 18%/year for the fixed charge rate (FCR) and the specific transmission route and distance selected from Prairie Island to the load center in St. Paul.

The total unit cost of delivered hot water is composed of the unit cost at the plant gate from Sect. 6 and the unit cost of transmission from Sect. 7. For the assumed transmission system utilization factors assumed in this study, the total unit cost results in mid-1982 dollars/$10^6$ Btu are given in Table 8.1 for 600- and

**Table 8.1. Total unit cost of delivered thermal energy from the Prairie Island plant (in mid-1982 dollars/$10^6$ Btu)**

| | Cycle I | Cycle II-A | Cycle II-B |
|---|---|---|---|
| 600 MW(t), firm service (delivered energy, $9.84 \times 10^{12}$ Btu/year), TCF[a] = 54.8% | 8.27 | 8.69 | 8.70 |
| 600 MW(t), oil-interruptible service (delivered energy, $8.36 \times 10^{12}$ Btu/year), TCF = 46.6% | 7.94 | 7.99 | 7.99 |
| 1200 MW(t), firm service (delivered energy, $14.76 \times 10^{12}$ Btu/year), TCF = 41.2% | 7.66 | 8.00 | 7.98 |
| 1200 MW(t), oil-interruptible service (delivered energy, $12.55 \times 10^{12}$ Btu/year, TCF = 35.0% | 7.23 | 7.18 | 7.15 |

[a]TCF = transmission capacity factor.

1200-MW(t)-capacity systems for single- and two-turbine retrofit cases, respectively. The three plant cycle options are as follows:

1. Cycle I—extraction from high-pressure (HP) element exhaust to new back-pressure turbines to 250°F hot water heat exchangers for direct transmission.
2. Cycle II-A—extraction from HP element exhaust plus HP bleeds to 350°F hot water heat exchangers for indirect transmission.
3. Cycle II-B—extraction from HP element exhaust to 250°F hot water heat exchangers for direct transmission.

## 8.2 EFFECT OF FIXED CHARGE RATE

The largest component of the total unit cost is the fixed cost for financing the capital investment of the transmission system. A credible range of FCR values from 0.15 to 0.21 was identified in Sect. 5 to cover higher and lower true interest rates and also shorter and longer periods of useful life. Therefore, the incremental unit cost for a 0.03 increment in the FCR was calculated for the investment costs in the plant retrofit and the transmission system. Figure 8.1 shows the resulting total and transmission unit costs over the range of 0.15 to 0.21 in FCR values.

## 8.3 EFFECT OF TRANSMISSION DISTANCE

As was stated in the introduction to this report, this study was performed to provide generic-type information rather than specific results only for the Prairie Island plant and Twin Cities load center. In this context, the specific results for the Prairie Island plant are generalized to consider transmission distances other than the 36-mile route selected in Sect. 4.

In general, the transmission distance should be minimized to reduce the transmission unit cost. This statement must be qualified because the Metcalf & Eddy study of transmission route options in Appendix C has shown that the shortest route may not always have the lowest construction cost because of high cost segments such as bridges or deep tunneling. The most economical route can therefore contain a mix of pipeline construction techniques—concrete culvert, stanchion support, deep tunneling, and bridge support. Since the 36-mile route from Prairie Island contains segments of all of these construction techniques, the total construction cost results can be used as a basis for estimating a representative unit transmission cost for other transmission distances. The 36-mile Prairie Island route consists of the following percentages of lengths—concrete culvert, 48.1%; stanchion-supported duct, 33.3%; tunnel, 12.5%; bridge support, 6.1%. The construction cost of other routes with combinations of pipeline construction techniques different from those represented by the Prairie Island route can be made on the basis of the unit cost—dollars per linear foot—in Sect. 4.

The unit transmission costs for the Prairie Island route range from 3.4 to 5.5 1982 dollars/$10^6$ Btu at an 18%/year FCR for the range of capacity factors

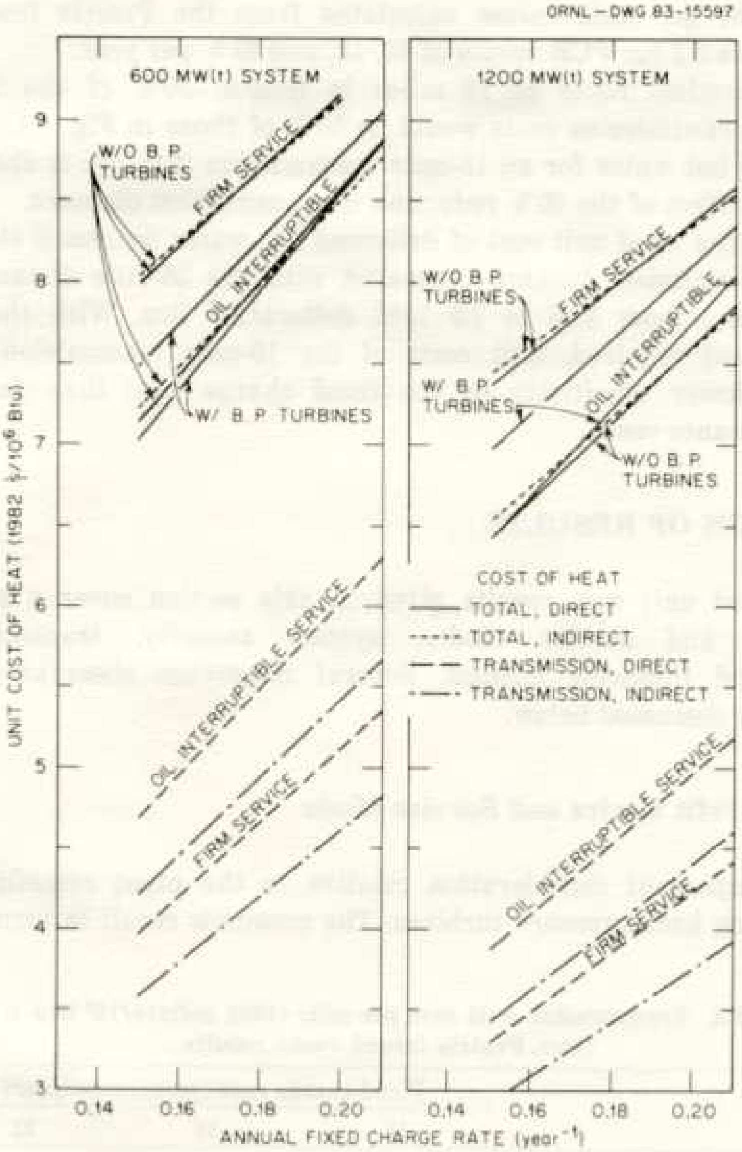



**Fig. 8.1. Unit cost of heat vs fixed charge rate for hot water transmission—36 miles—to Twin cities load center.**

considered in this study (see Fig. 8.1). To reduce the total unit cost of delivered hot water to be competitive with alternative sources of thermal energy, transmission distances less than 36 miles should be considered. Also, most of the construction cost—from 90 to 95%—of the transmission is represented by length-dependent components such as the pipelines, pumping stations, and pumping energy. Considering the pumping station requirements, the unit transmission cost per unit transmission distance will vary slightly with transmission distance. However, for this generic estimation purpose, we will assume that the unit transmission cost per unit transmission distance for the 36-mile route to Prairie Island applies to transmission distances between 18 and 36 miles (29 and 58 km). The unit transmission cost per mile values calculated from the Prairie Island route are tabulated in Table 8.2 for FCR values of 15, 18, and 21% per year.

For a hypothetical route of 18 miles in length—50% of the Prairie Island route—the unit transmission costs would be 50% of those in Fig. 8.1. The total unit cost of delivered hot water for an 18-mile transmission distance is shown in Fig. 8.2 to illustrate the effect of the 50% reduction in transmission distance.

As expected, the total unit cost of delivered hot water decreases significantly for the 18-mile transmission distance compared with the 36-mile distance, with total unit costs ranging from 4.45 to 7.0 1982 dollars/$10^6$ Btu. With the much lower capital investment required, unit costs of the 18-mile transmision distance also show a much lower sensitivity to the fixed charge rate than for the 36-mile transmission distance costs.

## 8.4 DISCUSSION OF RESULTS

The calculated unit cost results given in this section cover a range of plant retrofit cycles and service modes, system capacity, transmission system temperature, and financing options. Several important observations concerning these factors are discussed below.

### 8.4.1 Plant Retrofit Cycles and Service Mode

The most important consideration relative to the plant retrofit cycles is the value of including back-pressure turbines. The economic result in terms of total unit

**Table 8.2. Transmission unit cost per mile (1982 dollars/$10^6$ Btu-mile) from Prairie Island route results**

| | Fixed charge rate (percentage/year) | | |
|---|---|---|---|
| | 15 | 18 | 21 |
| **600-MW(t) capacity** | | | |
| Direct transmission | | | |
| Firm service | 0.110 | 0.129 | 0.149 |
| Oil-interruptible service | 0.130 | 0.152 | 0.175 |
| Indirect transmission | | | |
| Firm service | 0.0993 | 0.117 | 0.134 |
| Oil-interruptible service | 0.117 | 0.137 | 0.157 |
| **1200-MW(t) capacity** | | | |
| Direct transmission | | | |
| Firm service | 0.0917 | 0.107 | 0.123 |
| Oil-interruptible service | 0.108 | 0.126 | 0.144 |
| Indirect transmission | | | |
| Firm service | 0.0798 | 0.0944 | 0.109 |
| Oil-interruptible service | 0.0951 | 0.111 | 0.128 |

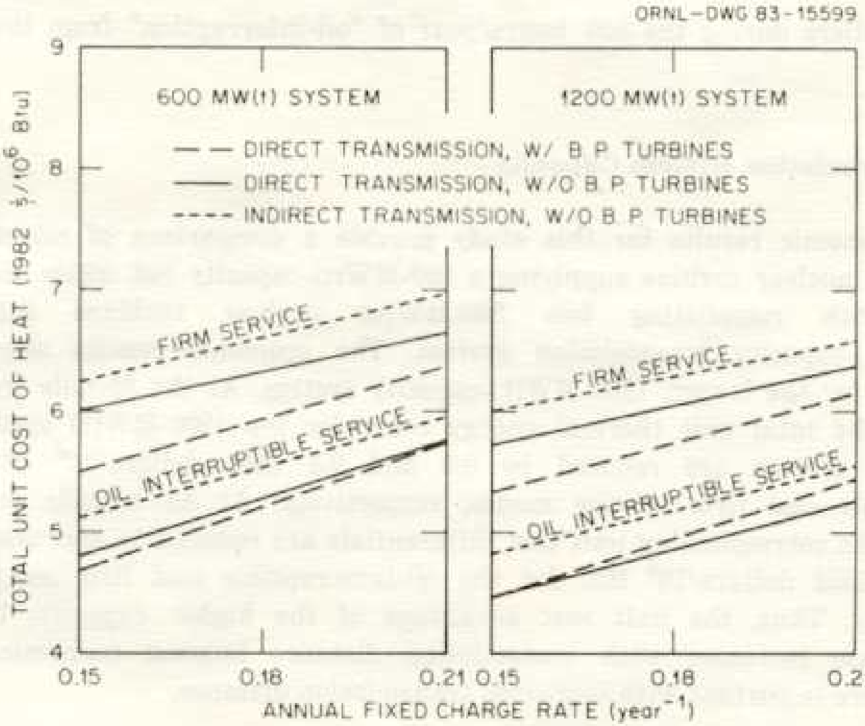



**Fig. 8.2. Total unit cost of heat vs fixed charge rate for an 18-mile transmission distance.**

cost of delivered hot water show that Cycle I, back-pressure turbines coupled with a 250°F transmision supply temperature, provides the lowest total unit cost for the firm service mode under all conditions of financing—that is, values of FCR. This is a result of the high value of electrical generating capacity via the high thermodynamic efficiency of the cycle, which translates into the lowest price for thermal energy purchased at the plant gate. Under the oil-interruptible service mode, the back-pressure turbine cycle economic advantage is not significant except at low FCR values. For the oil-interruptible service mode, all three cycles considered have very similar total unit costs, especially for the 36-mile transmission distance (see Fig. 8.1). Thus, the increased unit transmission cost with oil-interruptible service is essentially balanced by the reduced prices of thermal energy purchased from the cogeneration plant.

The reduced total unit costs for oil-interruptible service compared with firm service must be considered in terms of the unit cost of the next highest cost source of back-up thermal energy. If the unit cost of thermal energy from back-up sources exceeds the firm service total unit cost, then the firm service mode should provide the most economical choice for supplying the base-load thermal energy. For oil-fuel back-up heat sources, the fuel cost is \$6.25/$10^6$ Btu for \$30/bbl oil at an 80% combustion efficiency. Adding fixed and O&M costs, the total thermal energy cost would be at least \$10-12/$10^6$ Btu, depending on the annual production. Therefore, the firm service cogeneration unit cost will be more economical than using oil-fueled back-up boilers during the 500 hours/year of "oil-interruption" from the electrical system.

### 8.4.2 Transmission System Capacity

The economic results for this study provide a comparison of retrofitting one 560-MW(e) nuclear turbine supplying a 600-MW(t)-capacity hot water transmission system with retrofitting two 560-MW(e) nuclear turbines supplying a 1200-MW(t)-capacity transmission system. The economic results show a clear advantage for the larger, 1200-MW(t)-capacity system. At the 36-mile transmission distance, the total unit thermal energy costs for the 1200-MW(t) system vs the 600-MW(t) system are reduced by 0.6 and 0.8 1982 dollars/$10^6$ Btu for oil-interruptible and firm service modes, respectively. At an 18-mile transmission distance, the corresponding unit cost differentials are reduced to approximately 0.25 and 0.35 1982 dollars/$10^6$ Btu for the oil-interruptible and firm service modes, respectively. Thus, the unit cost advantage of the higher capacity, two-turbine retrofit case increases with transmission distance because economics of scale become more important with increased transmission distance.

### 8.4.3 Transmission System Temperature

The transmission system temperatures in this study allow a comparison of direct hot water transmission at 250°F supply and 90°F temperature difference with indirect transmission at 350°F supply and 170°F temperature difference. These two transmission system options balance the lower unit transmission cost for indirect transmission with the higher price of thermal energy from the less efficient plant retrofit cycle. For the 36-mile transmision distance, the indirect transmission (Cycle II-A) total unit costs are very close to the unit costs for the direct transmission without back-pressure turbines (Cycle II-B). The indirect transmission option has a slight economic advantage only at FCRs above 0.20 for the 36-mile transmission distance. At the 18-mile transmission distance, the indirect transmission mode has the highest total unit costs of the three plant cycles analyzed. Therefore, the indirect, higher temperature transmission system does not appear to offer any economic advantage for the range of conditions considered in this study.

### 8.4.4 System Financing Options

System financing options have been treated through considering a variation in the Fixed Charge Rate (FCR) from 0.15 to 0.21/year with a median value of 0.18/year. (The corresponding interest rate, system economic life, and other financial parameters are discussed in Sect. 5.3.) The total unit cost results for 36- and 18-mile transmission distances in Figs. 8.1 and 8.2 show the range of unit costs over the 0.06 FCR range.

As expected, the 36-mile transmission unit costs show the greatest sensitivity to the FCR, so the unit costs at an FCR of 0.21 are from 0.6 to 0.9 $\$/10^6$ Btu higher than at a FCR of 0.18. For the 18-mile transmission distance, the incremental unit cost ranges from 0.3 to 0.5 $\$/10^6$ Btu for an 0.03/year FCR increment. Therefore, combinations of high FCR and long transmission distances are to be avoided if reasonable transmission costs are to be obtained from a remote source of thermal energy.

