This thesis analyzes two systems that can meet low-grade heating and/or cooling demands by operating in conjunction with thermal-electric power plants. At the application site, one of the systems, HPS, has water-to-air heat pumps that are connected to the supply and return lines of a plant-to-user loop. During heating, the heat pumps extract energy from the loop carrying the outlet of the power plant condenser cooling water; and in the cooling season, they reject energy to the loop water that is returning to a cooling tower or reservoir. In the other system, designated BOS, the loop always carries in water that is heated to a moderately high temperature at the power plant by steam bled off from the turbines. Then, heating is by direct transfer and cooling is from absorption units that reject waste energy locally. Besides meeting the demands effectively, both systems decrease the overall thermal pollution.

The systems and their potential are presented first. Then a general study formulates the performances in terms of the principal variables — power plant efficiency, power plant condenser temperature,
demand size and temperature. Both the thermodynamically ideal and more practical systems are considered. The basic trend in each case is revealed by plotting and comparing the system effectiveness values.

This is followed by a much more detailed evaluation of the system in supplying domestic water heating, and comfort heating and cooling to two cities with served populations of 21,500 and 129,000. The analysis includes: (i) models of the cities, (ii) models of the three power plants representative of nuclear (LWR, FBR) and fossil-fueled plants, (iii) actual weather data from three locations in Oregon, (iv) equipment performance data (compressors, absorption chillers, etc.), and (v) other design and energy use data. This information is combined to estimate the maximum, yearly and monthly energy requirements; design of the equipment and distribution network; and the combined system performance. The performances of the systems are compared in terms of (i) the electricity available for other uses after the thermal demands of the city are met and (ii) the net cooling requirements at the power plant site. The economic potentials of the BOS and the HPS are discussed in terms of the plant-city distance that lowers their cost to that of the least-cost conventional alternatives.

Finally, the components of the HPS are discussed in sufficient detail so that their individual performance can be balanced for overall performance estimation. A FORTRAN program is developed to simulate the design for hourly performance under various conditions; and the design is improved until the yearly range of all major variables are totally acceptable. In addition to demonstrating the feasibility of the heat pumps in this application, the simulation effort provides
a specific engineering design and an improved estimation of its performance.
ANALYSIS OF HEAT PUMP AND STEAM BLEED-OFF SYSTEMS LINKED WITH THERMAL-ELECTRIC POWER PLANTS TO SUPPLY LOW-GRADE HEATING AND COOLING

by

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for their consent in extending my stay abroad well beyond the original arrangements, and finally to
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# TABLE OF CONTENTS

## I. INTRODUCTION

1-1. The Term "Low-Grade"  
1-2. Scope of the Low-Grade Thermal Demands  
1-3. Thermodynamic Considerations on Providing Heating or Cooling  
1-4. Two Possible Systems  
1-5. Power Plants

## II. GENERALIZED PERFORMANCE

2-1. Scope of Analysis  
2-2. Ideal Systems  
2-3. Actual Systems  
2-4. Conventional Systems  
2-5. Conclusions

## III. URBAN APPLICATIONS

3-1. Communities Served  
3-2. Energy Requirements: Design and Yearly Total  
3-3. Energy Requirements: Yearly Profile  
3-4. System Design  
3-5. System Performance  
3-6. Other Climates  
3-7. Economics

## IV. DIGITAL SIMULATION OF THE HPS

4-1. Overall Scope and Goals  
4-2. Concepts and Design of the Components  
4-3. Simulation of the HPS  
4-4. Simulation Results  
4-5. Conclusions

## V. CLOSURE

5-1. Conclusions Summary  
5-2. Recommendations

## REFERENCES

## APPENDICES

Appendix A  Seasonal Value of $T_0$  
Appendix B  Notes on General Performance Considerations  
Appendix C  Optimum Loop Temperature  
Appendix D  HPS Simulation Program
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Availability; Area.</td>
</tr>
<tr>
<td>$A_f$</td>
<td>Availability of a flowing mass.</td>
</tr>
<tr>
<td>$A_Q$</td>
<td>Availability of heat transfer.</td>
</tr>
<tr>
<td>AES</td>
<td>All-Electric Systems (with resistance heaters and compression air conditioners).</td>
</tr>
<tr>
<td>BOS</td>
<td>Steam Bleed-Off System (see Section 1-4).</td>
</tr>
<tr>
<td>C</td>
<td>Capacity for refrigeration of a compressor; correction factor in the degree-day method.</td>
</tr>
<tr>
<td>$C_p$</td>
<td>Heat capacity. (at constant pressure)</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient of Performance.</td>
</tr>
<tr>
<td>E</td>
<td>Energy. (in general)</td>
</tr>
<tr>
<td>FBR</td>
<td>Fast Breeder Reactors. (liquid metal cooled)</td>
</tr>
<tr>
<td>FFP</td>
<td>Fossil Fueled Plants. (or Advanced Reactors)</td>
</tr>
<tr>
<td>HPS</td>
<td>Heat Pump Systems. (see Section 1-4)</td>
</tr>
<tr>
<td>h</td>
<td>Enthalpy; Convective heat transfer coefficient.</td>
</tr>
<tr>
<td>K</td>
<td>Arbitrary constant.</td>
</tr>
<tr>
<td>L</td>
<td>Thermal pollution at the power plant/fuel energy input.</td>
</tr>
<tr>
<td>LWR</td>
<td>Light Water Reactors.</td>
</tr>
<tr>
<td>m</td>
<td>Mass.</td>
</tr>
<tr>
<td>NTU</td>
<td>Number of Transfer Units. (for heat exchangers)</td>
</tr>
<tr>
<td>P</td>
<td>Pressure.</td>
</tr>
<tr>
<td>Q</td>
<td>Heat Transfer.</td>
</tr>
<tr>
<td>R</td>
<td>Ratio. (thermal load/power plant fuel input)</td>
</tr>
<tr>
<td>T</td>
<td>Temperature.</td>
</tr>
</tbody>
</table>
Dead state temperature.

Time.

Over-all heat transfer coefficient.

Systems with furnace heating and furnace-absorption cooling.

Systems with usual heat pumps.

Work (or power) transfer.

Effectiveness - a Second Law (availability) based efficiency.

Energy efficiency.

Grade.

Difference.

Enthalpy change of reaction for power plant fuel. (= - Higher heating value)

Air; air-side.

Related to absorption cooling.

All-Electric System.

At the conditions of steam bled off from power plant turbine.

Bleed-Off System.

At the temperature of the power plant condenser; for an uncoupled, single purpose power plant.

Cooling application.

Heat pump condenser.

Power plant excluding steam generator.

Electrical.
ev  Heat pump evaporator.
F  Fuel (of the power plant)
f  Furnace; flow
HP  A heat pump unit.
HPS  Heat Pump System.
h  Heating application.
i  Inside; summation index.
in  Incoming.
L  Lost or wasted.
lm  Limiting.
loop  Power plant - application piping loop.
m  Mean. (average)
max  Maximum.
r  Refrigerant - side.
SG  Steam generator at the power plant.
o  Outside.
out  Outgoing.
UFS  Usual Furnace System.
UHS  Usual Heat Pump System.
0  Dead state. (state of zero availability)

(Subscripts may be used in combination, separated by hyphen or comma)

Overline
:
  Time rate.
'
  Specialized for the conditions of $\eta_{SG} = - \frac{\Delta H}{A_F} = 1.0$.
*  Total or effective value of a parameter in a long period.
# LIST OF TABLES

<table>
<thead>
<tr>
<th>Table</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>3-1</td>
<td>Symbols and Unit Sizes for City Layout.</td>
<td>59</td>
</tr>
<tr>
<td>3-2</td>
<td>Heating and Cooling Design Values for Portland, Oregon.</td>
<td>64</td>
</tr>
<tr>
<td>3-3</td>
<td>Domestic Hot Water Requirements.</td>
<td>66</td>
</tr>
<tr>
<td>3-4</td>
<td>Design Energy Requirements Per Unit for Portland Climate.</td>
<td>68</td>
</tr>
<tr>
<td>3-5</td>
<td>Yearly Space Heating and Air Conditioning Energy Requirements for Each Type of Building for Portland Climate and City of 129,000 Population.</td>
<td>69</td>
</tr>
<tr>
<td>3-6</td>
<td>Yearly Space Heating and Air Conditioning Energy Requirements for Each Type of Building for Portland Climate and City of 21,500 Population.</td>
<td>70</td>
</tr>
<tr>
<td>3-7</td>
<td>Heating Degree Days and Cooling Degree Hours Used for Monthly Distribution of Apartment Space Heating and Air Conditioning Requirements (65° Base).</td>
<td>77</td>
</tr>
<tr>
<td>3-8</td>
<td>Monthly Average Circulation Requirements of Plant-Town Loop for the Heat Pump System with the City of 129,000 in the Portland Climate.</td>
<td>87</td>
</tr>
<tr>
<td>3-9</td>
<td>Monthly Average Circulation Requirements of Plant-Town Loop for the Heat Pump System with the City of 21,500 in the Portland Climate.</td>
<td>88</td>
</tr>
<tr>
<td>3-10</td>
<td>Design and Climatic Conditions of the Three Locations.</td>
<td>91</td>
</tr>
<tr>
<td>4-1</td>
<td>The Position of the Controlling Elements During the Various Modes.</td>
<td>140</td>
</tr>
<tr>
<td>4-2</td>
<td>Simulation Results for the System-check Variables. (For each month the upper line gives the simulation results, and the lower line is from Chapter 3 analysis)</td>
<td>154</td>
</tr>
<tr>
<td>Table</td>
<td>Page</td>
<td></td>
</tr>
<tr>
<td>---------</td>
<td>------</td>
<td></td>
</tr>
<tr>
<td>4-3</td>
<td>158</td>
<td></td>
</tr>
<tr>
<td>Simulation Results for the System-performance Variables. (For each month, the upper line gives the simulation results, and the lower line is from Chapter 3 analysis.)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4-4</td>
<td>160</td>
<td></td>
</tr>
<tr>
<td>Total Space Heating and Cooling Loads and Degree Hours. (For each month the upper line gives the simulation results, and the lower line is from Chapter 3 analysis.)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4-5</td>
<td>164</td>
<td></td>
</tr>
<tr>
<td>The Effects of High Loop Flow Rate Operation During Space Cooling in May 1952.</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
# LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1-1</td>
<td>3</td>
</tr>
<tr>
<td>1-2</td>
<td>12</td>
</tr>
<tr>
<td>1-3</td>
<td>14</td>
</tr>
<tr>
<td>1-4</td>
<td>14</td>
</tr>
<tr>
<td>1-5</td>
<td>17</td>
</tr>
<tr>
<td>2-1</td>
<td>31-32</td>
</tr>
<tr>
<td>2-2</td>
<td>37</td>
</tr>
<tr>
<td>2-3</td>
<td>41</td>
</tr>
<tr>
<td>2-4</td>
<td>44</td>
</tr>
<tr>
<td>2-5</td>
<td>45</td>
</tr>
<tr>
<td>2-6</td>
<td>49</td>
</tr>
</tbody>
</table>

1-1 The variation of the heat transfer grade with the application temperature. (The dead state is specified at 500°F here; the basic nature of the grade curve remains the same as $T_0$ varies seasonally.)

1-2 A simplified schematic showing the concept of the HPS as adopted to space heating.

1-3 A simplified schematic showing the concept of the BOS at the power plant site.

1-4 A simplified schematic showing a typical absorption refrigeration unit for space cooling with the BOS [57].

1-5 Power plant efficiency as a function of the saturation temperature for the exhaust pressure [57].

2-1 (a-g) The performance of the ideal heating system as related to $T_C$, $T_h$, and $R_h$. The figures a through d present the BOS, while e and f are for the HPS. The two systems are compared in g. Note $c_0$ indicates the performance of an uncoupled power plant.

2-2 (a,b) The performance of the ideal BOS for cooling as related to variables $T_C$, $T_r$, $R_c$, and $T_{BO}$. Note that the performance of the ideal HPS during cooling equals that of the power plant itself.

2-3 (a-d) The performance of the actual heating systems as related to $T_C$, $T_h$, and $R_h$.

2-4 (a,b) Critical coefficients of performance in heating or cooling that equalize the overall effectiveness of the two systems.

2-5 (a-c) The performance of the actual cooling systems as related to $T_C$, $T_r$, and $R_c$. The performances shown individually in a and b, for the BOS and the HPS, are compared in c.

2-6 (a,b) The performance of the conventional heating systems are displayed in Figure a, and compared with that of the HPS in Figure b.
Meeting space cooling demands with a furnace - absorption unit combination. The performance displayed in Figure a is compared with that of HPS in Figure b.

Layout for city with 200,000 population. High density residential area of 129,000 people and the downtown area are served by the district heating system [57].

Detail of college and downtown area for city with a population of 129,000 served people [57].

Typical residential mile square in apartment area of city of 129,000 served people [57]. The symbols are described in Table 3-1.

Typical apartment-block A. Each apartment building is to be 3 stories in city of 129,000 served population, 2 stories in city of 21,500 served population [57].

Layout for city of 21,500 served population.

Detailed layout for the area in the city with 21,500 served residents. The symbols are described in Table 3-1.


Hourly profile of apartment hot water demand on an average day [63].

Averaged and smoothed rate profiles for monthly distribution of annual heating and cooling loads for non-residential buildings in Portland, Oregon. Data were based on information supplied by the Portland General Electric Company.