### 8.4.5 Comparison with Alternate Thermal Energy Sources

In order that the unit cost results based on hot water transmission from a retrofitted nuclear plant may have meaning, they must be compared with the unit cost for other potential sources of hot water thermal energy. The results of a previous economic analysis of new coal-fueled cogeneration power plants, performed for the future Minneapolis-St. Paul district heating market, provide a good basis of comparison for this study's results.[1] The thermal energy unit cost results from ref. 1 included a proportionate share of the plant capital and financing costs, so the thermal energy costs represent a fair distribution of the total annual costs of the plant. One of the cogeneration plants had electrical/thermal production capacities of 400 MW(e)/700 MW(t). The annual thermal production was 2,846,121 MW(t) h/year or $9.711 \times 10^{12}$ Btu/year, which compares closely with the $9.84 \times 10^{12}$ Btu/year from the 600-MW(t)-capacity firm service case in this study.

The lowest cost coal-fueled plant in ref. 1 was a closed-cycle gas turbine (CCGT) plant. The unit cost of thermal energy at the plant gate was $\$9.28/10^6$ Btu in May 1989 dollars, assuming a 15%/year fixed charge rate. Adjusting the fixed costs to an 18%/year FCR and de-escalating to mid-1982, the CCGT unit cost of thermal energy is $\$5.88/10^6$ Btu, of which $\$1.83/10^6$ Btu is fuel cost.

The total unit cost of delivered thermal energy for firm service and an 18%/year FCR is presented as a function of transmission distance in Fig. 8.3. The $\$5.88/10^6$ Btu cost at the plant gate for the 700-MW(t) CCGT plant compares with the \$3.60 and $\$3.81/10^6$ Btu unit cost at the plant gate for one- and two-turbine retrofit cases with firm service, respectively. The lower cost thermal energy for the retrofitted nuclear turbines allows the nuclear cogeneration heat source to compete with the lowest cost, new coal-fueled cogeneration source out to 18 to 20 miles of transmission distance. Therefore, the total unit cost of thermal energy delivered 36 miles from Prairie Island compares with the total unit cost delivered from a CCGT plant over a 16- to 18-mile transmission distance, assuming the same transmission cost/mile applies to the CCGT plant. The retrofitted nuclear cogeneration source of 600–1200 MW(t) of thermal energy capacity compares very favorably with a new coal-fueled cogeneration source if the transmission distance is kept below approximately 20 miles, for which the delivered thermal energy cost is approximately $\$5.90/10^6$ Btu.

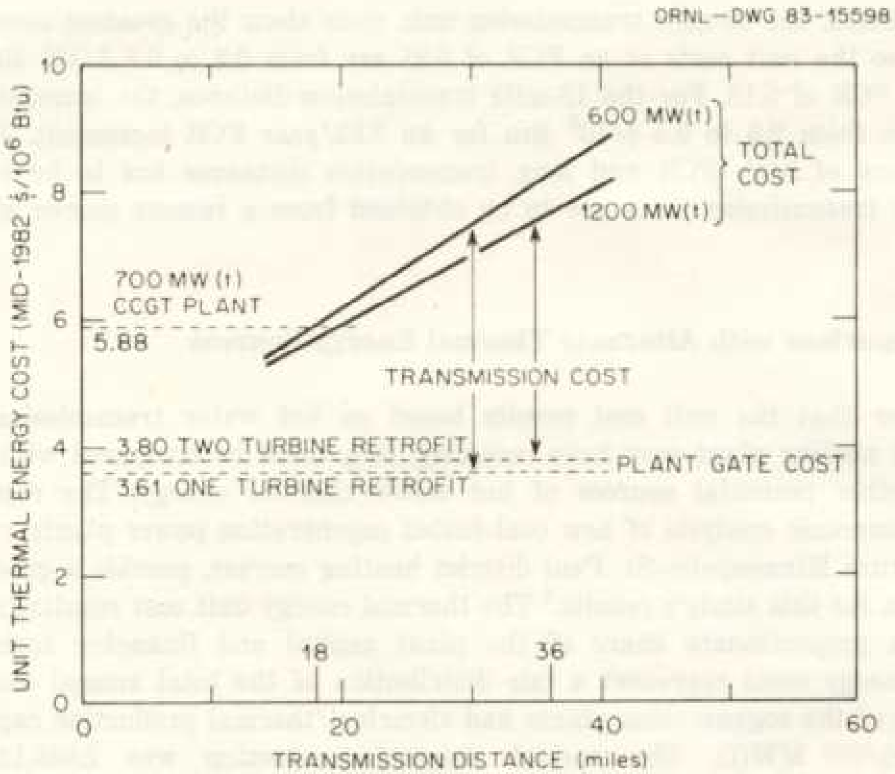



**Fig. 8.3. Comparison of "firm" thermal energy cost from retrofitted nuclear turbines with coal-fueled CCGT plant**

In terms of the lowest cost of cogenerated hot water, existing coal-fueled power plants operated on an intermediate-load basis should provide the lowest cost thermal energy. This is generally true because of two factors:

1. Transmission distance — Plants such as the NSP High Bridge plant in St. Paul and the Riverside plant in Minneapolis are often located relatively close to the district heating load center. Thus, the cost of large transmission pipeline systems can be held to a small portion of the total cost.
2. Thermal energy price at the plant gate — In general, an intermediate-load plant will have more hours/year of "excess" electric energy production than a base-loaded plant. Since the "excess" energy cost is always less than the "replacement" energy cost, which largely determines the price from a base-loaded plant, the purchase price from the electric utility should be similar to or less than the energy price from a base-loaded coal or nuclear plant. (This general statement could be invalid for a particularly inefficient turbine or for a high cost for retrofitting to cogeneration.)

Therefore, such existing power plants would be used first in supplying a growing hot water district heating market.

## REFERENCE FOR SECTION 8

1. G. A. Englesson et al., *Economic Comparison of New Coal-Fueled, Cogeneration Power Plants for District Heating and Electric-Only and Heat-Only Power Plants*, ORNL/TM-6830/P8, Oak Ridge National Laboratory, Oak Ridge, Tenn., May 1982.

# 9. SUMMARY AND CONCLUSIONS

## 9.1 SUMMARY OF MAJOR STUDY CHARACTERISTICS

This study is one of a series of investigations of the technical feasibility and economics of supplying a large U.S. metropolitan area such as Minneapolis/St. Paul with cogenerated thermal energy from utility-owned power plants. The purpose of this study was to determine the cost of hot water thermal energy from an existing PWR utility power plant delivered to a hypothetical Twin Cities area load center at a future time not earlier than 1992. Other related studies have considered existing coal-fueled plants, new coal-fueled plants, and new PWR plants as the sources of cogenerated hot water for district heating.

The two-unit Prairie Island plant of NSP was the specific plant chosen as the basis for determining the first-level retrofit design and capital estimate. Each of the 560-MW(e) Westinghouse turbines was assumed to be a potential source of up to 600 MW(t) of thermal energy, feeding into a hot water transmission system of either 600- or 1200-MW(t) capacity, depending on whether one or two turbines were retrofitted to cogeneration operation. Two transmission system design temperature levels were analyzed and capital costs developed; the first priority design was for 250°F supply and 160°F return temperatures for direct transmission to the distribution system, and the second priority design was for 350°F supply and 180°F return temperatures for indirect transmission through water-to-water heat exchangers at the load center connection. The two water transmission temperature levels also determined different turbine electrical capacity derates for cogeneration operation.

A load duration curve developed for a future 2600-MW(t) Twin Cities distribution system was used as the basis for the analysis of annual thermal energy supplied from a retrofitted Prairie Island plant and the corresponding average energy costs. The annual system thermal energy demand for this 2600-MW(t) system was $20.57 \times 10^{12}$ Btu ($21.7 \times 10^{7}$ GJ). Average thermal energy costs were also calculated for two service conditions. Firm service to the hot water transmission system requires the cogeneration plant to follow the demand for thermal energy in preference to the electrical system demand, which normally requires base-load operation of nuclear plants like Prairie Island. Therefore, average costs of thermal energy from Prairie Island were estimated by NSP for both "firm" service and "oil-interruptible" service. For the latter case, cogeneration of thermal energy would be restricted whenever high-cost, oil-fueled peaking generators were called on to meet the electrical system demand. The 600-MW(t) transmission system capacity factors were 54.8 and 46.6% for firm and oil-interruptible service, respectively, and the corresponding 1200-MW(t) system capacity factors were 41.2 and 35.0%, respectively.

## 9.2 SUMMARY OF RETROFITTED TURBINE PERFORMANCE CHARACTERISTICS

The analysis of practical steam extraction conditions and plant modifications for hot water cogeneration operation was performed by the Westinghouse Electric Corporation, suppliers of the Prairie Island plant turbines. A primary goal of this effort was to determine a retrofit design both including and excluding back-pressure turbines between the steam extraction locations and the hot water heat exchangers. Inclusion of back-pressure turbines increases the overall thermodynamic cycle efficiency by producing the maximum electrical energy for the steam-condensing condition established by the hot water supply temperature.

The Westinghouse analysis concluded that extraction of 165 psia steam from the crossunder piping between high-pressure (HP) and low-pressure (LP) turbine elements was the only practical extraction location for the large steam flows required—approximately 2 million lb/h—to produce 600 MW(t) of thermal energy. Consideration of steam velocity limit conditions led to the inclusion of a special valve having both reheat/stop and throttling functions in the crossover piping between the HP and LP turbine elements. For the direct transmission mode at 250°F/160°F supply/return temperatures, steam extraction from the HP to LP turbine element crossunder piping location could produce 600 MW(t) of thermal energy. However, for the indirect transmission mode at 350°F/180°F, an additional high pressure extraction from the No. 5 feedwater heater of the HP element is required to produce the higher water temperatures.

A summary of the primary conditions of the three cycles considered is shown in Table 9.1. The back-pressure turbine cycle results in the lowest electrical capacity derate and, hence, the highest overall cycle efficiency. The "net MW(e) reduction" values are from a nominal rating of 560 MW(e) per turbine unit; these electrical capacity derate values are not permanent capacity derates because the unit can be returned to full condensing operation at any time by stopping steam extraction for hot water production.

**Table 9.1. Summary of primary cycle conditions for a 560-MW(e) PWR retrofitted to cogeneration of 600 MW(t)**

| Back-pressure turbines | Net MW(e) reduction | Extraction steam flow ($10^6$ lb/h) | DH water supply/return temperature (°F) | DH water flow ($10^6$ lb/h) |
|---|---|---|---|---|
| Yes | 112 | 1.88 | 250/160 | 22.6 |
| No | 142 | 2.08 | 250/160 | 22.6 |
| No | 160 | *a* | 350/180 | 11.8 |

[a]Not determined.

## 9.3 SUMMARY OF ECONOMIC RESULTS

### 9.3.1 Investment Costs

The total investment costs for retrofitting the Prairie Island unit or units and construction of a 36-mile hot water transmission system to the load center in the Twin Cities have been developed. Table 9.2 summarizes these investment costs in mid-1982 dollars for the plant retrofit and transmission system options considered. The investment costs include an allowance for funds during construction (AFDC) and are based on operation beginning in mid-1992 with escalation of construction costs included between 1982 and 1992.

**Table 9.2. Investment costs (in millions of mid-1982 dollars) for Prairie Island plant retrofit and transmission system to the Twin Cities**[a]

| | Cycle I[a] | | Cycle II-A[b] | | Cycle II-B[c] | |
|---|---|---|---|---|---|---|
| | 600 MW(t) | 1200 MW(t) | 600 MW(t) | 1200 MW(t) | 600 MW(t) | 1200 MW(t) |
| Plant retrofit | 32.2 | 64.4 | 13.8 | 27.5 | 13.8 | 27.5 |
| Transmission system | 226.2 | 277.4 | 204.8 | 248.5 | 226.2 | 277.4 |
| Total | 258.4 | 341.8 | 218.6 | 276.0 | 240.0 | 304.9 |

[a]Cycle I: Extraction to back-pressure turbines; transmission at 250°F/160°F.
[b]Cycle II-A: Extraction to hot water heat exchangers; transmission at 350°F/180°F.
[c]Cycle II-B: Extraction to hot water heat exchangers; transmission at 250°F/160°F.

### 9.3.2 Unit Costs of Thermal Energy

The unit cost of thermal energy delivered to the Twin Cities load center includes the two major elements, the unit cost at the plant and the unit transmission cost. Both of these elements have investment cost components which have been calculated on the basis of a nominal fixed charge rate (FCR) of 18%/year with a range in FCR values from 15 to 21% per year considered in a sensitivity analysis. The unit costs are given in Table 9.3 for the system options considered in this study.

The predominant part of the unit cost at the plant gate is the total unit energy price the electric utility owning the nuclear power plant charges the transmission utility company. The total unit energy price, in turn, is predominantly determined by the electric utility to cover all costs associated with operating the nuclear plant in an electric/thermal energy cogeneration mode and does not include a capacity charge for the plant's initial investment cost. The unit energy prices in this study were estimated by NSP for projected Prairie Island operation in the 1992-1997 time period

### 9.3.3 Comparison of Unit Cost Results Based on Prairie Island Retrofit with Alternate Energy Sources

The present results for the lowest firm-service unit cost of thermal energy including 36.0 miles of transmission from Prairie Island range from $7.7 to

**Table 9.3. Unit cost of thermal energy from the Prairie Island plant (in mid-1982 dollars/$10^6$ Btu)**

| | Cycle I | Cycle II-A | Cycle II-B |
|---|---|---|---|
| 600 MW(t), firm service | | | |
| Cost at plant gate | 3.61 | 4.50 | 4.04 |
| Transmission cost | 4.66 | 4.19 | 4.66 |
| Total | 8.27 | 8.69 | 8.70 |
| 600 MW(t), interruptible service | | | |
| Cost at plant gate | 2.46 | 3.06 | 2.51 |
| Transmission cost | 5.48 | 4.93 | 5.48 |
| Total | 7.94 | 7.99 | 7.99 |
| 1200 MW(t), firm service | | | |
| Cost at plant gate | 3.80 | 4.58 | 4.12 |
| Transmission cost | 3.86 | 3.42 | 3.86 |
| Total | 7.66 | 8.00 | 7.98 |
| 1200 MW(t), interruptible service | | | |
| Cost at plant gate | 2.69 | 3.16 | 2.61 |
| Transmission cost | 4.54 | 4.02 | 4.54 |
| Total | 7.23 | 7.18 | 7.15 |

$8.3/$10^6$ Btu in mid-1982 dollars. Such costs for thermal energy are relatively high considering that the cost of distribution within the district heating system must still be added to obtain the total cost of delivered thermal energy before any allowance for profit. Therefore, the linear dependence of transmission unit cost on transmission distance was used to estimate hypothetical transmission distances less than the 36 miles from Prairie Island at which the retrofitted nuclear cogeneration costs would be competitive with unit costs from alternative central station sources of cogenerated thermal energy for hot water district heating.