Monthly average energy requirements for water heating, air conditioning, and space heating for the Portland climate and city with 129,000 served people.

Monthly average energy requirements for water heating, air conditioning, and space heating for the Portland climate and a city of 21,500 served people.
Figure

3-12 Pipelines for a typical square mile of the residential area in a city with 129,000 served people. The numbers indicate pipe sizes. All unmarked extensions are 3 inch extensions to apartment blocks. (Portland climate)

3-13 Pipelines for the downtown and college area in the city with 129,000 served people. The numbers indicate pipe sizes. (Portland climate)

3-14 Mains and branch mains for the piping system for the city with 129,000 served people. The numbers indicate pipe sizes. (Portland climate)

3-15 Pipelines for the 21,500 served population city. Numbers indicate pipe sizes. All unmarked extensions are 3 inch branches for apartment block A.

3-16a Schematic diagram of the heat pump operation during the space heating cycle.

3-16b Schematic diagram of the heat pump operation during the space cooling cycle.

3-17a Coefficient of performance for the heat pump unit.

3-17b Performance of the heat pump unit, operating as an air conditioner, as a function of power plant condenser temperature and loop temperature change.

3-18 The monthly average inlet and outlet temperature for a natural draft cooling tower operating at a 1100 MW power plant in Portland, Oregon with a constant temperature drop of 37.5°F. The solid line shows the temperature of the loop water received by the heat pumps of the HPS.

3-19 Piping system for a typical square mile area in the city with 129,000 served population in a Portland climate using the heat pump system with a temperature change of 40°F for the loop water.

3-20 Pipelines in the downtown and college area of the 129,000 served population city in a Portland location for the heat pump system with a temperature change of 40°F for the loop water.

3-21 Mains and branch mains of the piping system for the 129,000 served population in a Portland climate using the heat pump system with a temperature change of 40°F for the loop water.
Pipelines for 21,500 served population city in a Portland location for the heat pump system with a temperature change of 40°F for the loop water. Numbers indicate pipe sizes. All unmarked extensions are 4" lines for apartment block A.

Total monthly average energy requirements for space heating, air conditioning and hot water heating for 129,000 people living in a city of Portland climate, using the steam-bleed-off system.

Monthly average electricity output by a nominal 1100 MWe LWR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Portland climate. Meaning of symbols is explained in the text.

Monthly average electricity output by a nominal 1100 MWe FBR power plant serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Portland climate. Meaning of symbols is explained in the text.

Monthly average electricity output by a nominal 1100 MWe FFP power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Portland climate. Meaning of symbols is explained in the text.

Cooling requirements of nominal 1100 MWe FBR or FFP power plants serving the urban energy requirements of 129,000 people living in a Portland climate. Meaning of symbols is explained in the text.

Total monthly average energy requirements for space heating, air conditioning, and hot water heating for 21,500 people living in a city in a Portland climate, using the steam-bleed-off system.

Monthly average electricity output by a nominal 1100 MWe LWR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 21,500 in a Portland climate. Meaning of symbols is explained in the text.
<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>3-30</td>
<td>Monthly average electricity output by a nominal 1100 MWe FBR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 21,500 in a Portland climate. Meaning of symbols is explained in the text.</td>
</tr>
<tr>
<td>3-31</td>
<td>Monthly average electricity output by a nominal 1100 MWe FFP power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 21,500 in a Portland climate. Meaning of symbols is explained in the text.</td>
</tr>
<tr>
<td>3-32</td>
<td>Cooling requirements of nominal 1100 MWe LWR, FBR or FFP power plants serving the urban energy requirements of 21,500 people living in a Portland climate. Meaning of symbols is explained in the text.</td>
</tr>
<tr>
<td>3-33</td>
<td>Total monthly average electricity requirements for space heating, air conditioning, and hot water heating for 129,000 people living in a city in a Portland climate, using the heat pump system.</td>
</tr>
<tr>
<td>3-34</td>
<td>Monthly average electricity output by a nominal 1100 MWe LWR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Portland climate. Meaning of symbols is explained in the text.</td>
</tr>
<tr>
<td>3-35</td>
<td>Monthly average electricity output by a nominal 1100 MWe FBR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Portland climate. Meaning of symbols is explained in the text.</td>
</tr>
<tr>
<td>3-36</td>
<td>Monthly average electricity output by a nominal 1100 MWe FFP power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Portland climate. Meaning of symbols is explained in the text.</td>
</tr>
<tr>
<td>3-37</td>
<td>Cooling requirements of nominal 1100 MWe LWR, FBR, or FFP power plants serving the urban energy requirements of 129,000 people living in a Portland climate. Meaning of symbols is explained in the text.</td>
</tr>
<tr>
<td>3-38</td>
<td>Total monthly average electricity requirements for the city of 21,500 population in a Portland climate with the heat pump system.</td>
</tr>
</tbody>
</table>
Figure 3-39  Monthly average electricity output by a nominal 1100 MWe LWR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 21,500 in a Portland climate. Meaning of symbols is explained in the text.

3-40  Monthly average electricity output by a nominal 1100 MWe FBR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 21,500 in a Portland climate. Meaning of symbols is explained in the text.

3-41  Monthly average electricity output by a nominal 1100 MWe FFP power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 21,500 in a Portland climate. Meaning of symbols is explained in the text.

3-42  Cooling requirement of nominal 1100 MWe LWR, FBR, or FFP power plants serving the urban energy requirements of 21,500 people living in a Portland climate. Meaning of symbols is explained in the text.

3-43  Total monthly average electricity requirement for space heating, air conditioning and hot water heating for 129,000 people in an Astoria climate.

3-44  Total monthly average energy requirements for space heating, air conditioning and hot water heating for 129,000 people living in a city in a Pendleton climate.

3-45  Monthly average electricity output by a nominal 1100 MWe LWR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in an Astoria climate. Meaning of symbols is explained in the text.

3-46  Monthly average electricity output by a nominal 1100 MWe FBR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in an Astoria climate. Meaning of symbols is explained in the text.
3-47 Monthly average electricity output by a nominal 1100 MWe FFP power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in an Astoria climate. Meaning of symbols is explained in the text. 122

3-48 Cooling requirements of nominal 1100 MWe LWR, FBR, or FFP power plants serving the urban energy requirements of 129,000 people living in an Astoria climate. Meaning of symbols is explained in the text. 122

3-49 Monthly average electricity output by a nominal 1100 MWe LWR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Pendleton climate. Meaning of symbols is explained in the text. 123

3-50 Monthly average electricity output by a nominal 1100 MWe FBR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Pendleton climate. Meaning of symbols is explained in the text. 123

3-51 Monthly average electricity output by a nominal 1100 MWe FFP power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Pendleton climate. Meaning of symbols is explained in the text. 124

3-52 Cooling requirements of nominal 1100 MWe LWR, or FFP power plants serving the urban energy requirements of 129,000 people living in a Pendleton climate. Meaning of symbols is explained in the text. 124

4-1 Hierarchy of the subprograms in the HPS simulation program. 146
ANALYSIS OF HEAT PUMP AND STEAM BLEED-OFF SYSTEMS LINKED WITH THERMAL-ELECTRIC POWER PLANTS TO SUPPLY LOW-GRADE HEATING AND COOLING

CHAPTER I

INTRODUCTION

This thesis focuses on systems that can meet "low grade" thermal demands effectively. This introductory chapter discusses the concept of "grade" in reference to energy; reflects on the importance of these requirements in the national energy picture, and the very wasteful nature of many of the conventional methods meeting them; and conceptually considers two systems that show promise for meeting the low-grade requirements in an effective manner. In addition, the central station thermal-electric power plants, that are to be coupled to each of the two systems, are specified. Much of the material presented in subsequent chapters is devoted to analysis and modification of the systems introduced here.

1-1. The Term "Low-Grade"

The grade (in reference to energy) is defined as the ratio of availability to energy.\(^1\) This explicit definition has been

\(^{1}\)"Availability" (or "available energy") is a thermodynamic function indicative of the maximum (ideal) useful work (or heat) interaction capability of a system or process in the common atmosphere at \(T_0\) and \(P_0\). Extensive discussions and applications of availability and the related concepts may be found in many publications, for example References 1 through 9.
introduced only recently [7], even though the concept and the term itself has been in use for many years [9]. As the value of the grade is dimensionless and often less than unity, it can be used as a useful shorthand in availability considerations. Since chemical, nuclear, electrical, kinetic, and potential forms of energy are essentially completely available for useful work transfers, these energy forms and all work transfers are of high grade (almost unity). A heat interaction, however, cannot in general be fully converted to work, since its availability depends on the temperatures of the system boundary (T), and the dead state (T₀): ²

\[ A_q = (1 - T/T_0) Q \]

(1-1)

Noting that when T<T₀, Q<0 by convention, the grade Τ for heat transfers, which are of prime interest here, becomes

\[ \Gamma = \left| 1 - T_0 / T \right| \]

(1-2)

Clearly, \( \Gamma \leq 1 \) as long as \( T > T_0 / 2 \). Figure 1-1 illustrates the variation of grade with T for a specified T₀. Ironically, the very high levels of grade, much above 1 as shown in Figure 1-1, can be obtained only with heat transfer to sinks at T<T₀/2.

²The dead state refers to the state of equilibrium with the surroundings (hence of no availability). Therefore, in calculations for an instantaneous activity, the value to be selected for T₀ is the temperature freely occurring in the close-by surroundings at the time. Appendix A considers what value should be used in seasonal evaluations. As indicated there, a simple time average of the instantaneous T₀ values is usually an acceptable approximation.
Fig. 1-1. The variation of the heat transfer grade with the application temperature. (The dead state is specified at 500°R here; the basic nature of the grade curve remains the same as \( T_0 \) varies seasonally.)
The above development indicates that the closer the temperatures $T$ and $T_0$ are, the lower the grade. In this thesis, no strict upper-limit will be specified to denote "low-grade", but noting that $\Gamma < 0.3$ for heating below 212°F or cooling at above 32°F, low-grade could refer to any or all of the following applications: nearly all space heating and cooling, domestic water heating and similar needs; many industrial and commercial processes such as fish farming, soil warming, de-icing, recreational pool heating, fermentation, drying, desalination, sewage treatment, refrigerating and the like. The share of such demands in the overall energy allocation is very significant, as the next section illustrates.

1-2. Scope of the Low-Grade Thermal Demands

Low-grade applications constitute over half of all thermal demands, or about one-third of all energy resource usage. Specifically, in the USA, the annual space heating in the residential-and-commercial sector exceeds $10^{16}$ Btu, which is about 20 percent of the total national energy consumption. Water heating uses about 4 percent, and over 3 percent is consumed for space cooling. Of course, it is possible to reduce these significantly by broadly applying

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3 Since estimates are included in many recent publications, and they vary somewhat with time and assumptions, here only rounded-off values are cited without specific references. Detailed presentations can be found in References 10 through 19, and others.

4 A very low saturation level of comfort cooling, also, indicated by an annual growth of about 10 percent, is one of the factors keeping the allotment for this purpose comparatively small. Utilizing fairly effective systems (with an average COP of about 2.5) for this demand type is another major factor.
strict energy saving measures, but the present trend shows that meager efforts in that direction are now more than offset by new demands.

The effects of these low-grade demands on the energy balance of a building are equally impressive. The totals as well as the percentages depend on the location (climate) and the type (construction, operation) of building; but for the residential-and-commercial sector in the USA, on the average, over 50% of the yearly energy requirement is for space heating and almost 10% for space cooling. The latter naturally contributes much more heavily to the summer load: 56% for commercial buildings and over 30% for residences [13]. Too, water heating surprisingly requires more energy than all refrigeration and cooking combined. In fact, for a mild Portland, Oregon apartment design (presented in Chapter 3), the annual water heating is as high as 75% of the annual space heating load.

Even though there is no doubt of the significance of these demands, there has been few changes in the methods of meeting them. Since the discovery of fire, supplementary space heating has been achieved mostly by the combustion of wood and fossil fuels in fireplaces, stoves, local furnaces or boilers, and the like. These still serve about three quarters of the demand with efficiencies that start from a low of below 50%, often exceed 70%, and even reach over 90% in some designs - the energy losses being mainly from the stack. Much of the remaining demand is served by
electrical resistance heating which despite its poor net efficiency is showing a steady growth by capturing too much of the new demand.\footnote{5} The situation is more or less the same for water heaters. However, the possibility of using process or air conditioning waste heat for water preheating is apparent in many cases; and the projects that attempted to do it usually report a very successful operation \[22,39\]. With the exception of heat pump systems, space cooling is obtained by separate equipment. Electrical compression type units with a COP range of about 2.0 to 4.0 are common; but absorption chillers are also in use. Instead of using electricity for the compression, the latter utilizes thermal energy that is at a moderate temperature (200-300°F) and about equal in magnitude to the cooling load met at the evaporator. All other means, which include thermoelectric, evaporative, night sky-radiation, and natural ice, account for a minor fraction.

Since the energy requirements given above involve both the loads and the efficiencies, increasing the equipment effectiveness
has an effect equivalent to reducing the load in the same ratio. Yet, while the magnitude of the low-grade thermal demands helped to initiate worthy efforts to reduce those requirements by conservative building design and operation, supplementing such efforts with attempts on improving the methods used for providing the essential needs are severely limited due partly to the satisfactory appearance of the performance values just cited. However, effectiveness improvements may be applied not only independently from load reductions, but usually with greater range and less sacrifice. Therefore, the results of available energy considerations, summarized in the next section, are very illuminating for they demonstrate the source as well as the degree of resource wasting that the current heating systems involve; and in the process, also signal the paths to possible improvements.