The alternative central station cogeneration plant chosen for this comparison is a new coal-fueled, closed-cycle gas turbine (CCGT) plant of 400-MW(e)/700-MW(t) electric/thermal energy capacities. The thermal energy costs for this type of plant were determined to be the lowest of several coal-fueled central station plants analyzed by United Engineers and Constructors, Inc., for NSP and ORNL in a companion study of Twin Cities district heating application. The unit cost of thermal energy at the plant gate is estimated to be $5.88/$10^6$ Btu in mid-1982 dollars for the CCGT plant, which compares with the $3.60 and $3.81/$10^6$ Btu unit costs at the plant gate for the one- and two-nuclear-turbine retrofit cases of this study. At a fixed charge rate of 18%/year, the $2.07 to $2.28/$10^6$ Btu differential in unit costs at the plant gate is equivalent to 18 to 20 miles of transmission distance on the basis of the average transmission cost per mile determined in this study for the route from the Prairie Island plant to the Twin Cities. Therefore, the lower thermal energy costs from a retrofitted nuclear plant when compared with the most economical new coal-fueled cogeneration power plant, can compensate for a significantly increased transmission distance, 18–20 miles in the 600- to 1200-MW(t) transmission system capacity range.

## 9.4 SUMMARY OF MAJOR CONCLUSIONS

The major conclusions from the three major tasks within this study are summarized below.

### 9.4.1 Nuclear Plant Retrofit to Hot Water Cogeneration

1. Steam extraction for 600 MW(t) of hot water thermal energy at 250°F is limited to the high-pressure turbine element exhaust in a two-element Westinghouse turbine such as at the Prairie Island Nuclear Plant. For turbine operation at maximum steam flow, the 560-MW(e) nominal electrical capacity would be derated to 448 and 418 MW(e), respectively, at 600 MW(t) of thermal energy production, with and without 10-MW(e) industrial back-pressure turbines included in the retrofitted system design.
2. The total project costs for installing the turbine retrofit and a 600-MW(t) hot water production plant are \$52.40/kW(t) and \$21.80/kW(t) in mid-1982 dollars for the cogeneration plants with and without four back-pressure turbines included. The same costs were assumed for retrofitting the second turbine.
3. No feasibility problems were uncovered in developing the turbine retrofit design, and there were no problems anticipated with respect to nuclear safety or licensing from cogeneration plant operation. Several operational concerns were identified as having potential effects on the plant availability.

### 9.4.2 Transmission System Design and Cost

1. The lowest cost transmission system route between the Prairie Island plant and the High Bridge plant in St. Paul was 36 miles in length. This route was selected from an analysis of two similar routes having different geological and hydrological, and hence construction cost, characteristics.
2. On the basis of design and construction techniques in use in Scandanavian countries, 600- or 1200-MW(t) hot water transmission pipelines could be constructed between Prairie Island and the Twin Cities using several installation techniques such as concrete culverts, aboveground stanchions, and deep tunnels.
3. The total project costs in mid-1982 dollars per kW(t) of transmission capacity with 600 and 1200 MW(t) of direct mode transmission at 250/160°F supply/return temperatures are \$355 and \$217.50, respectively. The corresponding values for indirect mode transmission at 350/180°F supply/return temperature are \$321.67 and \$194.17, respectively.

### 9.4.3 Major Economic Analysis Results*

1. The most economic retrofit cycle and transmission mode for the Prairie Island turbine size [560 MW(e)] is the back-pressure turbine cycle producing hot water at 250°F/160°F supply/return temperatures. The unit investment costs for the turbine and plant retrofit contribute only \$0.6 and \$0.8 /$10^6$ Btu for this cycle with firm service from one- and two-turbine retrofits, respectively. These unit costs apply for an annual average thermal capacity of 55% for a one-turbine retrofit and 41% for a two-turbine retrofit.
2. The total unit cost of firm service thermal energy at the plant gate for a retrofitted nuclear turbine—estimated to be \$3.61 and \$3.80/$10^6$ Btu for one- and two-turbine retrofits, respectively—is approximately \$2-2.2/$10^6$ Btu less than the thermal energy cost from the most economical new coal-fueled cogeneration central station plant of comparable size. Therefore, a nuclear plant retrofit would be economic if its hot water transmission distance were no more than 20 miles greater than the transmission distance for the new coal plant alternative.
3. Retrofit of two turbines rather than one turbine and construction of a 1200-MW(t)-capacity transmission system rather than a 600-MW(t)-capacity system reduces the total unit cost of firm service thermal energy by \$0.8/$10^6$ Btu at 36 miles of transmission distance, even though the estimated annual average thermal capacity factor decreases from 55 to 41%.
4. The total unit cost of thermal energy delivered from the Prairie Island plant 36 miles to the Twin Cities load center is estimated to be \$8.3 and \$7.7/$10^6$ Btu for firm service from one- and two-turbine retrofits, respectively. Since the unit cost of thermal energy for an "oil-interruptible" service mode is only \$0.3 to \$0.4/$10^6$ Btu less than the firm service mode unit costs, firm service would represent the most economic service mode considering the cost of back-up thermal energy from fossil-fueled heat sources.
5. The economic attractiveness of cogenerated thermal energy from a retrofitted nuclear power plant such as Prairie Island depends in part on the transmission distance to other existing power plants in the same area—especially intermediate load plants—that could also be retrofitted for cogeneration operation. In general, transmission distances should be minimized to reduce the fixed investment costs of hot water for district heating. Therefore, practical applications of retrofitting a nuclear plant to cogeneration should emphasize reasonable transmission distances—less than ~20 miles—and transmission routes with a minimum of deep tunnelling and bridge crossings.

*These cost results are in mid-1982 dollars for an 18%/year fixed charge rate.

# Appendix A

# USE OF NUCLEAR-PRODUCED HEAT — LITERATURE SURVEY

# Appendix A
# USE OF NUCLEAR-PRODUCED HEAT — LITERATURE SURVEY

## A.1 PAST AND PRESENT

### A.1.1 United States

There is only one nuclear facility in the United States making use of nuclear energy for heating, the Experimental Breeder Reactor-II (EBR-II) located at the Idaho National Engineering Laboratory. This fast breeder reactor has been operating 17 years and has recently been employed for heating as well as its other functions. The reactor is a liquid metal cooled reactor. The hot metal is used to generate steam at 1250 psig and 800°F. Under full power, the plant generates 62.5 MW(t), and it can generate up to 19.5 WW(e).[1]

The energy needed for heating is taken from the turbine supply system at 1250 psig and 800°F. The steam is reduced 150 psig and 385°F so it can be used in space heating equipment. Steam at rates up to 12,000 lb/h has been diverted to the space heating equipment. In 1980, 64 million lb of steam were used in this manner.

The system has been operating since September 1974. Because of the high price of oil and the modest price they receive for sold electricity, the operation is estimated to have saved the facility $650,000 in 1980.

Since they are using header steam, this is not actually cogeneration, at least not in the sense that energy is used in a two-step cascade. In fact, in this system, there are no net energy savings. The dollar savings occur because an inexpensive energy source, nuclear fuel, is used to replace an expensive energy source, heating oil. Further, this is not district heating since the energy is used to heat only one facility.

Nevertheless, this is a U.S. example of use of thermal energy from a nuclear facility. It has been operating dependably for seven years.

### A.1.2 Canada

At Ontario's Hydro Bruce Nuclear Power Development, 3 million lb/h of steam operate their heavy water plant. In 1981, the amount of steam used for this purpose at that site will double. Steam from nuclear fuel from the Bruce units is estimated to cost about one-half as much as steam from coal.[2]

Bruce Nuclear Generating Station A consists of four units which were commissioned in 1977, 1977, 1978, and 1979. Each is capable of producing

791 MW(e) gross and 740 MW(e) net. The units use natural uranium dioxide ($UO_2$) and deuterium oxide (heavy water, $D_2O$) as a moderator. The coolant, pressurized heavy water, enters the reactor at 249°C (480°F) and leaves at 300°C (572°F) and 9.18 MPa (1332 psia). Four pumps are used for each reactor, and the hot heavy water is delivered to eight boilers. The boilers are capable of producing 10 million lb/h of steam at 256°C (492°F) and 2.27 MPa (620 psig).[3,4]

Bruce A then has four reactors, each capable of producing 10 million lb/h of steam for a total of 40 million lb/h. Of this, 3 million lb, or about 8%, is used to produce heavy water. When header steam is used, the system is characterized as being "parallel flow cogeneration."[5] It is stated that this has the advantage of maximizing the capacity factor of the reactors at all times, even when a turbine generator is unavailable, and if steam demand drops, the electrical production increases.

Any of these four units can be tapped for the 3 million lb/h of steam needed to operate the heavy water plant. In addition, the heavy water operation is doubling in 1981. The heavy water plant uses the exchange of deuterium between liquid water and gaseous hydrogen sulfide. At low temperatures, 90°F, the deuterium migrates to the water, and at high temperatures, 262°F, the deuterium migrates to the hydrogen sulfide. By counterflowing water and hydrogen sulfide in a perforated tray column, with the lower portion at the higher temperature and the top part of the column at 90°F, the deuterium is concentrated in the hydrogen sulfide at the midpoint of the column. Some of this gas is extracted and fed to a subsequent column where the process is continued. After three such columns, the enriched water passes to a distillation column where 99.75% $D_2O$ is produced. The steam is required to heat the columns and to distill the water. The distillation requires a surprising amount of steam since the very close boiling points, 212°F for $H_2O$ and 214.6°F for $D_2O$, require a large reflux and, therefore, a large reboiler. That is, each molecule of $H_2O$ or $D_2O$ must be vaporized many times in the column before a separation producing 99.75% pure $D_2O$ can be achieved.

Again, this operation is not strictly cogeneration nor is it district heating, but it is a necessary use of heat from nuclear power plants.

### A.1.3 Russia

The Russians are experimenting with the use of condenser water for greenhouse heating.[6] Both fossil-fuel-fired and nuclear power plants are being studied. A system in which hot water passes over the greenhouse skin or surface and acts as a thermal barrier is being investigated, and a dry system is being studied. The information available is not very complete. Since condenser water from large power plants is used for greenhouse heating in this country, this does not seem extremely interesting.

The USSR produces both electricity and heat from three nuclear reactors at Bilibino. The reactors are channel-type PWR. Each reactor has a total capacity of 100 MW(t). The thermal energy is used to supply heat to a community of 15,000 and for mining activity.[7] This facility is in a very remote region, and it would be difficult to transport fuel to the power plant.

### A.1.4 Sweden

In Sweden, the Agesta nuclear power plant supplied both heat and electricity for a decade.[7] This plant, which was capable of producing 55 MW(t) for heating and 12 MW(e), was started up in 1964. It supplied heat to a district heating network in a Stockholm suburb. It was shut down in 1974 when the operation of the small plant was no longer economical.

The Agesta nuclear power plant was a heavy-water-moderated, pressurized-water reactor. Its total production was 80 MW(t).

### A.1.5 Switzerland

Although no cogeneration is employed, a major use of nuclear energy for process steam has been in operation since 1979 at Niedergösgen.[8] The 920-MW(e) Gösgen nuclear plant supplies up to 47 MW(t) of superheated steam at 14 bars and 222°C (206 psia and 432°F) to a cardboard mill through a 1750-meter-long (1.09-mile) steam line. The process steam is raised in an evaporator from turbine steam.

## A.2 PLANNED OR PROPOSED

### A.2.1 United States

Probably the largest near-term use of thermal energy from a nuclear power plant in the United States could be the use of thermal energy from the Midland Power Plant.[9] The plant is being built in Midland, Michigan, by Consumers Power Company (Jackson, Michigan). It was planned to supply up to 4 million lb/h of steam to the Dow Chemical Company's industrial complex, which is adjacent to the nuclear station. This project has been significantly delayed by regulatory and financing problems, and its operational schedule is unknown at this time.

The plant will consist of two units with an aggregate production capability of 1300 MW electrical and 4 million lb/h of steam, or about 1200 MW thermal. Unit 2 will produce primarily electrical power but will also supply steam as a backup when Unit 1 is down. It will have a maximum rating of 805 MW(e).

Unit 1 will produce electrical power and steam for the Dow Chemical Plant. It will have a maximum electrical output of 541 MW at a process steam flow of 2 million lb/h. At design, the unit will produce 457 MW(e) and 4 million lb/h of steam. As the steam demand decreases, the electrical production increases until a rate of 547 MW is reached at 2 million lb/h of steam use. As the process steam use decreases below this point, it will be necessary to decrease the electrical output until a level of 425 MW(e) is reached at no process steam flow. At 2 million lb/h of process steam and below, the LP turbine is back end limited, and further decrease of process steam use will require a reduction of steam flow through the HP turbine and corresponding decreases in electrical output. During the initial years of operation at Midland, Unit 1 will be operating at or near the peak electrical capability much of the time.

The Dow plant will use two steam systems. A high-pressure process steam will use 400,000 lb/h of steam at 600 psig, and the low-pressure design flow will be at 175 psig. The smaller flow, high-pressure steam, will be produced from steam taken from the secondary steam flow as it leaves the containment building. This steam will not go to the turbine, and in a sense, this 10% of the process steam will not really involve cogeneration.

The low-pressure process steam will be produced at a rate up to 3,650,000 lb/h at 175 psig, and it will indeed be produced in a cogeneration mode. Nearly ten million pounds per hour of high-pressure steam will enter the turbine. At the penultimate stage in the high-pressure turbine, 3,960,000 lb/h of this steam will leave the turbine cycle through an uncontrolled extraction and flow to the evaporator building. In this building, the steam will evaporate the process steam.

The reactors are BWR type, and the fluid in the primary thermal loop will not leave the containment building. The steam sent to the turbines, therefore, will be secondary steam. This steam will be extracted or, in the case of the high-pressure steam, header steam will be taken and transferred to an evaporator building. In the evaporator building, the secondary steam will produce tertiary steam which will be sent to Dow Chemical for process steam. The evaporators, which will provide an additional mechanical barrier between the secondary steam and the process steam received by Dow, are one of two significant changes to the original 1970 design. The other design change is the use of a cooling pond rather than towers.

Because of conservation measures adopted by Dow over the years and relocation of some processes from Midland to other places, the steam demand at Dow's plant is currently only 2 million lb/h. Increased electrical output will result.

After the tertiary steam leaves the evaporator building it will be transported in two 48-in., 175-psig, and one 24-in., 600-psig, lines to Dow. The line lengths are about 1 to 1.5 miles. The condensate return from Dow will actually be a 60/40 mixture of condensate return and demineralized makeup water.

The heat rate for Unit 2 will be 10,390 Btu/kWh. The heat rate for Unit 1 at full extraction will be 8080 Btu/kWh. It is obvious that operation at this point will result in a savings of 2310 Btu for every kilowatt hour generated by Unit 1 as compared to generating electricity from the power only Unit 2 turbine generators.

### A.2.2 Sweden

In Sweden, where some of the earliest experience with nuclear district heating occurred, a number of plans have been under consideration over the past few years. In spite of an antinuclear population or force, there seems to have been small opposition to nuclear district heating. In fact, the opposition to nuclear district heating of late has been based on the assertion that nuclear district heating will prolong the life of existing power plants.

One of the proposed developments would be the joint Sweden-Finland project, SECURE (Safe Environmentally Clean Urban Reactor).[10] This is a small, low-cost reactor with properties that would allow location near urban centers, and it would allow nuclear heat for small urban centers. The reactor is 200 MW(t), with no electrical production. The reactor would be operated on light water at 7 atmospheres, and the water would be heated to 90–114°C. The reactor water would be used to heat district heating water to 60–95°C. The plant would serve a metropolitan area with about 100,000 population. The whole reactor installation would be underground.[10] This proposal may not be receiving active consideration at this time.