1-3. Thermodynamic Considerations on Providing Heating or Cooling

For comfort heating/cooling, and water heating, the required temperature is so close to $T_0$ that the difference between the energy and availability transfers becomes vast. Thus, most systems that meet these low-grade demands directly from high quality fuel resources are doomed to a very poor performance. This has been discussed previously [5-7, among others], and its significance in energy resource planning is detailed in recent publications [40,41]. In fact, while the energy conversion efficiencies of most common space or water heaters are above 0.6, the corresponding availability
effectiveness values are all below 0.25 and more like 0.1 when the conversion is directly from prime fuels [41]. On the other hand, systems whose output is in the form of a work transfer or equivalent suffer little in the availability analysis. Thus, for example, thermal-electric power plants appearing much less efficient than common home furnaces, actually utilize the natural resources two to three times more effectively (as the average $\epsilon \approx n \approx 0.4$). The implication is that the heating methods deserve more attention not only because of their large share in the natural energy allocation, but even more significantly because of the very low performance level of common systems. The rest of this section addresses itself to the general problem of elevating the performance of low-grade systems in the light of availability considerations.

Solutions to this problem could come from two directions: One is to obtain as much of the energy as possible from low-grade sources. These sources could be natural, such as hot springs and wells; or they could be an undesired output of another process, such as the exhaust of a steam turbine, warm air about an engine or computer, etc. Of course, these sources have been recognized and, sometimes, are utilized, as in district heating, total energy and special heat pump projects. Also, the equipment can be improved or replaced with a complete redesign in order to come closer to the perfect performance ideally possible. Actually this could be done in combination with the first step, but becomes more important when high grade sources must be used. Examples of this type
of effort are seen in the replacement of furnaces and electrical resistance heaters with total energy or heat pump systems.

The heat pump is particularly interesting as it can frequently combine both methods: In an ordinary heat pump, all the availability comes from the electricity (or another high-grade source) since the atmosphere can supply none. Yet a heat pump almost always utilizes the electricity much more effectively than the resistance heaters, due to a better design concept\(^6\) (which basically reduces the temperature gradients). But heat pumps can also use energy at the lowest grade. When the atmosphere is replaced with a source at a higher temperature, a part of the availability comes from this source; hence the electricity requirement is reduced, and the performance is improved. Furthermore, the same equipment may be used to deliver heating, or cooling, or better yet, simultaneous heating and cooling - in which case energy is transferred from where it is not wanted to where it is needed, with little loss and maximum effectiveness.

An alternative method exists with the use of moderate-temperature thermal energy - such as obtainable from geothermal wells, solar concentrators, industrial processes, exhaust of engines and gas or steam turbines. A number of total energy and district systems

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\(^6\)Thus, a deficiency, apart from the losses in electricity generation and transmission is inherent to the resistive heating. However, electrical heating may have superior performance if the stray losses from the heat pump are excessive due to high demand temperature or poor design. Also, a heat pump is impractical for certain applications because of space, weight, portability, cost, constraints, etc.
operate with one of these. In most of them the engines or turbines generate some electricity, and then their exhaust is used for heating by direct energy exchange, or for cooling by absorption chilling. This is an attractive scheme because the electricity usage, hence the losses (and the demand peak) are reduced.

The next section introduces for further evaluation two systems that incorporate some of the above ideas by thermally coupling the application equipment with a large thermal-electric power plant.

1-4. Two Possible Systems

At the present, about 85% of the USA electricity is generated through thermal energy. The percentage should get even higher in the future since the hydroelectricity has essentially reached its peak. The amount of thermal-electric conversion will, in all likelihood, increase with the growing demand. It is well known that in all thermal-electric conversions some of the thermal energy must be rejected (at a condenser for most types), that the amount is huge (nuclear plants reject over 65% of the input energy to their condensers), and that the foreseeable technology cannot reduce it much below 50% [42]. Yet, while this energy is not valuable for work, it is usable as thermal energy. In fact, since for practicality the condenser outlet is permitted to have a temperature up to about 30°F higher than $T_0$, the water is warm enough for direct use in some algae or seafood basins, recreational
pools, warm irrigation, etc. But many other demands, such as space or water heating will require upgrading the level. Two distinct methods, (i) the heat pump system (HPS), and (ii) the steam bleed-off system (BOS), which could be used are outlined next.

**HPS:** It is an exciting proposal to send the outlet of the power plant condenser cooling water to the major demand centers, and use it as the source or sink of heat pump units, and then bring the water back to the plant to complete the loop. Figure 1-2 illustrates the basic concept of such a Heat Pump System (HPS) for a space heating application. A similar or augmented system can be used for other heating demands; and a reversal of the refrigerant flow is all that is needed for cooling applications. During a primarily cooling period, the loop preferably brings the colder water from the power plant cooling system (or a local equivalent). The water warmed at the heat pump condenser need now be returned to these cooling systems for recycling, but some of it could be beneficially used for example in irrigation. The heat pumps of such a system are coupled to an almost ideal source/sink - quite dependable, clean and with a very acceptable temperature range that eliminates frosting and provides high performance. In this work, the performance of HPS is examined in

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Various topics related to the utilization of warm water from power plants are discussed in References 43 through 58.
Fig. 1-2. A simplified schematic showing the concept of the HPS as adopted to space heating.
detail, in Chapter 2 for general applications, and in Chapters 3 and 4 for an urban setting with a mild Portland, Oregon climate.

**BOS:** Recently (1971) the Oak Ridge National Laboratory (ORNL) completed an extensive study [57] on another system which supplies water hot enough (about 300°F) for direct use. As Figure 1-3 reveals, the circulating water is heated at the power plant by bleed-off steam. At the application site, heating is accomplished by direct energy exchange, while cooling is by absorption units such as shown in Figure 1-4, that reject energy to local cooling towers. In both cases, the cooled loop water returns to the plant for reheating. This setup, referred to here as the Bleed-Off System (BOS), could have several design variations. The schemes and temperatures shown here are adopted from Reference 57 with major urban uses in mind.

Bleeding some steam before its full expansion of course reduces the electricity output. But the heating demands are met efficiently, and thus there will be more electricity available for other uses than there would be if the same demands were met by resistance heating or by firing part of the fuel directly. Whether the reduction will be less than the electricity required by the HPS serving the same demands is less certain at this point.

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8 The examination of this application of the heat pumps was originally proposed by the thesis advisor, G. M. Reistad, in 1971. A proposal and a brief examination of a similar setup, but with a single-line loop and only space heating/cooling, was reported in 1951 [56]. The author and Dr. Reistad became aware of that work in 1974.
Fig. 1-3. A simplified schematic showing the concept of the BOS at the power plant site.

Fig. 1-4. A simplified schematic showing a typical absorption refrigeration unit for space cooling with the BOS [57].
To answer this, and other comparison questions, the evaluation of the BOS is carried out in parallel to that of the HPS in all cases considered in this work.

Nevertheless, several comparative observations about the systems are possible directly from the concepts. Most striking is that the heat pump system does not affect the internal operation at the power plant: part of the cooling water is simply rerouted to the demand site (or, perhaps a heat exchanger is used to keep plant site loop and application loop separate). This independence is highly welcomed by most utility planners. Also, the loop temperature is much lower for the HPS, hence the distribution system requires little or no insulation. Thus, the use of an existing network, such as present city water lines is in fact a possibility. But the allowable loop temperature drop is much smaller for the heat pump system, resulting in larger pipes and loop flow rates for the HPS. This diminishes the savings from reduction in insulation and raises the operation costs. For longer distances a slightly modified loop temperature profile could be considered if savings with higher ΔT outweigh the losses due to drop in performance.

The local equipment also differs, the heat pump unit being more compact than the heat exchangers plus absorption unit of the BOS. Further, the local cooling towers required by the BOS for the absorption units are optional in the HPS with the possibility of using the cooling system of the power plant itself.
Finally, the heat pump system is not very suitable when the heating temperature exceeds, say, 150°F.

The implication of this comparison is that while under some cases the choice could be made simply from the concept considerations, in general both systems promise satisfactory service and their performance should be analyzed for the specific application before the final choice is made.

1-5. Power Plants

In the following chapters the two systems are further specialized as required by the particular applications. However, the power plant, which is common to both systems and remains unchanged in the whole study, can now be discussed in full.

Both systems could be tied to any thermal-electric power plant with a condenser. The major plant specification for this work is the electricity generation efficiency as a function of the steam exhaust temperature. The data given in Figure 1-5 is adopted as representative of the three types of present and planned units: light water reactors (LWR), liquid metal cooled fast breeder reactors (FBR), and fossil fueled plants (FFP). All analyses are evaluated for each of these power plants. The nuclear plants have lower cycle efficiencies than the fossil fueled plants though the future advanced-nuclear plants can close the difference. Also, the efficiencies are for steam-to-electricity conversion and hence do not include the boiler related losses.
Fig. 1-5. Power plant efficiency as a function of the saturation temperature for the exhaust pressure [57].
which will be especially significant for FFP. The wide temperature range of Figure 1-5 is for use with BOS which bleeds steam at high temperatures. In general only a small portion of the steam is bled and the overall plant efficiency will not vary so drastically. On the other hand, there will be minor yearly fluctuation of the main condenser temperature as it depends on the outside conditions. Cooling towers in general result in the higher temperatures and a wider range than those with either reservoirs or once-through systems. In a specific case, the yearly profile of the condenser temperature must be specified.

One more item of importance about the plant is its location as related to the demand centers, since the economical distribution of the water is limited to a distance that depends on the system and usage rate. Some customers such as fish ponds, green houses, or some industry could be built near the power plant. Unfortunately, the environmental conditions force the power plants away from one major potential user - the cities. Nevertheless, a study of the present situation [59] indicates the possibility of a fairly large city within a reasonable distance of ten miles. Further, the possibilities of cooperation, and the other uses as mentioned above may prove to be sufficient incentives for the development of new population centers close to the power plants.

9 In a suggested projection for the USA (the 1980 estimate for the Indian Point plant), the cumulative population is 100,000 within five miles, 300,000 within ten miles, and 700,000 within fifteen miles.
Thus far, the role of the power plant has been stressed as a provider. However, as is well realized, the tremendous amounts of energy that must be disposed at large thermal-electric power plants have become a major burden as the number of the available sinks has decreased. Therefore, any possible utilization of this energy could provide a welcome relief. In fact, a wide range research effort on power plant waste water utilization [58] (which included the work reported in Chapter 3 of this thesis), and similar efforts in the USA and elsewhere [52, 53, 54] are sponsored by utilities mainly searching for new energy sinks for their future plants.

The bleed-off system decreases thermal pollution because the steam that heats the circulating water has produced some electricity without contributing to thermal pollution since the steam condenses as it heats the circulating water rather than the condenser cooling water. And the heat pump system decreases thermal pollution by decreasing the electricity required for heating by a factor up to the heat pump COP, and also by using low grade energy from the power plant rather than dumping it to the cooling system. There is a slight contribution even during cooling as the COP of the HPS would be better than that of common air-to-air units.
CHAPTER II

GENERALIZED PERFORMANCE

The potential users of the BOS or the HPS mainly influence the system performance through two variables - the magnitude and temperature of the demand. Several other variables are introduced by the power plant, its main cooling system and the climate. Therefore, a general study with these fundamental variables should reveal the principal trends and provide an initial estimation for any specific case. This chapter will examine and compare the performances of the two basic systems, BOS and HPS, in supplying purely heating or purely cooling. First, the ideal, and then an actual operation is analyzed; but for both cases, the power plant models will represent actual operation.

The approach taken is to consider a typical power plant and compare the overall performance of generating electricity and providing the specified thermal energy requirements when either the BOS or HPS is coupled to the power plant. Then, in essence, this approach looks for the best performance given that there is a specified energy source rate and a specified thermal requirement rate, and that all of the electricity generated is beneficial (supplied to an electrical grid).

A similar problem, but one to which this analysis is not directly applicable is when the split between thermal and electrical requirements is specified, and the goal is to obtain the best
performance indicated by the least resource input. This latter problem is more applicable to total energy systems than the base-loaded thermal-electric power plants considered here.

2-1. Scope of Analysis

The analysis is kept quite general by consideration of a number of variables:

Power Plant Types

Power plants are characterized for this analysis simply by their cycle efficiency as a function of steam exhaust conditions, as indicated in Figure 1-5.

Thermal Load Ratio

The thermal load ratio variables, $R_h$ and $R_c$, allow the analysis to be done without restricting attention to a specific plant size or thermal requirement. The variables $R_h$, the ratio of heating requirement to energy input to the plant, and $R_c$, the ratio of cooling requirement to energy input to the plant are considered for the range 0 to 0.4. Each system has a limiting $R_h$ or $R_c$ (mostly greater than 0.4; Appendix B evaluates these limiting $R$ values, and the remaining thermal pollution when serving lesser loads).

Condenser Temperature

The power-plant condenser temperature is set equal to the ambient (dead state) temperature plus 30°F:

$$T_C = T_0 + 30 \quad (^{°}F)$$

(2-1)
Variation of the condenser temperature is thus also a variation of the ambient temperature and reflects the effects of changing weather conditions on the system performance for a fixed $R_h$ or $R_c$. The condenser temperature is allowed to vary from 65°F to 120°F.

Temperature of Thermal Requirement

The temperature at which a thermal requirement is needed specifies its availability and therefore is a very important parameter in any thermodynamic analysis of systems such as those considered here. In the ideal system, heating temperatures are investigated for the range from condenser temperature to 400°F, while cooling temperatures are considered from 5°F below the dead state temperature to -10°F. In the actual case, a single heating temperature of 105°F and a single cooling temperature of 55°F are specified.

Because the system output includes both electricity and heat transfers at varying temperatures, the effectiveness ($\varepsilon$) is used as the thermodynamic evaluation parameter. As usual, effectiveness is defined as

$$\varepsilon = \frac{\text{availability increases}}{\text{availability decreases}}$$

which reduces to

$$\varepsilon = \frac{\dot{E}_e + \sum_i \dot{Q}_i (1 - T_0/T_i)}{\dot{A}_f}$$

10 For various definitions, discussion and the merits of effectiveness see, for example, References 3 through 5.
since all inputs are in terms of fuels and outputs are electricity and heat transfers.

2.2 Ideal Systems

In this section, the systems will be assumed to operate reversibly. In addition to the academic interest, the analysis should give indications about the dependence of a particular system effectiveness on the given parameters. Further, the ideal system is unique while there are infinite variations of the real systems, preventing their complete coverage.

The idealization, however, is limited only to the application equipment while the power plant itself remains representative of existing or planned units of the three categories indicated above. The formulation of an ideal system in this manner results in a general conclusion: The ideal BOS always has better performance than the ideal HPS at each given set of conditions. This is because in the ideal BOS the steam that is bled off is utilized in an ideal heating or cooling unit instead of in an imperfect power plant turbine for more electricity. On the other hand, although the ideal HPS would convert its electricity input to the desired thermal output with full effectiveness, it suffers from the losses in the generation of its electricity in the imperfect power plant.
Heating -- BOS (Ideal)

In the ideal BOS, any heating requirement \( \dot{Q}_h \) at temperature \( T_h \) is met by bleeding steam from the power plant turbine of an amount that will provide heating \( \dot{Q}_h \) when the steam condenses at \( T_h \). The steam that is bled off has generated electricity with efficiency \( \eta_h \), the power plant cycle efficiency for saturated exhausting conditions at \( T_h \). Then the total electricity generated for a constant energy (fuel, \( \dot{E}_F \)) input to the power plant is

\[
\dot{E}_e = \eta_C \eta_{SG} \dot{E}_F - \sum_i \frac{\dot{Q}_{h,i}}{(1 - \eta_{h,i})} (\eta_C - \eta_{h,i}). \tag{2-4}
\]

The effectiveness of the overall system is

\[
\epsilon = \frac{\eta_C \eta_{SG} \dot{E}_F - \sum_i \frac{\dot{Q}_{h,i}}{(1 - \eta_{h,i})} (\eta_C - \eta_{h,i}) + \dot{Q}_{h,i}(1 - \frac{T_0}{T_{h,i}})}{\dot{E}_F (A_f / -\Delta H)_F}. \tag{2-5}
\]

11 If the generation of the electricity needed by the heat pump is also permitted to be reversible, then the ideal HPS has the same performance as the ideal BOS, since ideal systems with identical outputs have equal performances. This point is also shown by Glicksman for cooling [60]. However, his statement on page 8, "It must be noted that the use of vapor absorption refrigeration by extracting steam from the turbine gives no advantage over a compression refrigeration system run by electricity if the efficiency of both systems compared to the thermodynamic ideal is the same.", is valid only under the assumption that the electricity generation for this compression refrigeration system is ideal.

12 In this and following equations a number of applications served by a single power plant is assumed and indicated by the summations. However, obviously the upper limit of \( i \) could not be specified and in the numerical presentation the sums are limited to a single term, the effect of different application temperatures being investigated separately.
Noting that $\eta_{SG} \dot{E}_F$ is $\dot{E}_{cyc}$, the energy input to the cycle, and defining the ratio

$$R_{h,i} = \frac{\dot{Q}_{h,i}}{\dot{E}_{cyc}}$$

(2-6)

yields

$$\epsilon = \left[ \eta_C + \sum_i R_{h,i} \left( 1 - \frac{T_0}{T_{h,i}} - \frac{\eta_C - \eta_{h,i}}{1 - \eta_{h,i}} \right) \right] \left( \frac{-\Delta H}{A_f} \right)_F \eta_{SG}.$$  

(2-7)

For nuclear fuels, $(-\Delta H/A_f)_F$ and $\eta_{SG}$ are both usually taken as unity and $\epsilon_{overall}$ reduces to the term inside the brackets of Equation (2-7). Since especially for FFP, a variety of fuels might be considered and since both $(-\Delta H/A_f)_F$ and $\eta_{SG}$ will vary with the fuel type (although both are usually close to unity), the effectiveness values determined here (and denoted by $\epsilon'$) will all be for $(-\Delta H/A_f)_F$ and $\eta_{SG}$ both equal to 1. Hence, by Equation (2-7),

$$\epsilon' = \eta_C + \sum_i R_{h,i} \left( 1 - \frac{T_0}{T_{h,i}} - \frac{\eta_C - \eta_{h,i}}{1 - \eta_{h,i}} \right).$$

(2-8)

From $\epsilon'$ easy conversion to $\epsilon_{overall}$ for any specific fuel can be made.

The term in parenthesis in the equation for $\epsilon'$ is always positive, thus $\epsilon' \geq \eta_C$ i.e. the combined system performance is better than the power plant alone. This can be easily shown by considering the steam that must be bled off to meet the heating requirement and comparing the availabilities of the output with and without bleed off:
The amount of steam in question requires heating of amount 
\( \hat{Q}_h/(1-n_h) \) from the steam generator. The availability output 
if the steam is bledd off is

\[
\hat{A}_{\text{bleed-off}} = \frac{\hat{Q}_h}{(1-n_h)} n_h + \hat{Q}_h (1 - \frac{T_0}{T_h}) 
\]  

(2-9)

and without bleed-off, it is

\[
\hat{A}_{\text{condensing}} = \left( \frac{\hat{Q}_h}{1-n_h} \right) n_C . \]  

(2-10)

Because the power plant turbine is imperfect, the fuller expansion

implies greater loss so that \( \hat{A}_{\text{condensing}} < \hat{A}_{\text{bleed-off}} \) and

\[
\frac{\hat{Q}_h}{(1-n_h)} n_C < \left( \frac{\hat{Q}_h}{1-n_h} \right) n_h + \hat{Q}_h (1 - \frac{T_0}{T_h}) 
\]  

(2-11)

or,

\[
1 - \frac{T_0}{T_h} = \frac{n_C - n_h}{1 - n_h} > 0 , \]  

(2-12)

as indicated above.

The effect of any particular variable on the combined system performance may now be predicted from Equation (2-8). With other variables kept constant, the effectiveness of the ideal BOS with heating will increase with -

(1) \underline{Increased \( n_C \):} Increasing \( n_C \) mainly increases the first term of Equation (2-8), concealing the opposite contribution of the last term. Higher \( n_C \) could result from either more efficient plant types or lower \( T_C \) for a specific plant. Figure 2-1a illustrates
both cases: as $T_C$ is lowered or plant efficiency increased, the system effectiveness increases regardless of the heating temperature. Figure 2-1b, a plot of $e'$ for the combined system divided by $e'_C$ for the power plant itself, shows that the improvement of coupling is greater for low efficiency plants than for high efficiency plants, as would be expected since the losses prevented by coupling increase as efficiency decreases.

(2) Increased $R_h$: As $R_h$ is increased, the effectiveness increases linearly. The magnitude of the improvement is indicated in Figure 2-1c for some selected cases.

(3) Increased $T_h$: Since a constant amount of heating at a higher temperature means greater availability and hence greater prevention of loss in the power plant turbine, initial reasoning indicates increased performance with higher $T_h$. But $T_h$ also affects the performance implicitly via $\eta_h$, and this effect is in the opposite direction, making the net result difficult to predict. Figure 2-1d indicates that as $T_h$ increases, the performance increases at a diminishing rate and finally even decreases slightly, a manifestation of the efficiency-curve slopes of the power plants.
Heating -- HPS (Ideal)

The ideal heat pump system receives electricity and condenser water at $T_C$ from the power plant to meet each heating requirement of $Q_h$ at $T_h$. The ideal heat pump coefficient of performance for these conditions is

$$COP = \frac{1}{(1-T_C/T_h)} \quad (2-13)$$

requiring electricity of an amount

$$\dot{E}_{HP} = \Sigma \frac{\dot{Q}_{h,i}}{COP_i} = \Sigma \dot{Q}_{h,i}(1 - T_C/T_h,i) \quad (2-14)$$

for the heat pump. The net electricity from the power plant for other uses is

$$\dot{E}_e = \eta_C n_{SG} \dot{E}_F - \Sigma \dot{Q}_{h,i}(1 - T_C/T_h,i) \quad (2-15)$$

and the effectiveness of the combined system is

$$\varepsilon = \frac{\eta_C n_{SG} \dot{E}_F - \Sigma \dot{Q}_{h,i}(1 - T_C/T_h,i) + \Sigma \dot{Q}_{h,i}(1 - T_0/T_h,i)}{(E_F) \frac{A_f}{(-\Delta H)_F}} \quad (2-16)$$

Therefore,

$$\varepsilon' = \eta_C + (T_C - T_0) \Sigma \frac{R_{h,i}/T_h,i}{(2-17)}.$$
Note that if the ideal heat pump system had its source at $T_0$ rather than $T_C$, the second term in the equation for $\epsilon'$ would be zero. But when the temperature of the power plant condenser is made available as the source, an improvement in performance is obtained. The simplicity of Equation (2-17) allows predictions of the effect of variable changes to be easily made. For other variables kept constant, the effectiveness of the ideal HPS will increase with -

(1) **Increased** $n_C$: Changing to more efficient power plants increases the effectiveness by simply increasing the first term of Equation (2-17).

(2) **Decreased** $T_C$: $T_C$ is a very important variable for the HPS. Decreasing $T_C$ will reduce the second term of Equation (2-17) but the accompanying improvement of the power plant efficiency is larger and results in a net increase in effectiveness as shown in Figure 2-1e. Figure 2-1f illustrates how the relative improvement of the HPS over the single power plant decreases as the condenser temperature is reduced.

(3) **Increased** $R_h$: For the HPS, since some heating improved the performance, increased heating will improve it even more.

(4) **Decreased** $T_h$: When the temperature difference that the heat pump must cover is reduced, with decreasing $T_h$, the advantage of having a source at $T_C$ rather than $T_0$ gains weight and results in better overall effectiveness. Figure 2-1e illustrated this with the $T_h = 105^\circ F$ curve lying above the $T_h = 330^\circ F$ curve of the same plant.
Heating -- Comparison of BOS and HPS

As seen, the variables $n_c$, $T_c$ and $R_h$ affect both systems in a similar manner; but the HPS performance is inversely related to $T_h$ while the BOS performance increases directly with $T_h$ for a large range.

As stated previously, the ideal BOS is always slightly better in performance than the ideal HPS as a result of less use of an imperfect power plant turbine. The two systems are quantitatively compared on Figure 2-1g. The FFP falls on an almost horizontal line so only a representative section at $T_c = 65^\circ F$ is indicated. Figure 2-1g illustrates that the superiority of BOS is much more pronounced for lower efficiency plants, higher $T_h$, and $R_h$. Calculations show the performance superiority of BOS over HPS is less than 3% for a heating temperature of $110^\circ F$ and less than 12% for a heating temperature of $300^\circ F$ for all plant types considered and all $R_h$ up to 0.4.
Fig. 2-1(a-g). The performance of the ideal heating systems as related to $T_C$, $T_h$, and $R_h$. The figures a through d present the BOS, while e and f are for the HPS. The two systems are compared in g. Note $\epsilon'_i$ indicates the performance of an uncoupled power plant.
Cooling -- BOS (Ideal)

In the ideal BOS, a cooling requirement \( Q_c \) at a temperature \( T_c \) is met by bleeding steam from the power plant turbine to operate an ideal absorption cooling system.\(^\text{13}\) The steam is bled off at a somewhat arbitrary temperature \( T_{BO} \). Since the system is ideal the heating temperature in the absorption unit, \( T_{ab} \), is equal to \( T_{BO} \). Then the amount of steam to be bled off and condensed at \( T_{ab} \), yielding \( Q_{ab} \), is specified since it must have the same availability as the cooling \( Q_c \) at \( T_c \):

\[
\dot{Q}_{ab} (1 - \frac{T_0}{T_{ab}}) = \dot{Q}_c (1 - \frac{T_0}{T_c})
\]

(2-18)

and therefore\(^\text{14}\)

\[
\dot{Q}_{ab} = \dot{Q}_c \left( \frac{T_c - T_0}{T_{cr}} \right) \left( \frac{T_{ab} - T_0}{T_{ab} - T_0} \right).
\]

(2-19)

Now, the analysis reduces to the ideal BOS with heating \( Q_{ab} \) at \( T_{ab} \) and the net electrical generation is

\[
E_e = \eta_c \eta_{SG} E_F - \sum_{i} \dot{Q}_c_i \left( \frac{T_c - T_0}{T_{ab} - T_0} \right) \left( \frac{T_{ab}}{T_c} \right) \left( \frac{\eta_c - \eta_{ab}}{1 - \eta_{ab}} \right)
\]

(2-20)

\(^{13}\)Note, \( Q_c \) is a negative quantity since the sign convention chosen here assigns a positive number to a heat transfer into a system.