One of the proposals for Sweden has been to transport hot water from the Basseback nuclear power plant to areas in southern Sweden, with a population of greater than one-half million, and to greater Stockholm, with a population of one and one-half million. This project may not be active at this time.

The use of thermal energy from the Forsmark nuclear power station to supply heat to Stockholm and adjacent district heating systems has been actively considered since 1978.

This plant has three BWR units, each capable of producing 1050 MW electrical.[11] Two turbine retrofits have been proposed, direct bleeding alternative and back-pressure turbine alternative.

With the direct bleed, the unit will produce 2000 MW(t) and 300 MW(e). With the back-pressure option, Unit 3 will produce 2000 MW(t) and 500 MW(e).

The hot water transport distance is 185 km. Pipe required will be 1.5-m diam for 2000 MW(t) and 1.4 m for 1700 MW(t). Maximum temperature will be 160°C, which is high by Sweden's standards, and the return will be 65°C. Maximum pressure will be 25 atmospheres, and four pumping stations will be required along the way plus one each at Forsmark and Stockholm.

The radioactive water will be triply isolated from the homes. Isolated heat exchangers will be in the system at Forsmark, in the city (Stockholm or other) and in the home. In addition, the arrangement at Fosmark is such that the pressure in the district heating system will be higher than the pressure in the steam system, and any leaks would be into the radioactive loop and not into the district heating system.

The transportation pipeline is expected to be one-third in tunnels and two-thirds in culverts.

The plant conversion cost is expected to be about $200 million for the direct bleeding process, or $330 million for the back-pressure operation. The pipeline construction cost is estimated to be $750 million, for totals of just under and just over one billion dollars, respectively.

The system would be base loaded, with the current local boilers used for peaking and backup.

A detailed design and cost estimate was completed in October 1981.[12] The project was not supported for federal financing in 1982, so it is currently not active. However, many design and economic data were developed for the concept in this project.

### A.2.3 Canada

The Bruce Nuclear Power Development is currently being used to produce steam for separation of heavy water. That operation is discussed in Sect. A.1.2, above. Now there is a proposal to use energy from the HWR to heat an industrial park.[13]

A 24-in., 2-mile-long steam line is planned, and a 10-in. condensate return line would also be required. The 500,000 lb/h of steam would deliver about 150 MW(t). The estimated cost is $1.90 (Canadian) per million Btu, or about $1.5 (U.S.) per million Btu. This is certainly an attractive price for steam.

### A.2.4 Finland

The SECURE reactor has been under consideration for the last several years.[10]

Studies for district heating at Helsinki with nuclear power have been made with the assumption that an extraction turbine would produce 860 to 1000 MW(e) and have a heat output of up to 800 MW(t). These plants would exceed the needed electrical capacity. Consideration is being given to the State Power Company's owning 50% of the electrical production. A long-distance hot water pipeline is envisioned.

### A.2.5 Switzerland

Recently, several projects have been activated to consider retrofitting existing nuclear power plant turbines to supply hot water district heating systems.[8] The most advanced project, called Refund, is considering steam extraction between high- and low-pressure turbine sections at the two-unit (350-MW(e) each) Beznan power plant. Initially, up to 52 MW(t) of 120°C hot water would be supplied to eight communities through main pipelines of 6 and 13 km (3.7 and 8 miles) in length. Startup of this system could be as early as the winter of 1983–84.

Other nuclear power plants being considered for retrofit to cogenerating for district heating include plants at Leibstadt and Muhleberg.

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## Appendix B

## PLANT AND TURBINE MODIFICATIONS

(*on microfiche—see inside back cover*)

## Appendix C

## TRANSMISSION LINE ROUTE OPTIONS

(*on microfiche—see inside back cover*)

## Appendix D

## PRICE OF THERMAL ENERGY

Appendix D

Task 5 Report

# PRICE OF THERMAL ENERGY

Prepared for
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January, 1983

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NOTICE

# APPENDIX D

## PRICE OF THERMAL ENERGY

### D.1 Introduction

Northern States Power Company (NSP) has been involved in negotiations with several potential thermal customers - both industrial customers and district heating systems. The basic approach, an incremental cost approach, presented here has been used to price out the thermal energy for those customers. Inherently, an incremental cost approach is conceptually simple but has a high degree of difficulty in application to specific cases. Further, negotiations have involved both the sale of direct boiler steam and cogenerated steam from NSP's coal fired generating plants. Northern States Power Company has contracted for, has constructed and is operating a $40 million, 5 mile high pressure steam pipe line for the sale of coal fired boiler steam to a paper company.

There is a significant difference between the cost of thermal energy and the selling price of thermal energy. "Cost" is what it costs to produce the energy, and "price" is what the thermal customers, based upon market conditions and terms and conditions of the sales agreement, will pay for it. Part of the difference between "cost" and "price" is to cover the risk associated with the specific terms and conditions included in the sales agreement, including the risks associated with the investment of capital. Further, the cost of thermal energy calculation is dependent upon certain basic assumptions. For the previously mentioned steam line, the Minnesota Public Service Commission approved an incremental cost approach for the calculation of steam cost but NSP negotiated a price with the thermal customer based upon risk and market conditions. One basic decision is if the thermal customer is going to pay a share of the rate base or cost of construction of the plant. Although NSP presently has no plans to construct additional nuclear capacity, for new plants, it may be appropriate for the thermal customer to pay for part of the plant and bear the associated risk; but, for an existing facility, the thermal customer would not pay for a share of the cost of constructing the facility. The thermal customer would pay for the incremental costs of using the facility. These costs include the cost of replacing the electricity that would have been generated by Prairie Island but now has to be generated by more expensive units. This replacement energy would be the cost of coal if the replacement energy was replaced by a coal unit or the cost of oil if the replacement energy has to be generated by an oil plant. This replacement energy cost will depend upon plant outages, new plant construction, and demand growth. Therefore, the thermal customer has two choices; pay for construction of the nuclear plant and receive energy prices based upon uranium, or not pay for construction of the plant and receive energy prices based upon oil/coal. Note that replacement energy is one of the basic indicators that utility planners use to signal the need for additional base load capacity - when the replacement energy charges exceed the revenue requirements (capital, O&M, and fuel) for a new facility, the new facility may be justified. Cogeneration reduces the electric derate for a given thermal output. Therefore, in the cost calculation, cogeneration reduces the cost of coal/oil replacement energy.

It is imperative to recognize that the incremental cost methodology used for developing the cost of thermal energy may develop a forecast of costs which are substantially lower than other methods which allocate part of the plant construction cost to the thermal customer. The assumption inherent in the forecast is that base load electric generating plants will be built by the electric utility to meet electric demand growth. The incremental pricing methodology used places substantial risk on the thermal customer. Under a scenario of electric demand growth and no additional construction, the cost of thermal energy from a nuclear plant or coal plant will approach the cost of oil. Paying for part of the construction of a new nuclear plant has a different set of risks associated with it, but once built, the energy price will be based on uranium. It is not immediately clear that a thermal customer would be willing, on a long-term basis, to take on the incremental risks of the electric utility associated with an incremental pricing methodology. Further, the prices developed should be used to indicate potential economic feasibility of this project and not to compare different thermal source options assuming different pricing methods. The potential variation of replacement energy is indicated by Figures D-8 and D-9, the January 1981 and 1982 NSP system incremental cost duration curves.

Due to the state of the art in predicting the effects of a thermal demand on future electric system costs, the costs for thermal energy should be considered indicative. The actual thermal costs will be based upon actual future electric system characteristics. Note that one of the major problems of forecasting replacement energy costs due to the need to model the interaction of a varying electric demand simultaneously with the varying thermal demand normally associated with district heating plants was avoided since the thermal demand here is for a base load plant such that the plant electric derate was assumed constant. The cost of thermal energy modeling does not explicitly include the effect of the 5 hour hot water transit time delay between the plant and the Twin Cities. This cost effect may not appear in the cost of thermal energy from Prairie Island but would appear in the total cost of thermal energy from various heat sources for supplying all of the energy requirements for the district heating system. Where the specific characteristics could not be modeled, best estimates based upon prior experience were used.

The cost of thermal energy from Prairie Island Units 1 and 2 was calculated for four conditions as follows:

| Service | Load ($MW_t$) | Electric Derate ($MW_e$) |
|---|---|---|
| 1. Firm | 600 or 1200 | 113 or 226 |
| 2. Firm | 600 or 1200 | 140 or 280 |
| 3. Oil Interruptible | 600 or 1200 | 113 or 226 |
| 4. Oil Interruptible | 600 or 1200 | 140 or 280 |

The two loads and derates correspond to one or two Prairie Island units being on line.

## D.2 Types of Thermal Service

### D.2.1 Firm Service

Firm thermal service is the continuous production and sale of thermal energy. Thermal sales would only be interrupted for NSP electric system or pool emergencies and plant forced or scheduled outages. Note that this is a slightly different concept than firm electric service. Figure D-1 indicates the thermal load duration curve and the energy to be serviced. Table D-1 indicates the thermal energy that can be produced monthly assuming an 80 percent availability. This is illustrated by Figure D-1. Note that presently NSP sometimes schedules nuclear outages in peak heating season month, and if cogeneration were instituted, the outages would have to be rescheduled, impacting the nuclear fuel cycle strategies. Table D-2 gives the thermal energy produced under a firm service agreement. This is illustrated by Figure D-2. If the plant is not interrupted for electric system emergencies, then the plant would lose its capacity value to the electric system, indicating that it would be appropriate for the thermal customer to pay a capacity charge. No capacity charge has been included.

D.2.2 Oil Interruptible Service

Under this service, thermal energy would be interrupted whenever NSP has to burn oil at oil fired peaking plants in order to meet the electric system requirements. Although NSP has its highest system peak in the summer, NSP burns more oil in the winter due to maintenance outages and due to the effects of long-term sales/purchases agreements with other utilities.

Figure D-3 indicates the unserved thermal energy due to oil interruption. Note that the cost figures are calculated for when NSP starts up its first #6 oil fired electric generating plant. Table D-3 gives the expected thermal energy that can be generated from Prairie Island on an oil interruptible basis.

D.3 Cost Calculation Procedure for Thermal Energy

In 1979, United Engineers prepared a report which described eight allocation methods in use in the United States or Europe. Only the Margen Method and the Equal Discount Method yielded cost allocations that produced electricity costs less than or equal to that which would have been incurred without cogeneration and at the same time, thermal costs which are less than those incurred by a heat only boiler (see Figure D-7).

The Margen Method is based upon the following formula:

Cost of Thermal Energy = Total Plant Costs with Cogeneration - Total Plant Costs without Cogeneration

This is equivalent to:

Cost of Thermal Energy = Increase in Total Plant Costs Due to Cogeneration

Further, the proposed Margen Method looked just at the plant and not at the electric system. If the Margen study had included system effects of cogeneration, which include capacity costs, replacement energy costs, and excess energy costs, the Margen formula would be:

Cost of Thermal Energy = Electric Costs to the Consumer with Cogeneration - Electric Costs to the Consumer without Cogeneration

OR

Cost of Thermal Energy = Increase in Plant Costs Due to Cogeneration + Increase in System Cost Due to Cogeneration

OR

Cost of Thermal Energy = Increase in Electric Costs to the Consumer Due to Cogeneration

If a "fair and equitable" division of cogeneration benefit means that the electric utility is to also benefit from cogeneration, the formula would be as follows:

Cost of Thermal Energy = Increase in Electric Costs to the Consumer + Part of the Cogeneration Savings

A difficult parameter to assess is the value of capacity. To avoid this, one approach is to charge for replacement and excess energy and avoid the capacity charge by selling only interruptible thermal energy. The increase in plant costs will be calculated directly.

The benefits of cogeneration is that less fuel is used to simultaneously produce electricity and thermal energy than if they were generated separately and the plant experiences less of an electric derate than if non-cogenerated energy was taken. It is the fuel saving that is to be divided "fairly and equitably" between the thermal and electric customer where the monthly fuel saving can be calculated as follows:

Thermal Energy Cost = Cost of Fuel Actually Used - Cost of Fuel that Would Have Been Used for Electric Generation

Thermal Energy Cost = Cost of Fuel Actually Used

Cost of Fuel per MBtu* Average Electric Output * Average Plant Heat Rate Without Cogeneration

$$\text{Fuel Cost Savings} = \frac{\text{Thermal Energy} \quad \text{Fuel Cost}}{\text{Primary System \& Steam Generator Efficiency}} - \text{Thermal Energy Cost}$$

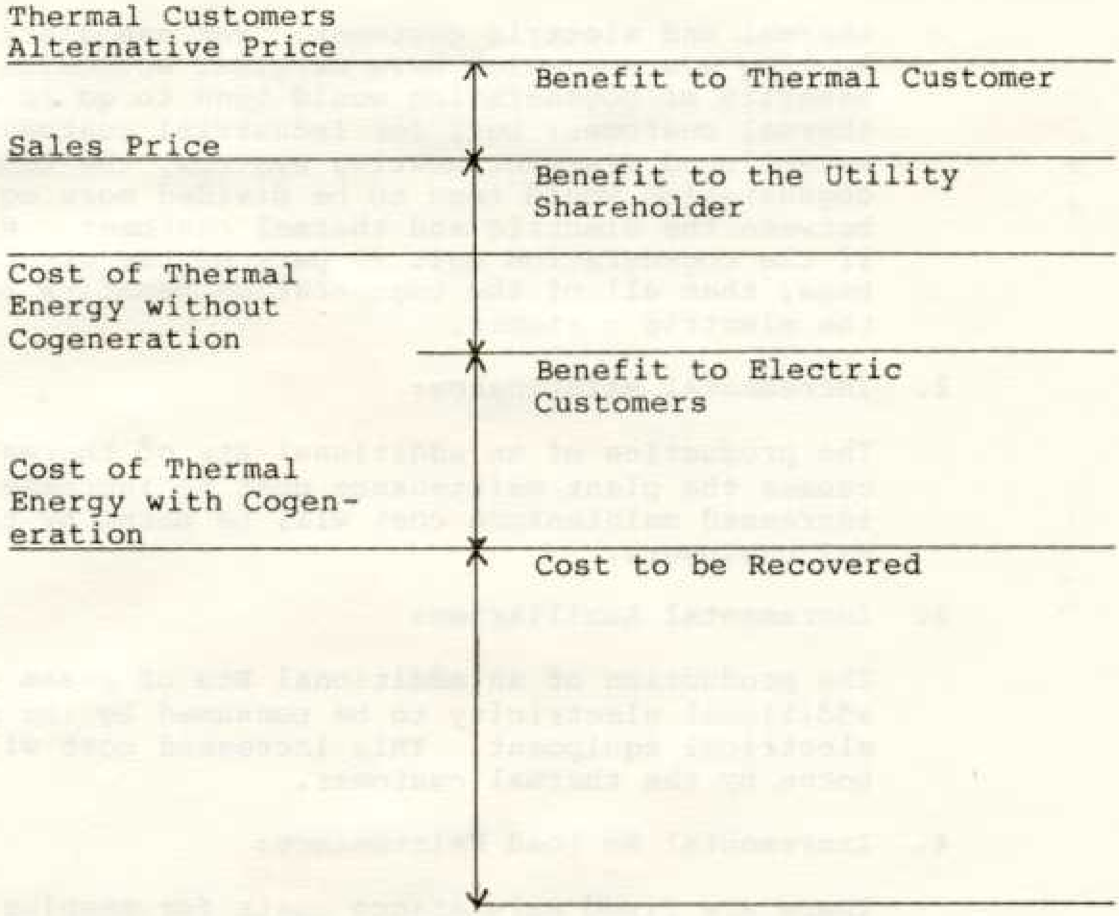
This savings is then divided between the thermal customer and the electric utility. Note that the decision over how the benefits of cogeneration are divided tends to be a secondary issue. The primary issues are:

1. Thermal Energy Seller - What are my costs and what can be reasonably charged for the service, considering costs, risk, etc.