\(^{14}\)Note that from Equation (2-18) the coefficient of performance of the ideal absorption unit, is given as

\[
\text{COP}_{ab} = \frac{-Q_c}{\dot{Q}_{ab}} = \frac{(T_{ab} - T_0)}{T_c} \cdot \frac{T_c}{(T_0 - T_c)}
\]

(2-18a)

Equation (2-19) then can simply be viewed as \( \dot{Q}_{ab} = |Q_c|/\text{COP}_{ab} \).
while the effectiveness of the combined system is

$$\varepsilon' = \eta_C + \sum R_i c_i, \sum \left[ \left( \frac{T_c - T_0}{T_c} \right) \left( \frac{T_{ab}}{T_c} \right) \left( \frac{\eta_C - \eta_{ab}}{1 - \eta_{ab}} \right) + \frac{T_0}{T_c} - 1 \right] \quad (2-21)$$

where $R_c$ is defined as

$$R_c \equiv - \frac{\dot{Q}_c}{\dot{E}_{cyc}}. \quad (2-22)$$

Thus, as in the heating with the ideal BOS, the overall effectiveness is equal to the power plant efficiency plus a positive number, resulting in improved performance over the power plant.

The effects of each of the variables can be examined in a manner similar to heating. The variables $\eta_C$ and $R_c$ must have influences similar to $\eta_C$ and $R_h$ in the heating. The influences are in the same direction, but the improvement over the power plant performance for an $R_c$ of say 0.3 is less than was the case for the heating at comparable conditions ($T_h = T_{ab}$, $R_h = 0.3$, same $T_C$) because the COP of the absorption cooling unit is always greater than 1 for the cases studied and therefore $\dot{Q}_{ab}$ is less than $\dot{Q}_h$.

Figure 2-2a shows the ratio of effectiveness for the combined system to that of the power plant.

Further, when other variables are held constant, the performance of the ideal BOS with cooling will increase with -

1. **Decreased $T_c$:** Lowering $T_c$ at a fixed $\dot{Q}_c$ increases the availability of cooling and hence the steam bleed-off, resulting in an improvement in the overall performance. Figure 2-2b shows the increased performance with decreased $T_c$. 

(2) Decreased \( T_{ab} \): In Equation (2-21), \( T_{ab} \) appears twice explicitly as well as several times implicitly via \( \eta_{ab} \) which increases as \( T_{ab} \) decreases. In the cases considered, \( T_{ab} \) is substantially larger than \( T_0 \) and therefore the increase of \( T_{ab}/(T_{ab} - T_0) \) as \( T_{ab} \) is decreased is outweighed by the decrease of \( (\eta_c - \eta_{ab})/(1 - \eta_{ab}) \), resulting in a net improvement in the overall system performance as illustrated in Figure 2-2b.

Recall in the heating case, an increase of \( T_h \) improved the overall performance. But an increase of \( T_{ab} \) does not improve the performance for cooling. The reason for the difference is that in heating as \( T_h \) increases, the availability of heating increases and \( \dot{Q}_h \) stays constant, but in cooling as \( T_{ab} \) increases while \( \dot{Q}_c \) and \( T_c \) stay constant, the availability of cooling (and therefore bleed-off) remains constant and \( \dot{Q}_{ab} \) decreases.

Cooling -- HPS (Ideal)

In the ideal heat pump system, a cooling requirement \( \dot{Q}_c \) at a temperature \( T_c \) is met by operating the heat pump as a refrigeration system with electricity from the power plant and with a sink temperature equal to the \( T_0 \). The net electricity from the power plant for other uses is

\[
\dot{E}_e = \eta_c \eta_{SG} \dot{E}_F - \dot{Q}_c \left(1 - \frac{T_0}{T_c}\right).
\]

(2-23)
But, the availability of the cooling is $Q_c (T_c - T_0)/T_c$ and hence, the overall availability output of the system is equal to $\eta_C \eta_{SG, F}$ and the effectiveness is

$$\varepsilon' = \eta_C.$$  \hspace{1cm} (2-24)

This simple result, independent of all the properties of the cooling load and the heat pump is predictable since here HPS ideally converts work (electricity) to a heat transfer (cooling) of equal availability.  

Cooling -- Comparison of BOS and HPS

Because the performance of the ideal HPS is the same as the power plant, the comparisons of the ideal BOS and ideal HPS for cooling reduces to the comparison of the ideal BOS with the power plant as given previously.

---

15 In the ideal HPS for heating, there was an improvement in the combined system effectiveness due to the utilization of some of the availability of the power plant condenser cooling water outlet which otherwise is wasted.
Fig. 2-2 (a,b). The performance of the ideal BOS for cooling as related to variables $T_C$, $T_c$, $R_c$, and $T_{BO}$. Note that the performance of the ideal HPS during cooling equals that of the power plant itself.
2-3. **Actual Systems**

The analysis of the ideal systems as presented in the previous sections shed light on the effect of the variables involved in terms of direction and degree of change of the overall system performance as well as the limit of any actual performance. However, any actual system may differ substantially from the ideal so that the degree and even the direction of change of the system performance with a particular variable may differ from that of the ideal case.

Variations in the performance of actual systems are so wide that selection of a particular system for study is necessarily somewhat arbitrary and of limited generality. Here the BOS selected for study is patterned after one examined by Oak Ridge National Laboratory [57]. The steam bleed-off occurs at two temperatures, 260°F and 310°F, with half the energy supplied at each temperature. The heating temperature is selected to be 105°F as an average for space heating and domestic water heating. Since the cooling temperature is constant at 55°F, the absorption units have been assumed to have a constant COP of 0.70.

The HPS selected contains a heat pump whose performance is related to the power-plant condenser temperature by Equation (2-25) for heating and Equation (2-26) for cooling:

\[
\text{COP} = 3.95 + 0.055 \ (T_c - 60) \quad (2-25)
\]

\[
\text{COP} = 4.89(T_c - 90)(T_c - 120)/675 - 3.70(T_c - 75)(T_c - 120)/450 + 2.32(T_c - 75)(T_c - 90)/1350 \quad (2-26)
\]
Equation (2-25) was constructed from ASHRAE [61] data with the conditions $T_h = 105^\circ F$, the temperature difference in the piping loop water is $20^\circ F$ across the evaporator, and the approach temperatures are all $5^\circ F$. Equation (2-26) is based on the same data and conditions, but here $T_c = 55^\circ F$ and $T_h$ does not apply.

Heating -- BOS (Actual)

Figures 2-3a and b illustrate the dependence of the actual BOS system on the variables condenser temperature, power plant type (power plant efficiency) and $R_h$. Heating temperature variation was not considered for the actual system. Figure 2-3a illustrates a decrease in overall system performance as $T_c$ increases and as power plant efficiency decreases by changing power plant types. This trend is the same as for the ideal BOS system and follows from the same reasoning.

Figure 2-3a further reveals a decrease in overall system performance as $R_h$ is increased. Also, Figure 2-3b shows the effectiveness of the actual BOS system to be lower than the power plant itself. Both of these effects arise from a common cause. In the ideal system, the performance of BOS was greater than the power plant alone because of the elimination of some losses in the turbine. However, as the performance of the heating system decreases from ideal, some value will be reached where the losses in the heating system are equal to the losses that would have been incurred had the steam been fully expanded rather than bled off.
Then the effectiveness of the overall system and the power plant itself will be the same. As the performance of the heating system decreases past this point (as turns out to be the case for the actual BOS selected here) the performance of the overall system must be poorer than the power plant in proportion to the value of $R_h$.

Heating -- HPS (Actual)

The actual HPS overall performance is indicated in Figure 2-3c. As in the ideal case, the performance increases as $\eta_C$ increases either from decreasing $T_C$ or by changing power plant type. Similar to the actual BOS, the actual HPS has a poorer performance than the power plant alone and consequently decreasing performance as $R_h$ increases as shown in Figure 2-3c. Again, heating temperature variation was not investigated.

Heating -- Comparison of BOS and HPS

Figure 2-3d illustrates a direct comparison of the actual BOS and HPS for the three power plant types, and one $R_h$ as a function of $T_C$. The BOS is superior for the LWR and FBR type plants for low condenser temperatures. The HPS increases in superiority as $T_C$ increases and as higher efficiency power plants are specified. Similar behavior continued for lower $R_h$ values.

A specification that can be quite valuable is the COP which the heat pump must have for the HPS to have the same overall
Fig. 2-3 (a-d). The performance of the actual heating systems as related to $T_C$, $T_h$, and $R_h$. 
performance as the BOS. This "critical COP" is plotted in Figure 2-4a. When the heat pump has a COP greater than the critical COP the HPS will have better performance than the BOS. Also shown on Figure 2-4a is the actual COP for the HPS investigated here.

Cooling -- BOS (Actual)

Figure 2-5a illustrates the performance of the actual BOS system, during cooling, as a function of condenser temperature, power plant type, and \( R_c \). All curves indicate that as \( R_c \) increases the overall system performance decreases. This is a result of the same reason that the performance decreased as \( R_h \) increased for the BOS system in heating.

For a given power plant, the performance corresponding to lower condenser temperature decreases at faster rates for increasing \( R_c \) resulting in crossing of the curves since at \( R_c = 0 \) the performance at the lower condenser temperature is higher. This faster decrease at lower \( T_c \) results largely from the tying of \( T_0 \) to \( T_c \). As \( T_c \) decreases from 120°F to 90°F, \( T_0 \) decreases from 90°F to 60°F and with the cooling temperature held constant at 55°F, the availability of the cooling requirement decreases considerably resulting in much poorer thermodynamic performance of the cooling system which has an essentially constant COP. Also, at low \( T_c \) the loss from bleed-off is higher since \( T_{ab} \) is constant.
Cooling -- HPS (Actual)

The performance of the actual HPS in cooling has the same type of performance change as in heating. The specific performance values are given in Figure 2-5b.

Cooling -- Comparison of BOS and HPS

Figure 2-5c compares the actual BOS and HPS in cooling operation for the three power plant types and one $R_c$ as a function of $T_C$. Another $R_c$ value is presented for the LWR type power plant to illustrate its influence for this overall comparison. The HPS is superior over a considerable portion of the range investigated.

Lackey [62] has previously investigated the cooling aspect of the BOS and the HPS (compressive refrigeration) and considered performance variation for various steam bleed-off conditions. He showed for the actual system considered that the compressive refrigeration had slightly better performance with greater advantage for the higher efficiency plants, but that the relative performance depended highly upon steam bleed-off condition as well as whether the absorption system was steam heated or hot water heated.

As in heating, a comparison can also be presented in terms of "critical COP". Figure 2-4b shows the critical COP for cooling applications.
Fig. 2-4 (a,b). Critical coefficients of performance in heating or cooling that equalize the overall effectiveness of the two systems.
Fig. 2-5. (a-c). The performance of the actual cooling systems as related to $T_C$, $T_C$, and $R_e$. The performances shown individually in a and b, for the BOS and the HPS, are compared in c.
2-4. Conventional Systems

Since both the BOS and the HPS are developed as alternatives to systems presently in common use, it is desirable to analyze the conventional systems in a similar manner. This section presents a brief examination of some of the representative conventional systems.

Heating -- Electrical resistance (AES)

To deliver $\dot{Q}_h$ at $T_h$, the system requires electricity of an amount equal to $\dot{Q}_h$. Then, in terms of $R_h$, the effectiveness of this system becomes

$$\varepsilon' = \eta_C - \sum_i R_{h,i} \frac{T_0}{T_{h,i}} (1 - \frac{T_0}{T_{h,i}})$$

$$= \eta_C - \sum_i R_{h,i} \frac{T_0}{T_{h,i}} ,$$

(2-27)

a simple result with predictable features. To illustrate, it is plotted in Figure 2-6a for $T_h = 105^\circ$F and $R_h = 0.3$. (This $T_h$ and $R_h$ values were also used in Figure 2-3 for the actual BOS and HPS analysis.)

Heating -- Usual heat pump, with source at $T_0$ (UHP)

This differs from the resistance heating case in that the electricity required is reduced by a factor equal to the COP, thus

$$\varepsilon' = \eta_C - \sum_i R_{h,i} \left( \frac{T_0}{T_{h,i}} + \frac{1}{\text{COP}_i} - 1 \right) .$$

(2-28)
With an ideal heat pump, this would reduce to $\varepsilon' = \eta_C$, but with the practical equipment the sum in parenthesis is always positive because the actual COP is always less than the ideal COP; hence addition of a non-ideal heat pump lowers the system performance as is expected. However, since $1/\text{COP}$ is usually less than 1, the sum in the paranthesis is less than $T_0/T_h$, and the performance with heat pumps is superior to that with electrical resistances - a well known but frequently ignored conclusion. Figure 2-6a also illustrates this performance. In the calculations, the COP values are based on the data also used for the actual HPS (Equations 2-25 and 26).

**Heating -- Fossil Fuel Furnace (UFS)**

Noting that the energy input to the power plant is reduced by $Q_h/\eta_f$, $\varepsilon'$ becomes

$$\varepsilon' = \eta_C - \sum_i R_{h,i} \left( \frac{T_0}{T_{h,i}} + \frac{\eta_C}{\eta_{f,i}} - 1 \right). \tag{2-29}$$

Clearly, the furnace is better than (i) the resistance heating as long as $\eta_f > \eta_C$, and (ii) the heat pump, if $\eta_f > \eta_C \cdot \text{COP}$. (Both $\eta_f$ and COP are influenced by $T_h$.) Examining Equation (2-29) further indicates that, with this setup it is possible to achieve an improvement even over an uncoupled plant since the sum in the parenthesis could be negative under certain (but rare) cases. This is possible because the furnaces alter the system while the electrical resistances and heat pumps (in UHS) simply expand it by converting one useful output to another.
The performance of the UFS is plotted also in Figure 2-6a by assuming a furnace efficiency of 0.77. For this typical case, the performance falls somewhere in between the AES and UHS, since
\[ \eta_C \cdot \text{COP} > \eta_F > \eta_C. \]

**Heating - Comparison**

To make the comparisons directly, the ratios of the conventional system effectiveness to that of the HPS are plotted in Figure 2-6b. Thus, this figure compares the conventional systems with the HPS, and through it, with each other and the BOS. The variation with \( R_h \) is not shown since the ratio would almost linearly decrease from 1.0 at \( R_h = 0.0 \) to the given values at \( R_h = 0.3 \). With the increased \( T_C \), the effectiveness-ratio displays a decrease for the AES (because while all systems lose in generating electricity at a higher \( T_C \), the HPS regains some with its improved COP), a milder reduction for the UFS (since there is less electricity generation), and essentially no change for the UHS (as \( T_0 \) follows up the \( T_C \)). Diverting some of the fuel to the furnaces benefits the system with the lowest efficiency most; therefore, the LWR curve is above FBR or FFP curves of the UFS. A plot of \( \varepsilon'/\varepsilon'_C \) would also indicate these points in a very similar manner. As expected, the very poor performance of the resistance heating is once again apparent in this figure. An extention of the same point is the very low \( R_{h-\text{lim}} \) for the AES (as given in the Appendix B, \( R_{h-\text{lim}} = \eta_C < 0.4 \)). The superiority of the HPS over the UHS is very slight and is due to the utilization of the availability of loop water in the HPS.
Fig. 2-6. The performance of the conventional heating systems are displayed in Figure a, and compared with that of the HPS in Figure b.
Close performance of the UHS and the HPS is not surprising. The purpose of advocating an HPS type system is not this slight advantage nor the utilization of some power plant waste heat, although decreasing the thermal pollution at the power-plant site is a significant side benefit. The attempt is to expand the heat pump utilization by providing a dependable thermal source/sink, and thus overcome one of the factors (namely the lack of good sources or the problems associated with the poorer sources) that suppressed their usage. Circulating water from any source could be a successful operation (even if it had no availability) as long as its temperature is not too close to freezing. The HPS investigated here simply benefits additionally from the warmer temperature of the water available from thermal-electric power plants.

Cooling -- Fossil Fuel Furnace + Absorption Refrigeration (UFS)

Only this system is considered here, because the direct cooling methods are not widespread and the HPS above have used a sink at $T_0$, thus encompassing the ordinary "air-conditioners" of the AES or UHS. Noting that the energy demand of the furnaces is $\sum_i \dot{Q}_{c,i} / (\eta_f \cdot \text{COP}_{ab,i})$ results in

$$
\varepsilon' = \eta_C - \sum_i R_{c,i} \left( \frac{\eta_C}{\eta_f \cdot \text{COP}_{ab,i}} + 1 - \frac{T_0}{T_{c,i}} \right) \cdot (2-31)
$$

Clearly, the performance improves with increased $\eta_C$ or $T_c$ and decreased $R_{c}$. The effects of these and $T_c$ are shown in Figure 2-7a for $\eta_f \cdot \text{COP}_{ab} = 0.6$ and $T_c = 55^\circ F$. (Hence, this figure is comparable
Fig. 2-7 (a,b). Meeting space cooling demands with a furnace - absorption unit combination. The performance displayed in Figure a is compared with that of HPS in Figure b.
with Figure 2-5 for the actual BOS and HPS.) At increased $T_C$ the effects of low $\eta_C$ and higher $T_0$ (meaning higher availability for cooling at a constant $T_C$) oppose each other. Equation (2-31) indicates that with higher $R_c$, the effect of $\eta_C$ is weaker and the effect of $T_0$ is stronger; but as seen in Figure 2-7a, the net effect of $T_C$ is hardly noticeable.

Figure 2-7b compares the UFS with the actual HPS (or UHS), and through the Figure 2-5c, with the BOS. Figure 2-7b shows that, as with heating, the furnace does better for low-efficiency power plants, but that the heat pump is still noticeably superior. The UFS would surpass the HPS only if $\eta_F \cdot \text{COP}_{ab} > \eta_C \cdot \text{COP}_{HP}$. In fact, here the furnace compares even poorer than it did in heating because $[\eta_F \cdot \text{COP}_{ab} / \eta_C \cdot \text{COP}_{HP}]_C < [\eta_F / \eta_C \cdot \text{COP}_{HP}]_h$.

2-5. Conclusions

Even though the ideal BOS is found to always have slightly better performance than the ideal HPS, nevertheless the analysis revealed quite comparable performances. Up to $R_h$ and $R_c$ of 0.3, the maximum superiority of BOS is 2.5% for heating at 105°F, 7.4% for heating at 300°F, and less than 10% for cooling at -10°F.

In the actual systems, HPS has much lower COP and still suffers from the losses in providing electricity to its heat pumps. However, the performance of the BOS is also reduced by lower absorption unit
COP, and by bleeding at much higher than the heating temperatures. Thus, in the actual case, the BOS is superior to the HPS (less than 3% in heating, 6% in cooling) only for certain applications, while the HPS is found to have slightly better performance than BOS for a large portion of the range of the parameters investigated. Up to \( R_h \) and \( R_c \) of 0.3, the maximum HPS superiority is about 5% in heating, and as much as 14% in cooling. Although other heating and cooling temperatures were not analyzed, it is expected that as \( T_h \) is increased, the BOS would improve relative to the HPS; but if \( T_c \) is decreased, the COP of the refrigeration systems in both BOS and HPS would decrease causing only little change in their relative position. Finally, it is seen that both the BOS and the HPS are thermodynamically preferable to the conventional systems, particularly the electrical resistance heaters.

For most applications, parameters involved will vary throughout the year. With these changes the operation should swing between the regions where either the BOS or the HPS is superior. Rough estimates of the yearly behavior could be obtained from above analysis by separately considering all basic variations of an application. However, due to the closeness of the performance of the BOS and the HPS, a more definite evaluation necessitates a detailed examination of variations in load and other parameters such as the piping losses, pumping requirements, "free" heating with HPS when heating occurs concurrently with cooling, etc. Such an examination is undertaken next for the domestic water heating and comfort heating and cooling of urban communities.
CHAPTER III

URBAN APPLICATIONS

The previous chapter presented the general performance of the two systems under consideration. One important parameter that emerged was $R$, the ratio of the thermal demand to the gross energy rate available to the system. In addition to its influence on the performance, this parameter is significant as an indicator of the extent of the benefit from the use of these systems. That is, the advantages such as savings in energy or decrease in pollution reach more significant amounts for higher $R$ values. Since the energy input to large power plants can be up to several thousand megawatts, rather large thermal demands are needed. These are also desired for economics, as the specific cost of distribution systems and the like will decrease with increased usage. Space and water heating in nearby urban centers offer one possible high-$R$ application. As indicated before, the same systems are also capable of beneficially serving for space cooling: the BOS with the use of absorption systems and local cooling towers eliminates some energy rejection at the power plant; and the HPS offers to the consumer a dependable sink with either local cooling towers or the power plant cooling system.

The intent of this chapter is to evaluate in detail the performances of these systems when adopted to several cities differing in location and population. Using the assumed city models, actual weather data and related information, the design, yearly total and
monthly average requirements are developed as well as the design of a distribution network. These together with equipment data determine the combined system performance. Here, the performance is evaluated in two parts: (i) the electricity available for other uses after the thermal demands of the city are met, and (ii) the net cooling requirement at the power plant site. The analyses for both systems, and an un-coupled plant are essential in determining the economical potential with respect to more conventional systems and the suitability of integration with other users.

The basics of the systems examined here resemble the practical systems of the previous chapter. Nevertheless, a new evaluation is made since much more detailed and accurate information is sought, including domestic water heating performance, and such points as the savings potential from interaction of concurrent heating and cooling and the effect of the plant-city distance in terms of the loop energy loss and pumping cost. Further, because the input and output types are identical among all systems, the comparison need not be in effectiveness, but can be in more practical and lucid output terms as indicated above.

3-1. Communities Served

After a limited search of trends and suggestions about community development, the basic layout of the city as developed by ORNL for their similar study [57] was selected. However, here two different sized, smaller cities are considered: one with a total population
of 33000 of which 21,500 live in an apartment area that will be served by the energy system, and another with a total population of 200,000 of which 129,000 live in apartments served by the energy system. For convenience, these cities will be referred to by their served population (the 21,500 city and the 129,000 city). The overall population only affects the system indirectly by changing the size of the downtown, and the minimum distance from the power plant. The smaller city will probably be applicable in a greater number of circumstances since nearness to a power plant is advantageous and since systems mostly applicable to new urban settings are considered.

Both cities will be analyzed for a mild Willamette Valley (Portland, Oregon) climate.\textsuperscript{16}

The remaining two parameters, besides population, and location (climate), that characterize the community are population density and plant-city distance. The density is being taken as a fixed value within each: for the 129,000 city it is kept the same as ORNL's reference city, and for the 21,500 city the density is selected as two-thirds of the other to partially reflect the changes in going to a smaller community. The plant-city distance is first chosen as the shortest distance possible under current siting

\textsuperscript{16}As indicated previously, this phase of the research was sponsored by local utilities as a part of an integrated analysis considered for application with the several nuclear plants now in the planning stage. In that work [58], two additional climates were considered: an even milder coastal (Astoria), and fairly continental Eastern Oregon (Pendleton). The analysis and results for these locations are presented here in a summarized form.
practices and trends. Data presented in Reference 59 indicates 2.5 miles for the 21,500 city and 4.5 miles for the 129,000 city as the minimum distance from power plant to the city center. In the economic analysis the role of greater distances may easily be considered.

The conceptual layouts of these two cities are shown in Figures 3-1 through 3-6.\textsuperscript{17} Figures 3-1 through 3-4 show the complete layout for the 129,000 city while Figures 3-5 and 3-6 indicate the changes for the 21,500 city.

Figure 3-1 shows the general layout of the city, which consists of six one-mile square units of high-density apartments and a two square-mile downtown and college area. The high-density apartments and the downtown constitute the served area. The power plant is located a distance of 4.5 miles from the city center (3 miles from the closest edge of high-density residential area).

The composition of the downtown and college area is detailed as shown in Figure 3-2. Figure 3-3 shows a typical high-density residential mile-square for the 129,000 city. Figure 3-4 represents an apartment-block (A) which consists of 8 apartment buildings each with an enclosed floor area of 8,400 square feet per story. The apartments are to be 3 stories in the 129,000 city and 2 stores in the 21,500 city. The symbols and associated square footages are tabulated in Table 3-1.

\textsuperscript{17}To have a better continuity of the text, the figures of this chapter are collected at the end of the chapter, on pages indicated by the List of Figures.
The general layout of the 21,500 city is presented in Figure 3-5. The total served area is slightly over a mile-square. The layout chosen reflects a location along a geographical line such as a river, ocean, or freeway. Figure 3-6 details the served portion of the city, again the symbols are as in Table 3-1. The symbol A2/3 represents the apartment block in the 21,500 city containing two-thirds the square footage of enclosed space of the apartment block in the 129,000 city. Similar interpretation is to be given to the fractions in the downtown area.
TABLE 3-1. Symbols and Unit Sizes For City Layout.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Area (ft^2)</th>
<th>Number of Units in City of 129,000</th>
<th>Number of Units in City of 21,500</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Apartment</td>
<td>201,600</td>
<td>192</td>
</tr>
<tr>
<td>S</td>
<td>Local Shopping</td>
<td>25,300</td>
<td>36</td>
</tr>
<tr>
<td>ES</td>
<td>Elementary School</td>
<td>90,000</td>
<td>24</td>
</tr>
<tr>
<td>SS</td>
<td>Secondary School</td>
<td>312,000</td>
<td>6</td>
</tr>
<tr>
<td>RI</td>
<td>Research Institute</td>
<td>260,000</td>
<td>1</td>
</tr>
<tr>
<td>C</td>
<td>College/block</td>
<td>65,000</td>
<td>14</td>
</tr>
<tr>
<td>H</td>
<td>Hospital</td>
<td>200,000</td>
<td>2</td>
</tr>
<tr>
<td>SO</td>
<td>Shopping-Office Building</td>
<td>272,000</td>
<td>17 1/2</td>
</tr>
<tr>
<td>D</td>
<td>Department Store</td>
<td>120,000</td>
<td>2</td>
</tr>
<tr>
<td>M</td>
<td>Motel/Hotel</td>
<td>92,000</td>
<td>2</td>
</tr>
<tr>
<td>T</td>
<td>Amusement Center</td>
<td>150,000</td>
<td>1 1/2</td>
</tr>
</tbody>
</table>

(1) The city of 21,500 actually has 48 two-floor apartment blocks, the equivalent area of 32 three-floor apartment blocks.
3-2. **Energy Requirements: Design and Yearly Total**

To design the thermal system and the distribution network for an area, it is necessary to calculate the maximum energy requirements of the space heating and cooling as well as the corresponding energy requirements for the domestic water heating. Since the city water temperature changes gradually, the daily variation of the water heating requirement is too weak to shift the system peak. Energy requirement for domestic water heating at different hours during a day is estimated from the data given in Reference 63. To obtain the maximum energy demand for space heating and cooling, the heating and cooling loads found as outlined by ASHRAE [64] are adjusted by the efficiency of the local equipment.