2. Thermal Energy Purchaser - How much can be paid for the service, considering alternatives, costs, risk, etc.

Therefore, the division of the benefits of cogeneration would indirectly appear in the negotiation of the purchaser and seller, and not an analytical calculation of the cost components and who they are assigned to.

Presently, the Minnesota Public Utility Commission does not regulate the price the electric utility charges the thermal customer for thermal energy. The Commission does regulate the cost effects of the thermal sale on the electric customer. Presently, Minnesota State law requires that the electric customer not be affected negatively by cogeneration and that the benefits of cogeneration be shared between the thermal and electric customer in a fair and equitable manner as follows:




The Commission could charge the thermal customer a "use fee" for use of a facility being paid for by the thermal customer or even take part of the plant rate base and assign it to the thermal customer. This could be done at remaining undepreciated rate base of the plant or at the incremental cost of new plant construction. Hopefully, this would be offset by a replacement energy charge reduction. This indicates that the approval of the PUC will have an indirect effect on the price paid for thermal energy by the thermal customer.

## D.4 Detailed Description of Cost Components

The following are the incremental costs of selling thermal energy.

1. Incremental Fuels:

   The additional fuel required to produce thermal energy will be charged to the thermal customer. For cogenerated thermal energy, the fuel savings will be divided in a fair and equitable manner between the thermal and electric customer. For new district heating systems which have marginal economics, the benefits of cogeneration would tend to go to the thermal customer; but, for industrial customers or established district heating systems, the benefits of cogeneration would tend to be divided more equally between the electric and thermal customer. Note that if the cogeneration unit is part of the electric rate base, then all of the cogeneration benefits belong to the electric customer.

2. Incremental Maintenance:

   The production of an additional Btu of thermal energy causes the plant maintenance cost to increase. This increased maintenance cost will be borne by the thermal customers.

3. Incremental Auxiliaries:

   The production of an additional Btu of steam causes additional electricity to be consumed by the plant's electrical equipment. This increased cost will be borne by the thermal customer.

4. Incremental No Load Maintenance:

   There are fixed maintenance costs for keeping a nuclear plant in operaton. When the plant is in operation for thermal production only, these no load maintenance costs will be borne by the thermal customer. Since nuclear plants are base loaded, these costs are expected to be minimal at the nuclear plants but would be increased at other plants required to be on line for replacement energy generation.

5. Incremental No Load Auxiliaries:

   There are plant electrical requirements at zero load that are increased if the plant is ready to produce thermal or electrical energy. For those hours that the plant is in service for thermal production only, the thermal customers will pay the cost of no load auxiliaries. Since nuclear plants are base loaded, these costs are expected to be minimal at the nuclear plant but would be increased at other plants required to be on line for replacement energy generation.

6. Spent Fuel Disposal:

These are costs associated with the disposal of spent fuel, etc. The cost associated with increased spent fuel being produced shall be borne by the thermal customer. These costs are included as part of the fuel costs.

7. Replacement Energy:

   When an electric generating plant is used to produce thermal energy, it cannot generate as much electricity. If the electric system requirements indicate that the plant should be used to generate electricity, either the thermal production has to be interrupted or the electricity has to be generated by a less economical unit. The increased cost in electricity production costs will be borne by the thermal customer. The cost of coal ash disposal if replacement energy were produced by a coal fired plant would also have to be included as part of the thermal energy cost. Note that replacement energy costs are calculated after all electric purchases and sales, including new long-term contracts with other utilities. This reflects the thermal customers paying the true energy value of existing facilities.

8. Excess Energy:

   For a cogeneration plant, the electric system requirements may indicate that the cogeneration plant be taken off line or at a lower capacity because electricity may be produced by more economical units. If the thermal customer desires, the cogeneration plant may remain on line, but the thermal customer has to pay the differential costs due production of more electricity from a less economical unit. Since nuclear units are base loaded, these costs are expected to be minimal.

9. Cold Start Credit:

   There are costs associated with starting up a boiler from the cold condition. If the continuous requirements of the thermal customer causes less cold boiler starts to be incurred, the thermal customer will receive a credit for these costs being avoided. Since nuclear plants are base loaded, these costs are expected to be minimal.

10. Startup Costs:

    If the thermal customer causes a plant to be started up from cold to hot, the thermal customer will be charged the startup costs. These startup costs can be associated with a plant used to produce thermal energy or a plant used to produce replacement energy.

11. Incremental Operating:

    If the requirements of the thermal customers are such that additional plant operators are required, the cost of this additional labor will be borne by the thermal customers. These costs are expected to be minimal for nuclear cogeneration.

12. Thermal Equipment Operating & Maintenance Expense:

    To provide thermal energy, additional equipment associated only with the production of thermal energy will require operation and maintenance. The costs associated with this operation and maintenance will be charged to the thermal customer. This would also include the cost of makeup water for the thermal system.

13. Revenue Requirements for Thermal Investments:

    Since the thermal utility will be maintained separately from the electric utility, there is no electric utility revenue requirement for the thermal production equipment investments. These revenue requirements are to be paid by the thermal customer, but the revenues are expected to be separate from electric revenue requirements.

    Since the incremental cost approach is proposed, there are no revenue requirements to be paid by the thermal customer to the electric utility for existing or future electricity production equipment. For example, in the future, if additional plant equipment is required to keep plant acceptable for electric production, the electric customer will bear those costs. This assumes that it is the economic choice for the electric customer to have such additional equipment installed and maintain the plant as a viable electricity production facility.

    Note that the value of a generating plant is derived from its energy generating capability and its capacity. Any changes in energy generation costs are recaptured through replacement energy. Since the thermal customer will be interrupted when the generating unit is required for electric system emergencies, the capacity value of the plant has not decreased and for some older coal fired plants, it may be increased.

14. Property Taxes, Sales Taxes and Franchise Fees:

    Any additional property taxes, sales taxes or franchise fees due to thermal sales shall be borne by the thermal customer. These will not be included in the electric utility expenses and, therefore, have no effect on electric utility revenue requirements and must be borne by the thermal customer.

15. Standby No Loads:

    During winter operation, more than one plant may be required to be in service for thermal energy supply in order to provide reliability. The costs associated with maintaining a boiler in hot standby condition for backup or standby service shall be paid by the thermal customer. For this study, these costs are minimal.

16. Reduced Sales for Resale:

    Due to thermal production reducing electricity generation potential and sales for resale, the thermal customer will reimburse the electric customer for this opportunity cost. Sales for resale are electric sales to other utilities.

17. Fuel Carrying Charge:

    Due to the long lead time for nuclear fuel, there are interest charges associated with nuclear fuel prior to the fuel being used in the reactor. These costs are included in the fuel charge.

18. Working Capital:

    The thermal utility will have to provide its own working capital, the carrying charge on which is to be borne by the thermal customer. Working capital will be handled directly by thermal utility.

19. A & G:

    The administrative and general costs associated with the sale of thermal energy will be paid by the thermal customer. These costs could also include allowance for error margin and undetermined expenses.

20. Insurance:

    The cost associated with insuring equipment associated with the production of only thermal energy shall be borne by the thermal customer and handled directly by thermal utility.

21. Service Charge:

    Charge to cover risk and profit for thermal utility investment, etc. Service charge will be handled directly by the thermal utility. The amount of profit will depend upon market conditions for alternative energy sources, risk, investment and other terms and conditions of the sales agreement. This charge is the difference between cost and price.

22. Decommissioning:

    Since the nuclear plant has an expected life and that life is dependent upon the energy produced, the decommissioning costs shall be prorated between the thermal customer and electric customer.

23. Electric Pool Effects:

    The power pool that NSP is a member of gives NSP a credit for the electric generating capacity of Prairie Island. As long as the capacity is available for system emergencies, MAPP should allow the NSP capacity value. NSP may have to prove its generating capacity by a 100 percent electric load test for annual FERC or pool capacity credits. No allowance will be made for this potential cost effect.

24. Electric System Planning Effects:

    The ability to produce electric power in a given area is factored into the design of the electric transmission system. Power plants in a given area may reduce the transmission requirements for power delivery from other parts of the system to that area. If the plant is no longer available to produce as much electricity, the transmission to the area of the plant may have to be upgraded. This effect is more likely with plants located in metropolitan areas. This effect is mitigated since the plant will always be available for system emergencies. The increased short-term availablity of the units due to the unit either being hot or on line more of the time may also increase local transmission system reliability. No allowance has been made for this potential effect.

25. Seasonality and Fuel Cycles:

Maintenance and refueling outages for large units are scheduled to produce the lowest annual electric system generation cost. Outage schedules are planned in advance with many constraints dictating when certain units can come off line. With the new constraint of no nuclear outages in the peak thermal load winter months, the annual electric system generation costs may be higher. Requiring nuclear units to be avalable in winter may have some significant costs associated with fuel cycle strategies. These costs have not been included. Note that the cost of thermal energy is increased when any large base load unit, coal or nuclear, is out of service in the winter months. The outages are scheduled to minimize electric system costs. No allowance will be made for this potential cost.

26. Reduced Thermal Discharge to the Environment:

The economic benefit of reduced thermal discharge appears in the lessening of negative environmental impacts and less use of the cooling water system resulting in less electric consumption by cooling water pumps and cooling towers and less O&M on that equipment. These cost effects will not be included.

27. Prior Thermal Customers:

Any prior thermal customer, cogenerating or non-cogenerating, would derate other plants. This would increase the cost of replacement energy to later customers who signed contracts for thermal energy. These cost effects have not been included.

28. Public Utility Commission:

The Public Utility Commission may decide that the electric customer should receive more than incremental costs. The Minnesota Public Utility Commission recently decided that a 0-5 percent adder on nonfuel, noncapital thermal energy costs should be credited to the rate payer.

29. Thermal Energy Plant Modifications:

When an existing plant is upgraded to serve thermal energy requirements, significant plant investments may be made for the thermal customer that improve the plant availability and efficiency also for the electric customer. These investments are justified for the thermal customer but not for the electric customer, although the electric customer benefits and does not contribute to the investment.

D.5 Replacement Energy

The most significant cost components of the thermal energy costs is the estimate of the replacement energy costs due to reduced generation at Prairie Island because of steam sales from the units.

The main tool used in preparing these estimates was NSP's PECOS production simulation model. This model uses probability analysis to optimally simulate the operation of generating units on the NSP system.

The production cost solution of PECOS uses an equivalent load duration curve and the available generator data to provide probable generator usage data for the system. The program also estimates the overall cost of energy production over a given time period.

From the program output, it is possible to determine the cost penalty associated with replacing any generation lost by reducing the output of the units.

This penalty is then multiplied by the hours of reduced generation and the $MW_e$ derate caused by the steam sale, to provide the total cost penalty associated with the reduced generation.

One of the problems with the PECOS production model is that it solved on a monthly, quarter monthly, or annual basis making virtually impossible to exactly model the effects of load reductions on various units. However, it is NSP's belief that the cost penalties derived are accurate and the best estimates that can be provided until the ability to model actual effects of steam sales from a given generation unit becomes available.

Table D-4 gives the replacement energy costs for Prairie Island. Figure D-5 shows a sample annual electric system incremental cost duration curve with the Prairie Island cost indicated and its relationship to the replacement energy charge. NSP also has large coal fired units (Sherco plant) which are also base loaded. Figure D-6 shows their relationship to the Prairie Island costs. Note that if these coal fired units were converted to cogeneration and they had the same derate ($MW_e/MW_t$) characteristics, the cost of replacement energy would be the same. The variability of the heat rate due to the thermal and electrical load match is shown in Figure D-4. Therefore, to minimize the cost of thermal energy, the important variables are:

1. The $MW_e/MW_t$ derate characteristics irrespective of coal or nuclear

2. Proximity of plant and user

3. Base load or peaking thermal and use of plant

Note that for non-base load electric generating units, the thermal customer would avoid most replacement energy charges, but would be responsible for more of the no load costs of operation. Further, if the electric customer did not receive part of the benefits of cogeneration, the incremental electric generation cost would remain the same and the units' order of economic dispatch would remain unchanged. Therefore, large amounts of excess energy would be generated, the incremental cost of which the thermal customer would be responsible for.

District heating systems have predominately winter loads. Therefore, the derate on the plant would occur during the winter where the replacement energy is substantially higher. Therefore, for the type systems, the price and cost of thermal energy would be substantially higher. The price and cost of thermal energy increases with any increase in the time-wise correlation of thermal demand and replacement energy cost. This has been the general experience NSP has had in estimating the cost of thermal energy from small district heating systems with little industrial load. Note that the calculation of replacement energy for actual cost calculations would require a more detailed evaluation.

## D.6 Detailed Cost Calculation

Since the Prairie Island units are base loaded, the thermal energy cost calculations are greatly simplified. For a base load unit, the thermal energy production costs can be estimated by the following formula:

$$\text{Thermal Energy Production Cost } (\$/MWH_t) = \text{Replacement Energy Cost } (\$/MWH_e) * \frac{MW_e \text{ Derate}}{MW_t \text{ Demand}}$$

This thermal energy production cost estimate includes the following cost components:

1. Incremental Fuels
2. Incremental Maintenance
3. Incremental Auxiliaries
4. No Load Maintenance
5. No Load Auxiliaries
6. Spent Fuel Disposal
7. Replacement Energy
8. Excess Energy

These production costs are all included in the replacement energy cost calculation. When a unit is not base loaded, the calculation would be a direct calculated estimate of cost components 1-8 with the benefits of cogeneration appearing as a lower incremental fuel cost. For Prairie Island, the production cost of thermal energy is not the Prairie Island production cost, but the cost of replacing the Prairie Island derate by other units. (Cost of Thermal Energy = Prairie Island Production Cost + Replacement Energy Cost) Therefore, the cost of thermal energy will depend primarily on the NSP system production/electric demand characteristics and secondarily on the price of uranium.

To obtain a price of thermal energy for estimating purposes and based on prior experience, it was assumed that the remainder of the cost components, except capital costs and decommissioning costs, would be 20 percent of the production cost. Under this assumption, the thermal customer would have no guarantee as to what the future incremental cost of thermal energy would be. The total price of thermal energy is shown on Table D-5. Note that the costing methodology has given all of the benefits to the thermal customer, which may not necessarily be the case. The electric customer is paying for the plant capital costs and the thermal customer is paying only for incremental production costs. For making the initial significant capital investment, it would be appropriate for the electric utility to receive some of the benefits of cogeneration.

The risk for the thermal customer previously discussed is the potentially volatile nature of replacement energy costs. Figures D-8 and D-9 show NSP's incremental cost duration curves for January 1981 and January 1982, indicating the potential wide variation in incremental costs.

In 1982, the revenue requirements for Prairie Island decommissioning are \$5.2 million. Assuming an average annual plant capacity of 1,020 $MW_e$, the decommissioning costs to be paid by the thermal customer can be approximated as follows:

Assumptions:

| | |
|---|---|
| Revenue Requirements | \$5.2 million |
| Plant Derate Capacity | 1,020 $MW_e$ |
| Plant Derate Due to Thermal Sales | 2 * 140 $MW_e$ |
| Plant Annual Thermal Output | 153,216 MWD |

Calculation: 140 $MW_e$ derate/unit

$$\$5.2 * 10^6 * \frac{2 * 140\ MW_e}{1{,}020\ MWe} * \frac{1}{153{,}216\ MWD * 24H} = \frac{\$0.39}{MWH_t}$$

1982 Decommissioning Cost (\$/$MWH_t$):

| | Derate 113 $MW_e$/Unit | 140 $MW_e$/Unit |
|---|---|---|
| Oil Interruptible | 0.31 | 0.39 |
| Firm | 0.27 | 0.33 |

It takes approximately 5 hours for the water leaving Prairie Island to reach the Twin Cities. Note that this implies that when the thermal peak during the day is served, the energy to serve it is produced at 5 hours earlier, i.e., 5 a.m. to 9 a.m., when the electric system demand is low. If the Prairie Island units were essentially base loaded for both thermal and electric purposes, there would be no economic effect on the cost of thermal energy due to the transit time. Under the 1,200 $MW_t$ scenario, one of the units would see sigificant variation during the year but on a seasonal basis than on a daily basis. This seasonal effect would tend to mitigate the transit time benefit.

During the spring and fall of the year, there are occasions when the electric system demand is low and base load units are required to operate at partial load. Depending on the specific coal fired units and their ability to load follow and back down to low loads, it may be necessary to reduce the load on Prairie Island. This partial load operation may also be required by safety, technical specifications, operational constraints, or license restrictions. The effects of cogeneration previously calculated assumed base load operation but the thermal efficiency (heat rate) of the thermal cycle varies with both thermal and electric load (see Figure D-4). The effect of this shows up in varying thermal energy fuel costs.

Further, if one assumes that the electricity produced during the derated condition (i.e. output of 600 $MW_t$)

would have been otherwise produced at the $OMW_t$ heat rate, the difference between 1,650 $MW_t$ and 400 $MW_e$ or 460 $MW_t$ times the full power heat rate is the portion energy input to the turbines that is charged to the thermal customer. With the 160 $MW_e$ derate, the thermal energy cost to the thermal customer is approximately 67 percent of the cost of steam coming from the reactor and with a 120 $MW_e$ derate, the thermal energy is approximately 47 percent of the cost of steam coming from the reactor. Note that these costs include plant costs only. Note also that if cogeneration reduces the cost of nuclear steam 33 percent to 53 percent, cogeneration reduces the cost of replacement energy 33 to 53 percent. Another view of this is that in 1982 dollars, cogeneration saves only about $\frac{\$0.90}{MWH_t}$ to $\frac{\$1.44}{MWH_t}$ in production costs at Prairie Island, but in replacement energy costs, cogeneration saves $\frac{\$3.73}{MWH_t}$ to $\frac{\$5.98}{MWH_t}$ for firm thermal service and $\frac{\$2.59}{MWH_t}$ to $\frac{\$3.93}{MWH_t}$ for oil interruptible thermal service.

TABLE D-1

DISTRICT HEATING SYSTEM

TOTAL ENERGY REQUIREMENT

| Month | Average Demand $MW_t$ | Average Energy $MWD_t$ |
|---|---|---|
| 1 | 1,560 | 47,424 |
| 2 | 1,560 | 47,424 |
| 3 | 1,332 | 40,492 |
| 4 | 1,070 | 32,528 |
| 5 | 800 | 24,320 |
| 6 | 600 | 18,240 |
| 7 | 400 | 12,160 |
| 8 | 340 | 10,336 |
| 9 | 230 | 6,992 |
| 10 | 170 | 5,168 |
| 11 | 130 | 3,952 |
| 12 | 70 | 2,128 |
| | | 251,164 MWD |

TABLE D-2

FIRM SERVICE ENERGY SUPPLY

| | (2 Units) | | (1 Unit) | |
|---|---|---|---|---|
| Month | Average Demand $MW_t$ | Average Energy $MWD_t$ | Average Demand $MW_t$ | Average Energy $MWD_t$ |
| 1 | 1,200 | 36,500 | 600 | 18,250 |
| 2 | 1,200 | 36,500 | 600 | 18,250 |
| 3 | 1,200 | 36,500 | 600 | 18,250 |
| 4 | 1,070 | 32,528 | 600 | 18,250 |
| 5 | 800 | 24,320 | 600 | 18,250 |
| 6 | 600 | 18,240 | 600 | 18,240 |
| 7 | 400 | 12,160 | 400 | 12,160 |
| 8 | 340 | 10,336 | 340 | 10,336 |
| 9 | 230 | 6,992 | 230 | 6,992 |
| 10 | 170 | 5,162 | 170 | 5,162 |
| 11 | 130 | 3,952 | 130 | 3,952 |
| 12 | 70 | 2,128 | 70 | 2,128 |
| | Total Energy | 225,318 | Total Energy | 150,130 |

| | | |
|---|---|---|
| Total Energy Supplied by Cogeneration* (Assuming 80% Availability) | 180,254 $MWD_t$ | 120,104 |
| Energy Supplied By Heat Only Boilers | 70,910 $MWD_t$ | 105,214 |
| Total District Heating* | 251,164 $MWD_t$ | 251,164 |

System Thermal Requirement

* To be adjusted by PI Twin City line losses/heat gains

TABLE D-3

OIL INTERRUPTIBLE ENERGY SERVICE

(2 Units)

| | |
|---|---|
| Total Energy Supplied by Cogeneration (Firm) | 180,254 $MWD_t$ |
| $1 - \frac{\text{Avg. Hours Int.}}{8{,}760} * 1.15*$ | .85 |
| Total Energy Supplied** by Cogeneration (Oil Interruptible) | 153,216 $MWD_t$ |
| Energy to be Supplied by Heat Only Boilers | 97,948 $MWD_t$ |
| Total District Heating System Energy Requirements** | 251,154 $MWD_t$ |

* Estimate of coincidence factor for thermal and electric system peak demands

** To be adjusted by PI Twin City line losses/heat gain

TABLE D-3 (continued)

OIL INTERRUPTIBLE ENERGY SERVICE

(1 Unit)

| | |
|---|---|
| Total Energy Supplied by Cogeneration (Firm Service) | 120,104 $MWD_t$ |
| 1 - $\frac{\text{Avg. Hours Int.}}{8,760}$ * 1.15* | .85 |
| Total Energy Supplied by Cogeneration (Oil Interruptible)** | 102,088 $MWD_t$ |
| Total Energy Supplied by Heat Only Boilers | 149,065 $MWD_t$ |
| Total District Heating System Energy Requirements** | 251,154 $MWD_t$ |

*Estimate of coincidence factor for thermal and electric system peak demands

** To be adjusted by PI Twin City line losses

TABLE D-4

Prairie Island Steam Analysis
Replacement Energy Values

(Future Dollars)

(No Capacity Costs Included)

| | Replacement Energy Value ($/MWh$_e$) | Oil Interruptible Replacement Energy Value ($/MWh$_e$) | Prairie Island Cost ($/MWh$_e$) |
|---|---|---|---|
| 1992 | 75.3 | 52.5 | 22.8 |
| 1993 | 105.1 | 59.0 | 24.3 |
| 1994 | 122.0 | 65.8 | 25.9 |
| 1995 | 127.8 | 68.3 | 27.6 |
| 1996 | 144.5 | 75.1 | 29.5 |
| 1997 | 158.6 | 82.2 | 31.4 |

| | Hours of Operation Prairie Island | | Hours of Operation* Pathfinder |
|---|---|---|---|
| 1992 | 6925 | | 673 |
| 1993 | 6898 | | 1147 |
| 1994 | 6898 | | 1266 |
| 1995 | 6898 | | 1235 |
| 1996 | 6920 | | 1335 |
| 1997 | 6898 | | 1289 |
| | | Average | 1158 |

* Hours of Pathfinder = Hours of oil interruption
Pathfinder is the first #6 oil
Peaking unit dispatched for electric service

TABLE D-5

PRICE OF THERMAL ENERGY

TOTAL PRICE - FUTURE DOLLARS

| | FIRM SERVICE | | OIL INTERRUPTIBLE SERVICE | |
|---|---|---|---|---|
| | 140 $MW_e$ derate $/$MWH_t$ | 113 $MW_e$ derate $/$MWH_t$ | 140 $MW_e$ derate $/$MWH_t$ | 113 $MW_e$ derate $/$MWH_t$ |
| 1992 | 21.41 | 17.29 | 15.09 | 12.18 |
| 1993 | 29.75 | 24.02 | 16.90 | 13.64 |
| 1994 | 34.48 | 27.85 | 18.81 | 15.18 |
| 1995 | 36.11 | 29.15 | 19.52 | 15.74 |
| 1996 | 40.79 | 33.52 | 21.41 | 17.28 |
| 1997 | 44.74 | 36.11 | 23.41 | 18.89 |

TABLE D-6

PRICE OF THERMAL ENERGY

TOTAL PRICE - 1982 DOLLARS

| | FIRM SERVICE | | OIL INTERRUPTIBLE SERVICE | |
|---|---|---|---|---|
| | 140 $MW_e$ derate $/$MWH_t$ | 113 $MW_e$ derate $/$MWH_t$ | 140 $MW_e$ derate $/$MWH_t$ | 113 $MW_e$ derate $/$MWH_t$ |
| 1992 | 9.65 | 7.79 | 6.89 | 5.57 |
| 1993 | 12.46 | 9.97 | 7.13 | 5.76 |
| 1994 | 13.17 | 10.64 | 7.31 | 5.90 |
| 1995 | 12.75 | 10.30 | 7.03 | 5.67 |
| 1996 | 13.25 | 10.70 | 7.11 | 5.72 |
| 1997 | 13.39 | 10.82 | 7.16 | 5.77 |

TABLE D-7

COST OF THERMAL ENERGY

(PRODUCTION COSTS - FUTURE DOLLARS)

| | FIRM SERVICE | | OIL INTERRUPTIBLE SERVICE | |
|---|---|---|---|---|
| | 140 $MW_e$ derate $\$/MWH_t$ | 113 $MW_e$ derate $\$/MWH_t$ | 140 $MW_e$ derate $\$/MWH_t$ | 113 $MW_e$ derate $\$/MWH_t$ |
| 1992 | 17.57 | 14.18 | 12.25 | 9.89 |
| 1993 | 24.52 | 19.79 | 13.76 | 11.11 |
| 1994 | 28.46 | 22.98 | 15.35 | 12.39 |
| 1995 | 29.82 | 24.07 | 15.94 | 12.86 |
| 1996 | 33.72 | 27.21 | 17.52 | 14.14 |
| 1997 | 37.01 | 29.87 | 19.18 | 15.48 |

TABLE D-8

COST OF THERMAL ENERGY

(PRODUCTION COSTS - 1982 DOLLARS)

| | FIRM SERVICE | | OIL INTERRUPTIBLE SERVICE | |
|---|---|---|---|---|
| | 140 $MW_e$ derate $\$/MWH_t$ | 113 $MW_e$ derate $\$/MWH_t$ | 140 $MW_e$ derate $\$/MWH_t$ | 113 $MW_e$ derate $\$/MWH_t$ |
| 1992 | 7.77 | 6.27 | 5.42 | 4.38 |
| 1993 | 10.11 | 8.08 | 5.62 | 4.54 |
| 1994 | 10.70 | 8.64 | 5.77 | 4.66 |
| 1995 | 10.35 | 8.36 | 5.53 | 4.47 |
| 1996 | 10.77 | 8.69 | 5.60 | 4.51 |
| 1997 | 10.88 | 8.79 | 5.64 | 4.55 |

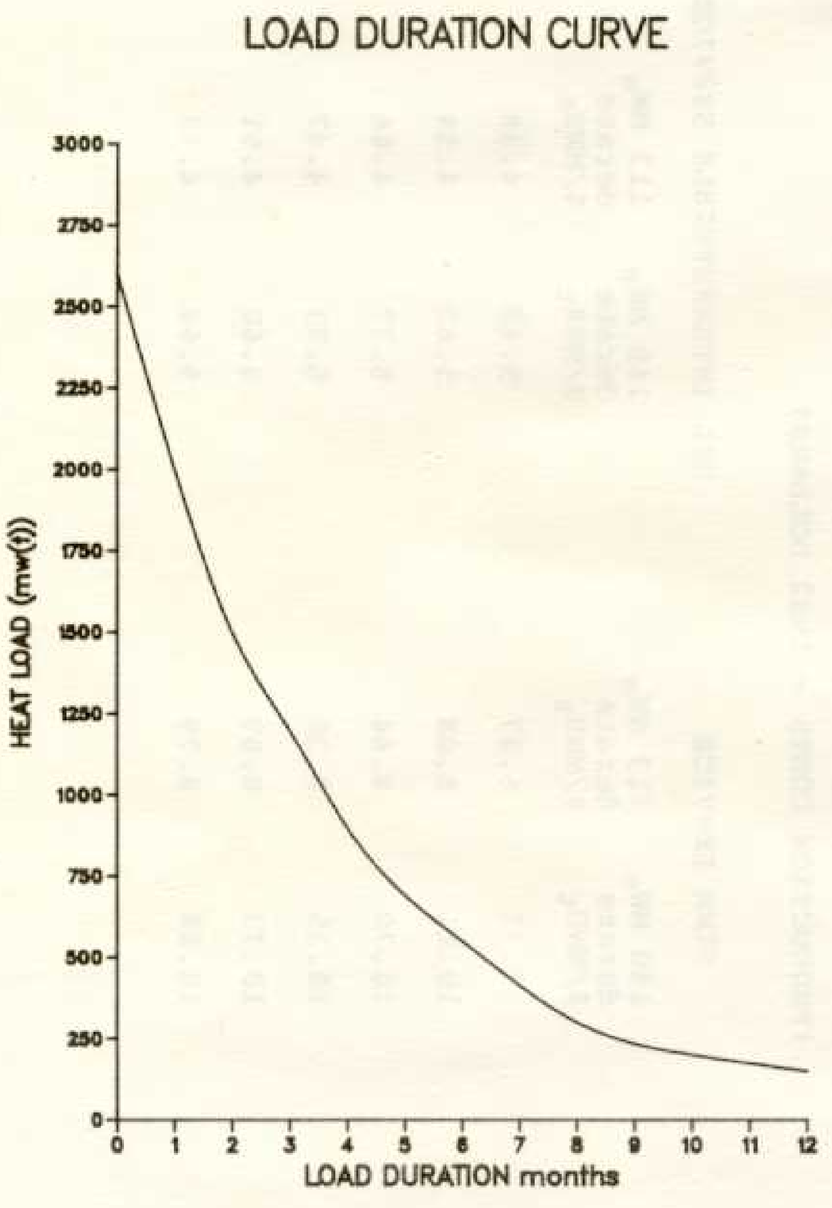

LOAD DURATION CURVE
HEAT LOAD (mw(t))
3000
2750
2500
2250
2000
1750
1500
1250
1000
750
500
250
0
0
1
2
3
4
5
6
7
8
9
10
11
12
LOAD DURATION months