To determine the economics of a particular thermal system, it is further necessary to estimate the energy use throughout the year. One method of doing this would be to integrate a load balance equation divided by the local equipment efficiency, with respect to time, over an extended period such as one year. This method is referred to as computerized energy analysis of buildings, since a computer must be used in this type of analysis for any building of substantial size. Another procedure that has been used combines the modified degree-day method and the cooling load-factor method: The energy requirement for heating during the year is estimated as

\[ Q = \left[ \frac{\text{Design Heating Load}}{\text{Design Temp. Diff.}} \right] [24] [\text{Degree Days}] C \]
where the constant $C$ is an adjustment factor to compensate for different ratios of energy use to design load for different types of buildings. And the yearly cooling load is taken to be equal to the design (maximum) cooling load occurring for a certain fraction of time per year. The fraction of time - estimated for the particular building and location - is the load factor.

The above discussion indicates how the design and yearly thermal energy requirements of a space may be determined. In this work, it is desired to determine the thermal energy requirements of an entire city. To accomplish this, any one of a number of different models might be considered. The following briefly-described models span the spectrum from the very detailed to the very approximate.

A. Each building in the city is specified with regard to construction, occupancy, and operating schedule. Each building is analyzed to determine its design and yearly loads. The total city requirement is determined as the sum of the individual building requirements, with adjustment for diversity. This model should give accurate results, but only for the specified city selected. Whether the results represent an "average" city, of the population in question, however, depends upon how close the individually specified buildings represent averages of building construction, occupancy, and operating schedule. Perhaps more important, this model would require tremendous amounts of time and money to evaluate, even with the fastest computers.
B. The city consists of various quantities of a relatively small number of different types of buildings. Each of the different building types is completely specified for construction, occupancy, and operating schedule. Each typical building then is analyzed for design and yearly loads. The total city requirement is the sum of the requirements of the individual building types times their respective quantity within the city, with adjustment for diversity. This model could also give quite accurate results but, again, whether this represents an "average" city depends on how well each of the building types selected represent the averages of those buildings included in the given type.

C. The city consists of various quantities of a relatively small number of different types of buildings. The design and yearly loads for each building type are specified by taking the average of a number of buildings which the given type is to represent. This model can be very accurate or very approximate depending upon how many buildings are analyzed to obtain the average values to be used with each type, but should give reasonable results for an "average" city, if several buildings are considered in each category.

D. The city consists of a given amount of floor space, and the thermal energy requirements - design and yearly use - are specified by assuming given values which are to represent properly weighted averages throughout the city.

Model "C" is the choice for this study for the following reasons: The reason for wanting the city model is to determine representative energy requirements for an average city of a selected
size and population. So, "average" requirements are desired. Models "A" and "B" require an "average" city to be completely specified and would require too much effort to be practical for getting average energy requirements. And Model "D" appears to be too approximate for practicality since commercial buildings and apartments vary so much in their energy requirements. Also, changing the mixture of the commercial and residential buildings within the city would be difficult if Model "D" were used.

Space Heating and Cooling. For the city model chosen as outlined above the design values for heating and cooling for each building type are given in Table 3-2.

Since an entire city rather than a specific building is being considered, the modified degree-day method of estimating energy use for heating, and the load-factor method of estimating energy use for cooling are considered to be sufficiently accurate. The value of the C factor in the modified degree day method is taken as 0.71 for apartments and 0.58 for all others. The value of the load factor is taken 0.05 for apartments, 0.09 for elementary schools and 0.18 for all others.

18 The factor 0.71 is based on actual data of energy use of electricity heated apartments in Portland. The factor 0.58 is based on Portland General Electric Company computer analysis of office type buildings ranging from 5,000 to 100,000 ft², and extended here to other building types.

19 The factor 0.05 is based on an estimate given by utilities in Portland. And the factor 0.18 is based on the computer analysis also used above. The value for ES is reduced since they are assumed closed during three summer months.
TABLE 3-2. Heating and Cooling Design Values for Portland, Oregon.

<table>
<thead>
<tr>
<th>Unit</th>
<th>Heating (Btu/hr ft(^2))</th>
<th>Cooling (ft(^2)/ton)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Apartments (A)</td>
<td>25(1)</td>
<td>600(2)</td>
</tr>
<tr>
<td>Shopping (S)</td>
<td>22(5)</td>
<td>275(5)</td>
</tr>
<tr>
<td>Elementary Schools (ES)</td>
<td>30(3)</td>
<td>400(3)</td>
</tr>
<tr>
<td>Secondary Schools (SS)</td>
<td>30(3)</td>
<td>400(3)</td>
</tr>
<tr>
<td>Research Institute (RI)</td>
<td>22(5)</td>
<td>375(5)</td>
</tr>
<tr>
<td>College (C)</td>
<td>30(6)</td>
<td>400(6)</td>
</tr>
<tr>
<td>Hospitals (H)</td>
<td>42(7)</td>
<td>300(7)</td>
</tr>
<tr>
<td>Shopping-Office (SO)</td>
<td>22(4)</td>
<td>375(4)</td>
</tr>
<tr>
<td>Department Stores (D)</td>
<td>22(5)</td>
<td>375(5)</td>
</tr>
<tr>
<td>Motels (M)</td>
<td>29(8)</td>
<td>450(8)</td>
</tr>
<tr>
<td>Amusement (T)</td>
<td>22(5)</td>
<td>375(5)</td>
</tr>
</tbody>
</table>

(1) Conservative estimate based on actual heating requirements of electrically heated apartments in the Portland area.

(2) Estimate based on limited survey of apartments, and utility personnel "Rule-of-thumb" design value. It is interesting to note that Megley [22] reports apartment design values of 22 Btu/hr ft\(^2\) for heating and 594 ft\(^2\)/ton for Boston, Mass.

(3) Conservative estimate based on a limited survey of elementary, junior high, and high schools in the Portland area.

(4) Conservative estimate from Portland General Electric Computer analysis of office buildings ranging from 5,000 to 100,000 ft\(^2\).

(5) Assumed the same as the building in the "Shopping-Office" category.

(6) Assumed the same as the elementary and secondary schools.

(7) Adjusted from the value given by ORNL [57] in relation to the SO design change from their conditions to Portland conditions.

(8) Adjusted from the value given by ORNL [57] in relation to the apartment design change from their condition to Portland conditions.
Domestic Hot Water. The water heating requirements for the city are estimated using values listed in Table 3-3. The energy required for heating the hot water is assumed to be equal to the energy required to raise the temperature of the water from the inlet temperature of city-supplied domestic water to a temperature of 140°F. The temperature 140°F is used here because this is the temperature for which estimated hot water use rates are tabulated. The energy required to offset the energy losses from the water heater are assumed to be accounted for by an over estimation of the water use. The temperature of the domestic water supply varies during the year, and the profile assumed representative of the Portland area is shown in Figure 3-7.

The hot water requirements of the city affect the design of the system as a whole by possibly increasing the required flow of water from the power plant at a particular time. To evaluate this, the hot-water-use profile for a day was determined by slightly modifying that of the apartments (the apartments represent greater than 80 percent of the hot water use during the year), whose profile is estimated by ASHRAE [63] as shown in Figure 3-8. The consequences of such a profile are basically as follows: If the pipe design from the power plant is governed by the heating plus the hot water heating, the maximum demand will be at about 7:30 in the morning and the hot water heating hourly contribution will be roughly one-seventh the total daily use. If the pipe design is governed by the space cooling plus hot-water heating, the maximum demand will be in the afternoon and the hot water heating contribution will be the average hourly use for
### TABLE 3-3. Domestic Hot Water Requirements

<table>
<thead>
<tr>
<th>Unit</th>
<th>Use</th>
</tr>
</thead>
<tbody>
<tr>
<td>Apartments</td>
<td>30 gal/day person(^{(1)})</td>
</tr>
<tr>
<td>Shopping-Offices</td>
<td>2 gal/day employee</td>
</tr>
<tr>
<td>Hospitals</td>
<td>100 gal/day bed</td>
</tr>
<tr>
<td>Hotels</td>
<td>50 gal/day room</td>
</tr>
<tr>
<td>Public Schools &amp;</td>
<td>35 gal/week student</td>
</tr>
<tr>
<td>Universities</td>
<td></td>
</tr>
<tr>
<td>Cleaning (except in</td>
<td>30 gal/day 10,000 ft(^2)</td>
</tr>
<tr>
<td>Apartments)</td>
<td></td>
</tr>
</tbody>
</table>

\(^{(1)}\) Conservative estimate based on 90 gallons/day-family for a family of three with dishwasher and automatic washer.
the day, when considering roughly 17 hours of reasonable demand per day (i.e., total daily use divided by 17).

**Results.** Table 3-4 shows the design energy requirements for space heating, space cooling, and hot water heating for each type of unit in the Portland climate. These values, when multiplied by the number of units in the town yield the city design requirement.\(^{20}\)

Table 3-5 presents the yearly energy requirements for space heating and cooling for each type of building for the Portland climate and the city of 129,000 population. These values yield the total yearly energy requirements for the 129,000 population city to be:

<table>
<thead>
<tr>
<th>Requirement</th>
<th>Energy Requirement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Space Heating</td>
<td>(1.61 \times 10^{12}) Btu</td>
</tr>
<tr>
<td>Space Cooling</td>
<td>(0.87 \times 10^{12}) Btu</td>
</tr>
<tr>
<td>Hot Water Heating</td>
<td>(1.20 \times 10^{12}) Btu</td>
</tr>
</tbody>
</table>

Similarly, Table 3-6 shows the yearly energy requirements for each type of building for the Portland climate and the city of 21,500 population. From these and the hot water requirements, the total yearly energy requirements of the 21,500 population city are:

<table>
<thead>
<tr>
<th>Requirement</th>
<th>Energy Requirement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Space Heating</td>
<td>(2.63 \times 10^{11}) Btu</td>
</tr>
<tr>
<td>Space Cooling</td>
<td>(1.38 \times 10^{11}) Btu</td>
</tr>
<tr>
<td>Hot Water Heating</td>
<td>(1.96 \times 10^{11}) Btu</td>
</tr>
</tbody>
</table>

\(^{20}\)The analysis of the loads found in this section must be postponed until the performance of the systems can be specified. However, it may be noted here that the water heating load is almost as large as that for space heating, and the ratios of the corresponding loads of the two types are slightly greater than six (the population ratio) due to slightly larger "downtown" proportion of the 129,000 city.
TABLE 3-4. Design Energy Requirements Per Unit for Portland Climate.

<table>
<thead>
<tr>
<th>Unit</th>
<th>Space Heating (MBtu/hr)</th>
<th>Cooling (tons)</th>
<th>Summer(^{(1)}) (MBtu/hr)</th>
<th>Winter(^{(2)}) (MBtu/hr)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>5.04</td>
<td>336</td>
<td>0.865</td>
<td>2.313</td>
</tr>
<tr>
<td>S</td>
<td>0.56</td>
<td>67</td>
<td>0.017</td>
<td>0.045</td>
</tr>
<tr>
<td>ES</td>
<td>2.70</td>
<td>225</td>
<td>0.230</td>
<td>0.616</td>
</tr>
<tr>
<td>SS</td>
<td>9.36</td>
<td>780</td>
<td>0.799</td>
<td>2.137</td>
</tr>
<tr>
<td>RI</td>
<td>5.72</td>
<td>693</td>
<td>0.173</td>
<td>0.462</td>
</tr>
<tr>
<td>C</td>
<td>1.95</td>
<td>162</td>
<td>0.166</td>
<td>0.445</td>
</tr>
<tr>
<td>H</td>
<td>8.40</td>
<td>667</td>
<td>2.600</td>
<td>6.953</td>
</tr>
<tr>
<td>SO</td>
<td>5.98</td>
<td>725</td>
<td>0.181</td>
<td>0.484</td>
</tr>
<tr>
<td>D</td>
<td>2.64</td>
<td>320</td>
<td>0.080</td>
<td>0.213</td>
</tr>
<tr>
<td>M</td>
<td>2.67</td>
<td>204</td>
<td>0.834</td>
<td>2.231</td>
</tr>
<tr>
<td>T</td>
<td>3.30</td>
<td>400</td>
<td>0.100</td>
<td>0.267</td>
</tr>
</tbody>
</table>

\(^{(1)}\)Daily use in July divided by 17.

\(^{(2)}\)Daily use in January divided by 7.
TABLE 3-5. Yearly Space Heating and Air Conditioning Energy Requirements for Each Type of Building For Portland Climate and City of 129,000 Population.