FIG D-1

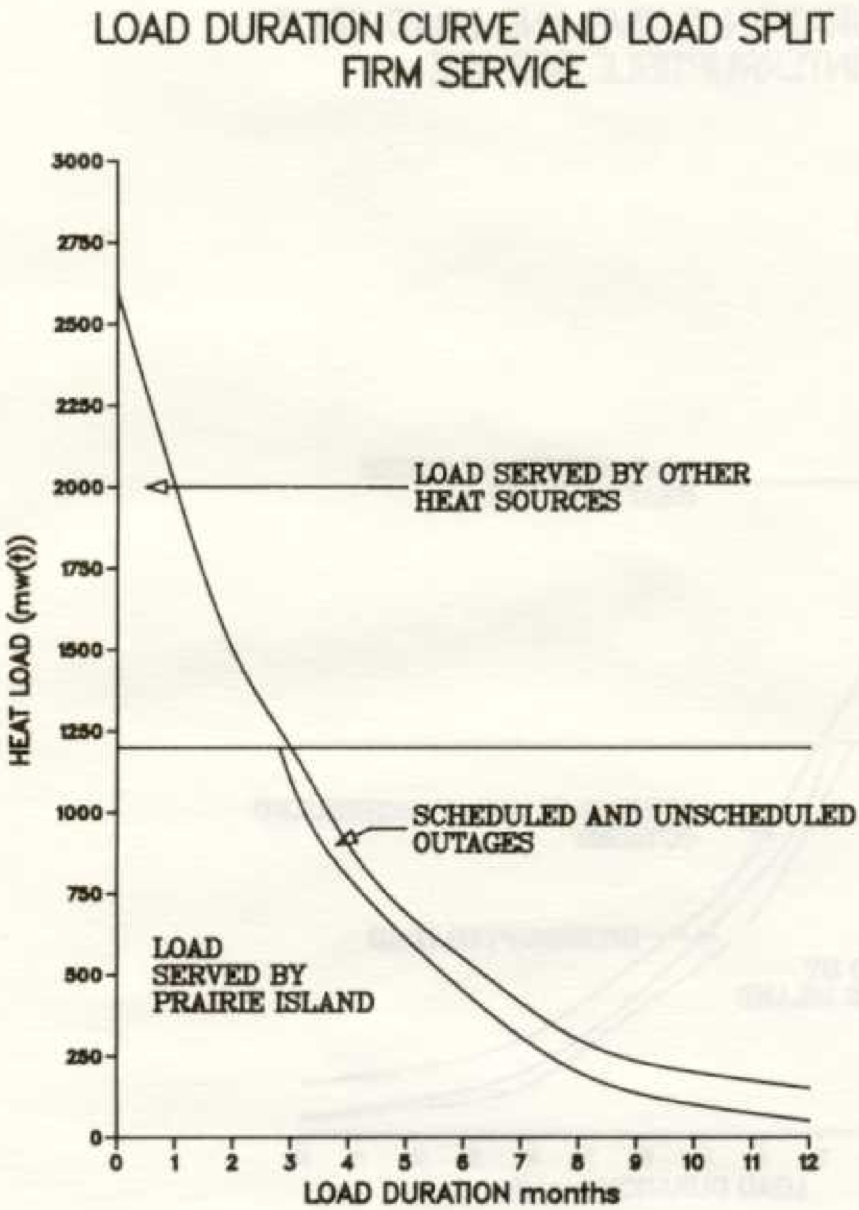

LOAD DURATION CURVE AND LOAD SPLIT
FIRM SERVICE
HEAT LOAD (mw(t))
3000
2750
2500
2250
2000
1750
1500
1250
1000
750
500
250
0
0
1
2
3
4
5
6
7
8
9
10
11
12
LOAD DURATION months
LOAD SERVED BY OTHER HEAT SOURCES
SCHEDULED AND UNSCHEDULED OUTAGES
LOAD SERVED BY PRAIRIE ISLAND

FIG D-2

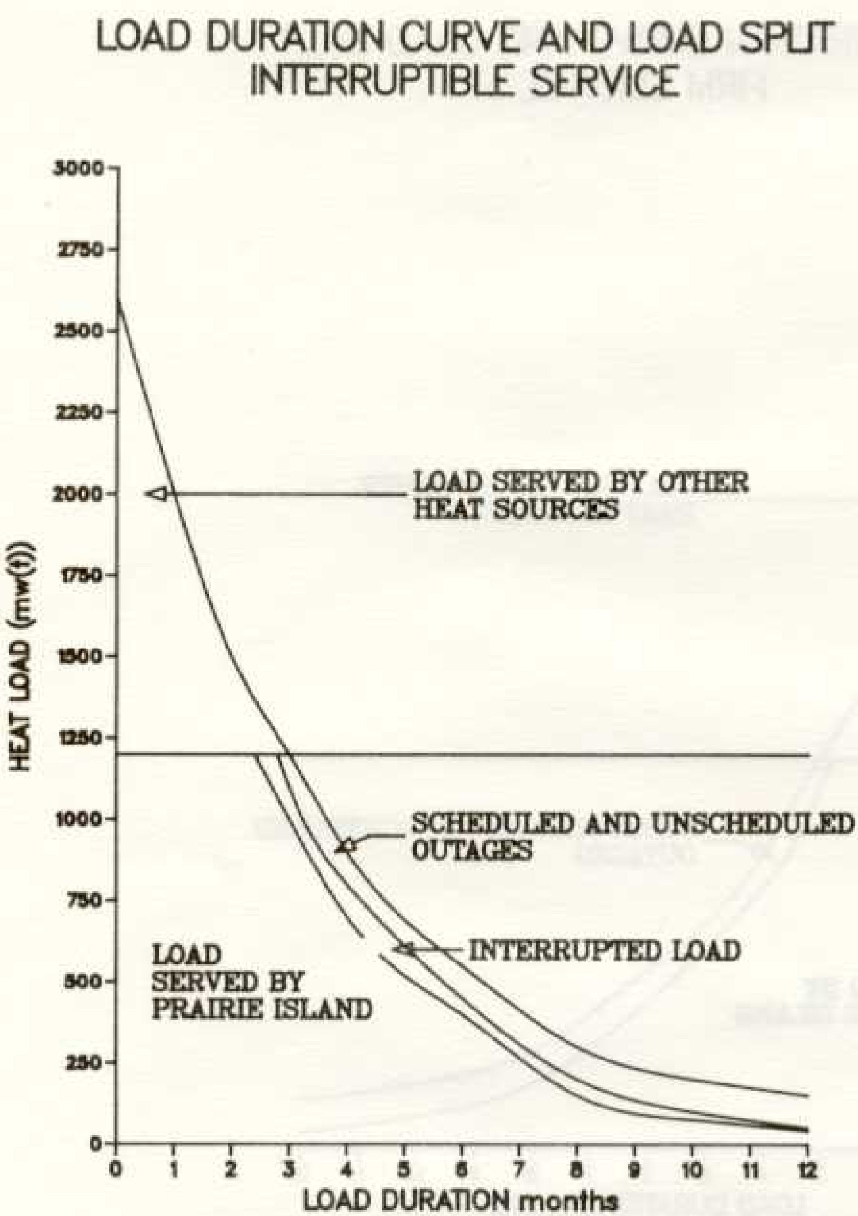

LOAD DURATION CURVE AND LOAD SPLIT
INTERRUPTIBLE SERVICE
HEAT LOAD (mw(t))
3000
2750
2500
2250
2000
1750
1500
1250
1000
750
500
250
0
LOAD SERVED BY OTHER HEAT SOURCES
SCHEDULED AND UNSCHEDULED OUTAGES
INTERRUPTED LOAD
LOAD SERVED BY PRAIRIE ISLAND
0
1
2
3
4
5
6
7
8
9
10
11
12
LOAD DURATION months

FIG D-3

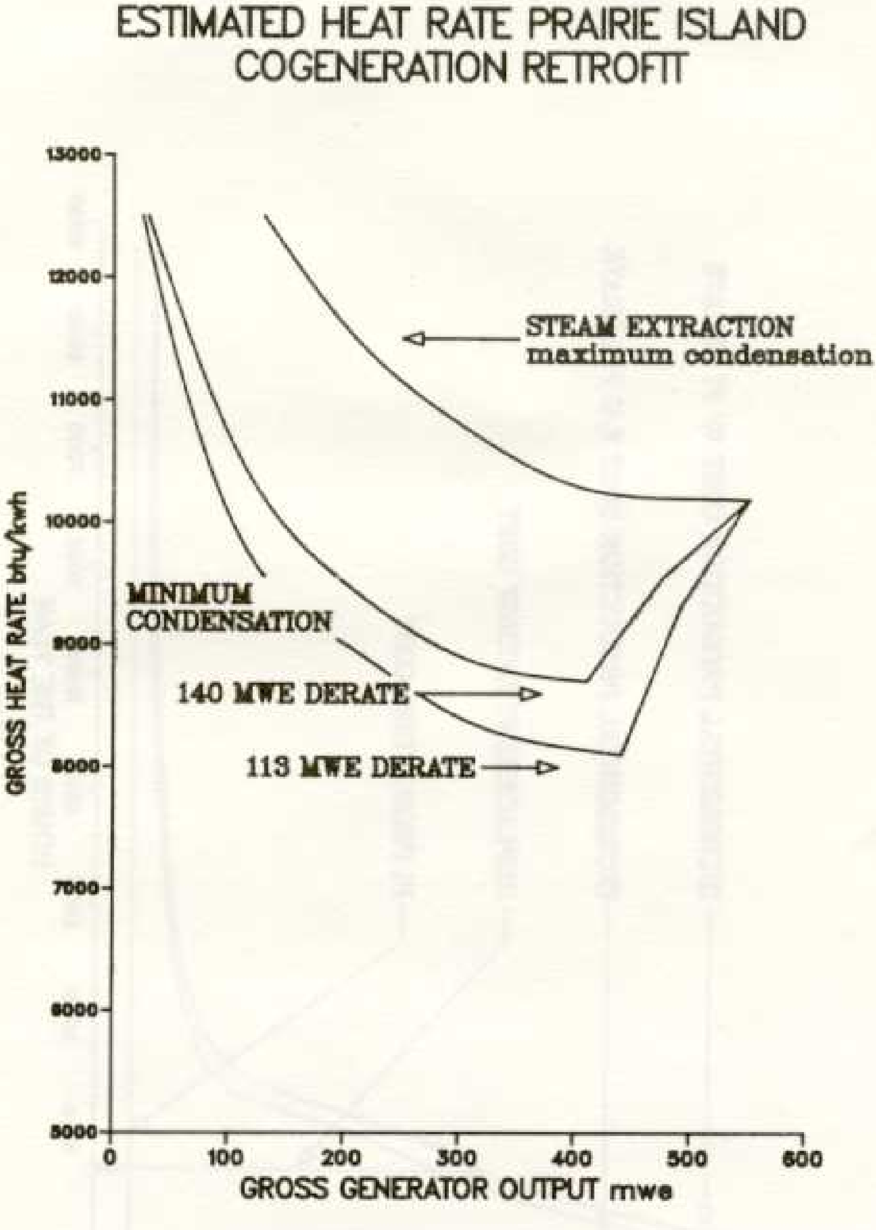

ESTIMATED HEAT RATE PRAIRIE ISLAND
COGENERATION RETROFIT
13000
12000
11000
10000
9000
8000
7000
6000
5000
GROSS HEAT RATE btu/kwh
0
100
200
300
400
500
600
GROSS GENERATOR OUTPUT mwe
STEAM EXTRACTION
maximum condensation
MINIMUM
CONDENSATION
140 MWE DERATE
113 MWE DERATE

FIG D-4

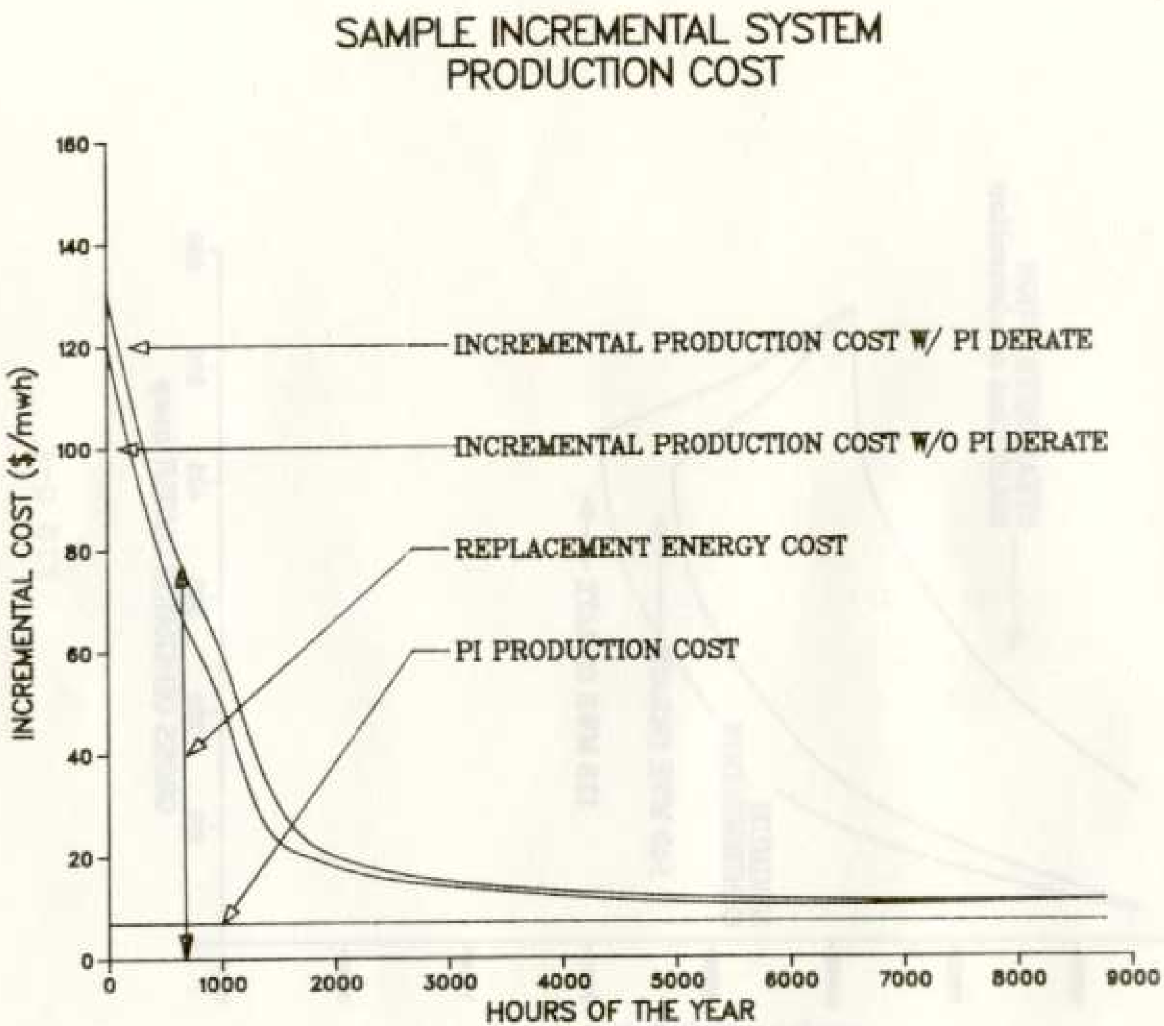

SAMPLE INCREMENTAL SYSTEM
PRODUCTION COST
INCREMENTAL PRODUCTION COST W/ PI DERATE
INCREMENTAL PRODUCTION COST W/O PI DERATE
REPLACEMENT ENERGY COST
PI PRODUCTION COST
INCREMENTAL COST ($/mwh)
HOURS OF THE YEAR
0
20
40
60
80
100
120
140
160
1000
2000
3000
4000
5000
6000
7000
8000
9000

FIG D-5

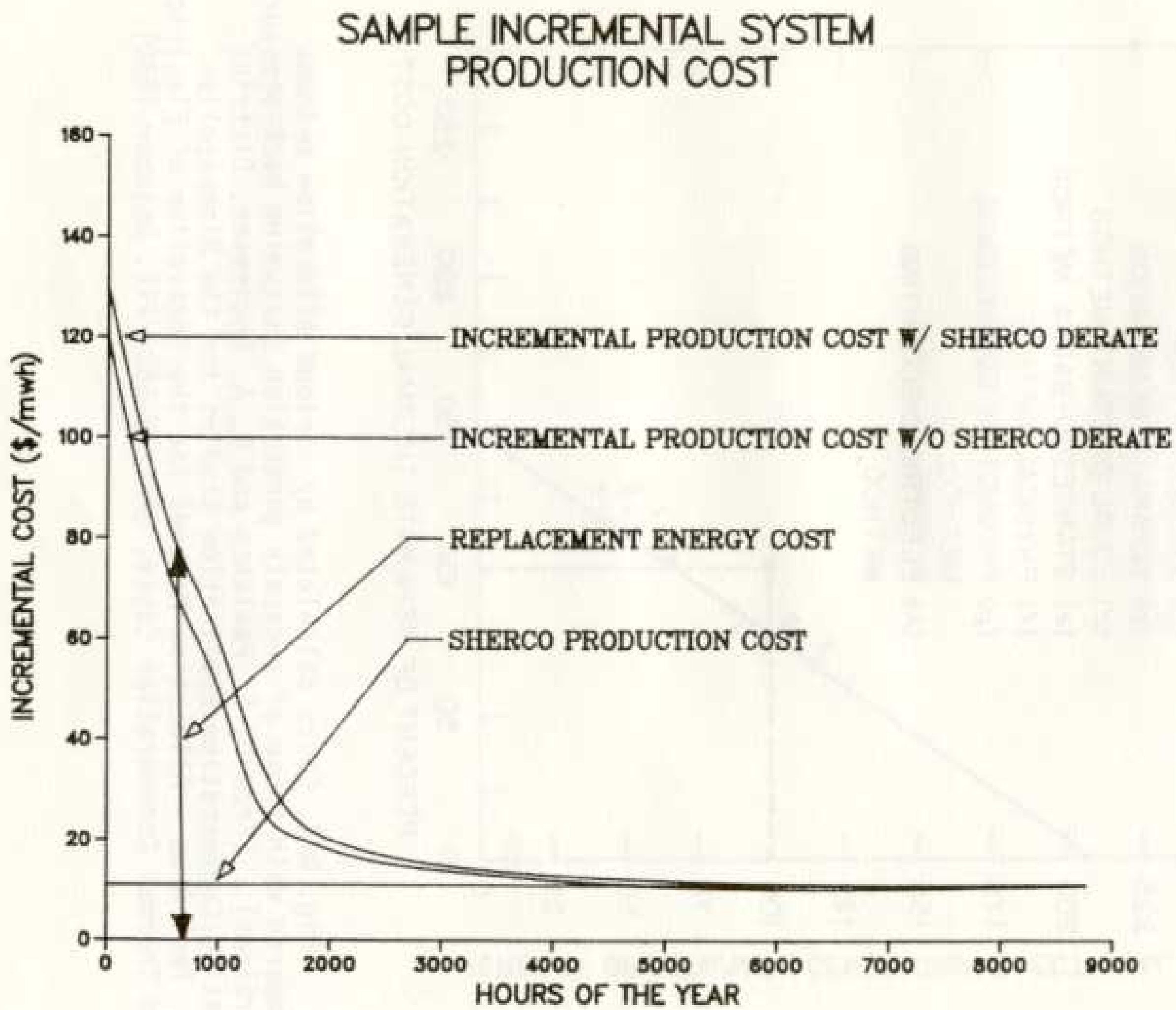

SAMPLE INCREMENTAL SYSTEM
PRODUCTION COST
INCREMENTAL PRODUCTION COST W/ SHERCO DERATE
INCREMENTAL PRODUCTION COST W/O SHERCO DERATE
REPLACEMENT ENERGY COST
SHERCO PRODUCTION COST
INCREMENTAL COST ($/mwh)
HOURS OF THE YEAR
0
20
40
60
80
100
120
140
160
0
1000
2000
3000
4000
5000
6000
7000
8000
9000

FIG D-6

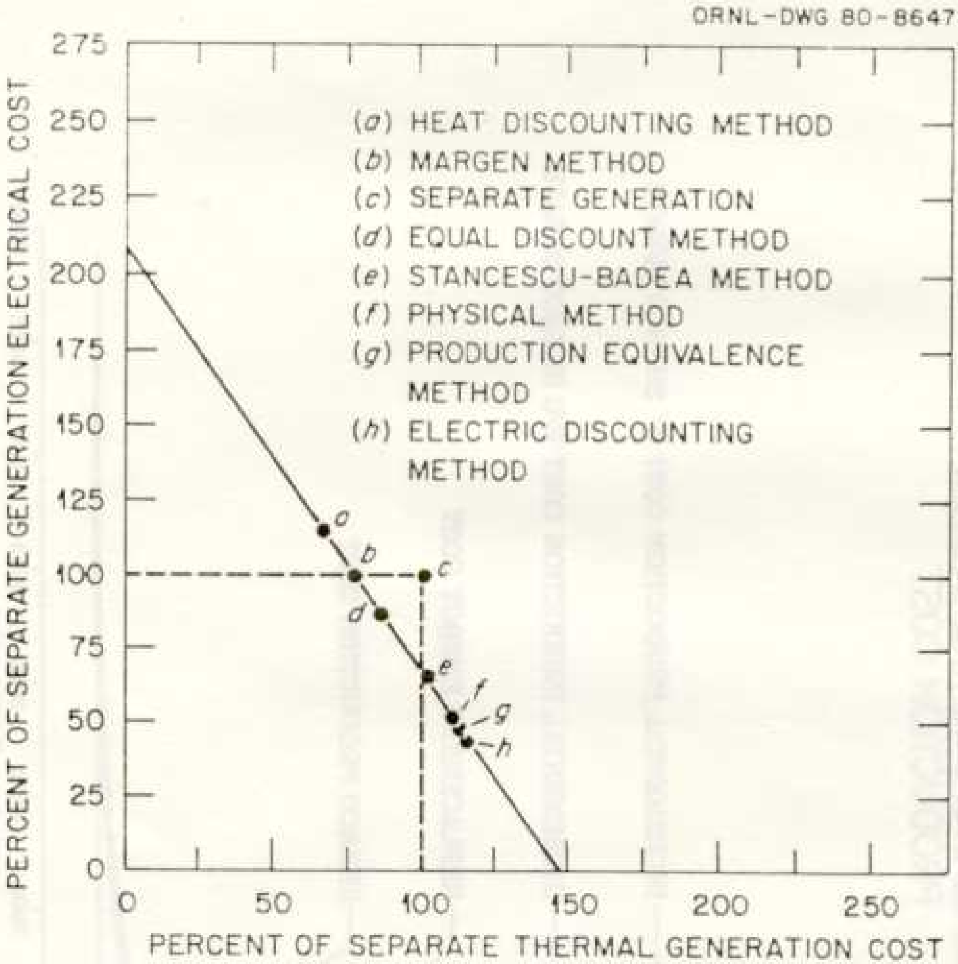



Fig. D-7. Costs calculated by various allocation methods compared with those of separate generation (assuming back-pressure turbines). (From G. F. Pavlenco and G. A. Englesson, District Heating/Cogeneration Application Studies for the Minneapolis-St. Paul Area - Allocation Methods for the Separation of Electrical and Thermal Cogeneration Costs, ORNL/TM-6830/P12, October 1980).

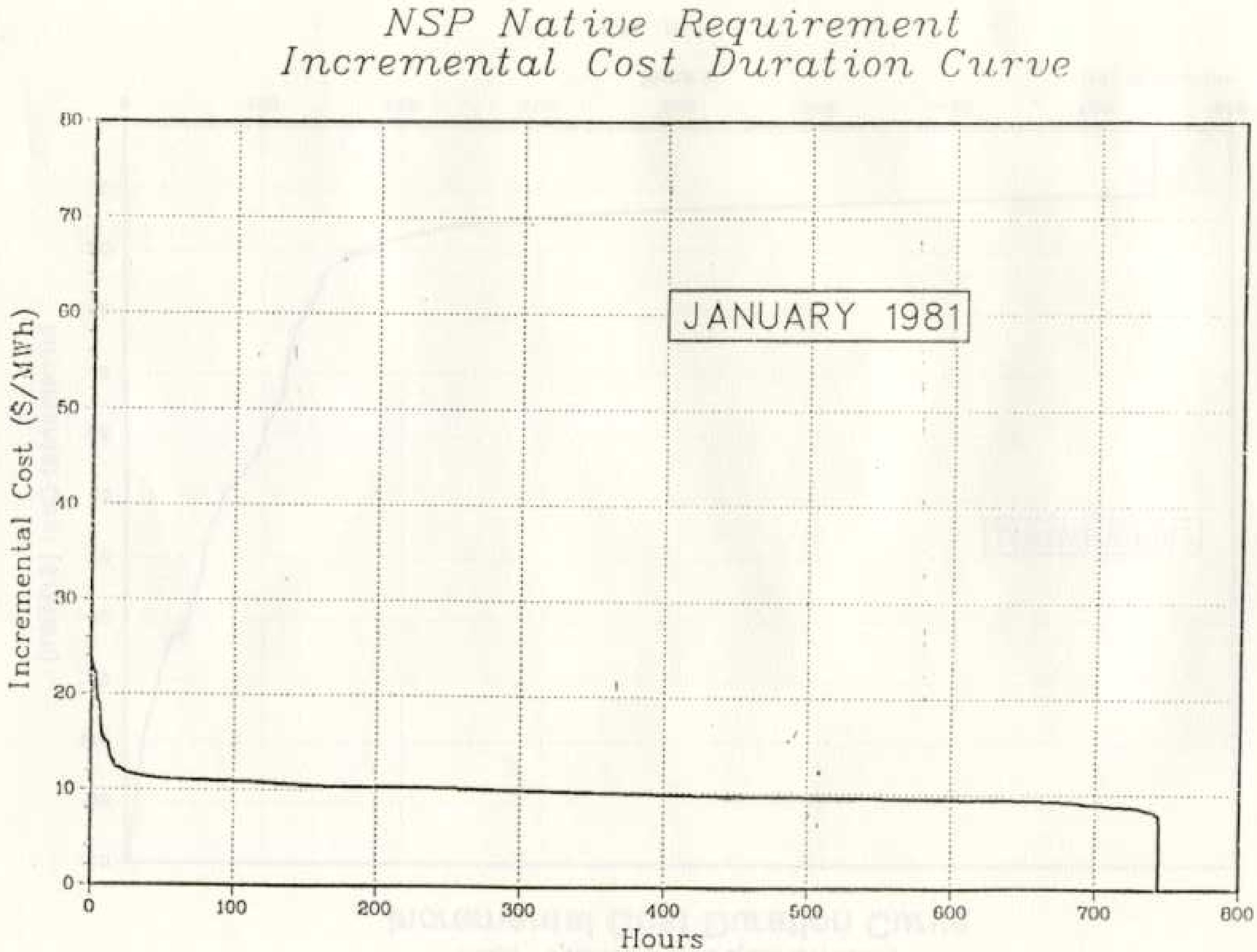

NSP Native Requirement
Incremental Cost Duration Curve
JANUARY 1981
Incremental Cost ($/MWh)
Hours
0
10
20
30
40
50
60
70
80
0
100
200
300
400
500
600
700
800

FIG D-8

NSP Native Requirement
Incremental Cost Duration Curve

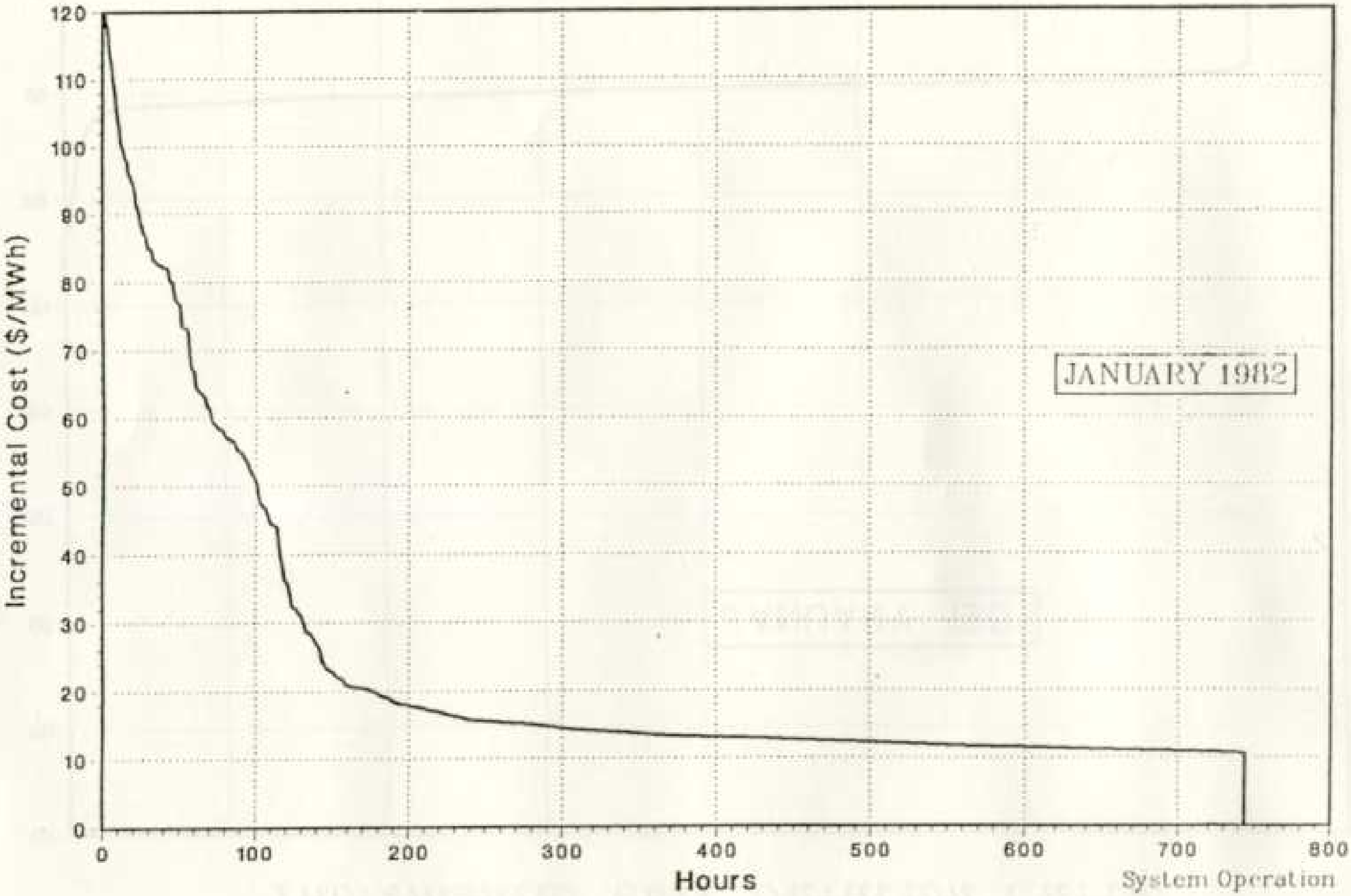



FIG D-9

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