<table>
<thead>
<tr>
<th>Unit</th>
<th>Number of Units in City</th>
<th>Space Heating $10^{10}$ Btu</th>
<th>Space Cooling $10^6$ ton-hr.</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>192</td>
<td>127.37</td>
<td>28.25</td>
</tr>
<tr>
<td>S</td>
<td>36</td>
<td>2.16</td>
<td>3.83</td>
</tr>
<tr>
<td>ES</td>
<td>24</td>
<td>6.97</td>
<td>4.26</td>
</tr>
<tr>
<td>SS</td>
<td>6</td>
<td>6.04</td>
<td>7.38</td>
</tr>
<tr>
<td>RI</td>
<td>1</td>
<td>0.62</td>
<td>1.09</td>
</tr>
<tr>
<td>C</td>
<td>14</td>
<td>2.94</td>
<td>3.59</td>
</tr>
<tr>
<td>H</td>
<td>2</td>
<td>1.81</td>
<td>2.10</td>
</tr>
<tr>
<td>SO</td>
<td>17.5</td>
<td>11.26</td>
<td>20.01</td>
</tr>
<tr>
<td>D</td>
<td>2</td>
<td>0.57</td>
<td>1.01</td>
</tr>
<tr>
<td>M</td>
<td>2</td>
<td>0.57</td>
<td>0.64</td>
</tr>
<tr>
<td>T</td>
<td>1.5</td>
<td>0.53</td>
<td>0.95</td>
</tr>
<tr>
<td>Unit</td>
<td>Number of Units In City</td>
<td>Space Heating $10^6$ Btu</td>
<td>Space Cooling $10^6$ ton-hr</td>
</tr>
<tr>
<td>------</td>
<td>-------------------------</td>
<td>---------------------------</td>
<td>-----------------------------</td>
</tr>
<tr>
<td>A</td>
<td>32.00</td>
<td>21.23</td>
<td>4.71</td>
</tr>
<tr>
<td>S</td>
<td>6.00</td>
<td>0.36</td>
<td>0.64</td>
</tr>
<tr>
<td>ES</td>
<td>4.00</td>
<td>1.16</td>
<td>0.71</td>
</tr>
<tr>
<td>SS</td>
<td>1.00</td>
<td>1.01</td>
<td>1.23</td>
</tr>
<tr>
<td>RI</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td>C</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td>H</td>
<td>0.25</td>
<td>0.23</td>
<td>0.26</td>
</tr>
<tr>
<td>SO</td>
<td>3.00</td>
<td>1.93</td>
<td>3.43</td>
</tr>
<tr>
<td>D</td>
<td>0.33</td>
<td>0.09</td>
<td>0.17</td>
</tr>
<tr>
<td>M</td>
<td>0.67</td>
<td>0.19</td>
<td>0.21</td>
</tr>
<tr>
<td>T</td>
<td>0.25</td>
<td>0.09</td>
<td>0.61</td>
</tr>
</tbody>
</table>
3-3. **Energy Requirements: Yearly Profile**

The energy requirements of the city vary considerably throughout the year, so to determine the overall effect of the city operation on the power plant outputs (both electrical and warm water), and on other possible components of an integrated operation, it is necessary to estimate the energy requirements throughout the year. At this point monthly average values are considered as the next step. Two separate methods are used, one for the apartments and another for the other buildings.

**Apartments:** The apartment-energy-use variation throughout the year was modeled by considering degree days for heating and degree hours for cooling. The values used for the Portland area are listed in Table 3-7. The total yearly heating was divided into monthly contributions based on the number of degree days for the month. Similarly, the total yearly cooling was divided into monthly contributions according to the number of degree hours in the month. The monthly energy requirement for domestic water heating was calculated as outlined in the "Design and Yearly Total" section above, based on the monthly temperatures given in Figure 3-7.

**Other Buildings:** The buildings in the city besides apartments were considered to have thermal-energy requirement profiles similar to shopping-office buildings for heating and cooling. There are two justifications for this assumption: First, the apartments and the shopping-office buildings together require greater than 80% of the thermal energy for the city. Second, the other buildings
in the city for the most part have operation schedules similar to the shopping-office buildings.

The profile selected as representative of the shopping-office buildings is shown in Figure 3-9. This profile was selected as an approximate average of profiles from computer analysis of several office buildings in Portland and then smoothed on a rate basis to obtain an approximate but representative profile. The energy required for water heating was calculated in the same manner as for the apartments.

Figures 3-10 and 3-11 show the yearly load profile of water heating, space heating and cooling for the 129,000 city and the 21,500 city respectively for the Portland climate. Again, analysis of these curves will be deferred until the particular energy system is chosen.
TABLE 3-7. Heating Degree Days and Cooling Degree Hours Used for Monthly Distribution of Apartment Space Heating and Air Conditioning Requirements (65° Base).

<table>
<thead>
<tr>
<th>Month</th>
<th>Heating Degree Days (1)</th>
<th>Cooling Degree Hours (2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>J</td>
<td>825</td>
<td>0</td>
</tr>
<tr>
<td>F</td>
<td>644</td>
<td>0</td>
</tr>
<tr>
<td>M</td>
<td>586</td>
<td>4</td>
</tr>
<tr>
<td>A</td>
<td>396</td>
<td>250</td>
</tr>
<tr>
<td>M</td>
<td>245</td>
<td>848</td>
</tr>
<tr>
<td>J</td>
<td>105</td>
<td>1300</td>
</tr>
<tr>
<td>J</td>
<td>25</td>
<td>3280</td>
</tr>
<tr>
<td>A</td>
<td>28</td>
<td>1852</td>
</tr>
<tr>
<td>S</td>
<td>114</td>
<td>316</td>
</tr>
<tr>
<td>O</td>
<td>335</td>
<td>0</td>
</tr>
<tr>
<td>N</td>
<td>597</td>
<td>0</td>
</tr>
<tr>
<td>D</td>
<td>753</td>
<td>0</td>
</tr>
</tbody>
</table>

(1) Portland Airport [65].

(2) Average for 1950-1959, supplied by R. Helm of Portland General Electric.
3-4. **System Design**

The concepts of both systems to be considered were outlined before in the last section of the first chapter. This section will further particularize the systems to the current application, describing the design and its basis. For the most part, the discussion is directed to the apartment systems since the apartments are the bulk of the load and the system that satisfies the apartment requirements could also be applied in their general format to the commercial buildings.

**Steam Bleed-Off System (BCS).** The conceptual design of this system at the power plant was shown in Figure 1-2. The 300°F water leaving the power plant is pumped by one or more pumping stations to and throughout the city. For the apartments, the hot water comes to a central plant for each apartment block where it is used to (1) heat water for domestic uses, (2) heat water (perhaps direct contact) for space heating, and (3) operate an absorption refrigeration system to produce cold water for space cooling. The loop water then returns to the power plant at a temperature which ranges from approximately 150°F to 210°F. The reason for the temperature variation on the returning loop water is that the absorption system can cool loop water only to about 210°F while space heating can cool it to about 150°F and hot water heating can cool the loop water even lower.

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21 In the design of this system, the work of Miller, et.al. [57] has been relied on. The attempt here is to re-design their system for the new locations and cities that are to be considered while maintaining their basic system layout. Consequently, every detail of the system is not specified here.
For the steam bleed-off system, which uses hot water for cooling as well as heating processes the design energy requirements for cooling such as presented in Table 3-4 can be changed to loop heating requirements to operate the absorption refrigeration system. The thermal design for the system is then specified by the maximum of space heating plus the hot water daily energy requirement divided by seven, or the maximum of absorption unit heating plus the hot water daily energy requirement divided by seventeen. The piping design is further specified by the fact that during heating the loop water returns to the power plant at a temperature of about 150°F while during peak cooling the return water temperature is close to 210°F.

For the Portland location, the design is governed by the space cooling plus hot water heating, the values of which are presented in Table 3-4. For the city of 129,000 the summer peak design is 1850 MBtu/hr (536 MW) of heating. Designing the system with a diversity of 0.9 results in a thermal design value of 1647 MBtu/hr (482 MW) with a design flow rate of hot water of $1.6 \times 10^7$ lbm/hr (34,000 gal/min). The piping is designed with a mean friction loss of 2.5 ft. of water per hundred feet of pipe. The resulting piping design is shown in Figures 3-12 to 3-14.

For this system the peak pumping capacity would require total pumping power of about 26,500 hp. Total yearly use of electricity for pumping is roughly estimated at 15,000 megawatt hours. The heat loss from the piping assuming the loss rates of Reference 57 amounts to approximately 50 MBtu/hr, or $4.4 \times 10^5$ MBtu for the entire year.
Similarly for the city of 21,500, the thermal design value is 267 MBtu/hr (78MW) with the piping as shown in Figure 3-15 and the associated pumping requirements and energy loss by heat transfer from the pipes as given below:

- Peak Pumping Power: 2400 horsepower
- Electrical Energy for Pumping (yearly): 1840 megawatt hrs
- Heat Loss from Piping (yearly): $9.7 \times 10^4$ MBtu

**Heat Pump System (HPS).** The system, as depicted in Figure 1-3 and amplified in Figure 3-16a for heating and Figure 3-16b for the cooling cycle, basically consists of a heat pump unit, located at the point of consumption, that provides the space heating, cooling, and a substantial portion of the domestic hot water requirements while exchanging energy with the water circulated from the power plant. Since domestic water heating represents a substantial portion of the total energy requirements in residential applications, it is highly desirable to accomplish as much of it as possible while using the coefficient of performance advantage of the heat pump. This is the reason a low temperature storage tank at 120°F and a high temperature storage tank at 160°F are shown. Showers and basins draw from the 120°F tank while the laundry and dishwasher is served from the 160°F tank. The electrical resistance heaters are used to keep the temperature above 120°F in the main tank and to boost it to 160°F in the small tank.

Energy for space heating is delivered to a forced air system through the main condenser section while a short section of the
water tank acts as a desuperheater to raise the tank temperature during space heating. Water heating is handled as a separate mode during which the 3-way valve A switches the air unit out and 3-way valve B brings the rest of the tank heating element into the circuit. During both heating cycles, the circulating water gives energy to evaporate the refrigerant at the loop heat exchanger.

During space cooling, the operations of the compressor and water heater are the same as in space heating. But, now the refrigerant flows from the water heater to the loop heat exchangers, where it condenses and rejects energy to the loop water. After expansion, the refrigerant flows through the refrigerant-to-air coil (which was the condenser during heating but now is the evaporator) where air is cooled for circulation to the space. Consequently, during space cooling some of the water heating is "free" and loop water is not cooled, but heated.

The performance of the heat pump unit changes as the evaporator and the condenser temperatures vary. Consequently, the power plant condenser temperature and the design temperature drop for the water circulating from the power plant to the city alters the performance of the heat pump. The heating performance variation of the heat pump as a function of these two variables is shown in Figure 3-17a, which was constructed using representative refrigeration cycle performance data as a function of evaporator and condenser temperatures for refrigerant 22 or 12 as presented by ASHRAE [61] while assuming terminal-temperature differentials of 5°F for the heat-pump evaporator
and the power-plant condenser. Performance with various loop-water temperature drops are shown.

The performance of the unit during space cooling (reverse-cycle operation) is assumed to be the same as during heating for given evaporator and condenser temperatures. Figure 3-17b shows this performance as a function of the power-plant condenser temperature, when terminal-temperature differentials of 5°F are assumed for the unit condenser and the power plant condenser. Again, performance with various loop-water temperature changes are illustrated.

The outlet temperature profiles for the power plant condenser and cooling tower are shown in Figure 3-18. The top curve gives the temperature of the loop water as received in the city except for four months (June 1 - September 30) when it is given by the lower profile. For certain periods, it would be possible to obtain better loop water temperature by mixing flows from these two outlets. This optimum loop water temperature is evaluated in Appendix C as a function of the gross cooling and heating loads, but is not incorporated into the analyses to keep them general.

The piping design for this system is governed by the larger of -

(1) the low-temperature energy-supply requirements for the heat-pump evaporator to provide space-heating design and the coincident water-heating design (1/7 of daily use).

(2) the high-temperature energy-sink requirements for the space cooling minus the coincident water-heating design (1/17 of daily use).
The piping design is also highly dependent on the designed temperature change of the loop water. Simplified economic analyses were made to determine the most desirable temperature change among the choices 20, 30, and 40 degrees Fahrenheit. The 40°F temperature change turned out to be the most economical for the Portland location. Larger temperature changes were not investigated because the freezing point of the loop water would be too closely approached, unless the power plant was artificially operated at a higher condenser temperature than indicated by the profile presented in Figure 3-18. These artificially produced higher condenser temperatures did not appear economically desirable.

For the city of 129,000 in the Portland climate and a loop water temperature change of 40°F, the governing requirements are respectively -

(1) energy supply = 1320 MBtu/hr
(2) energy sink = 1305 MBtu/hr

Closeness of these numbers indicate a good balance between summer and winter loads.

For a loop-water temperature change of 40°F, and assuming a 0.9 diversity, the HPS has a design flow rate of $2.97 \times 10^7$ lbm/hr for the city of 129,000 served population with a Portland climate. The piping designs of the HPS serving both the 129,000 and 21,500 population cities located in a Portland climate are presented in Figures 3-19 through 3-22.
3-5. System Performance

The purpose of this section is to evaluate the performances of both of the energy systems. It is necessary to look at the power plant in conjunction with each system. The approach here is to assume that the power plant operates at a constant input of high-quality thermal energy; the variables then considered are the net electrical output and the net warm water discharge from the power plant after the thermal energy requirements of the city have been met. The use of a cooling tower in this location results in the power plant condenser temperature profile given in Figure 3-18. Other characteristics of the three plant types considered are given in Section 1-5.

Steam Bleed-Off System

Figure 3-23 shows the average monthly energy requirements of the bleed-off system for the city of 129,000 in the Portland climate. The graph is cumulative and shows the contributions of water heating, space heating, and heating for absorption cooling.

Figures 3-24, 3-25 and 3-26 show the effects that meeting the requirements with the profiles of Figure 3-23 have on the overall performance of LWR, FBR, and FFP type power plants. The top curve of each figure (labeled GROSS) represents the electrical output from the power plant when no thermal requirements have been satisfied. It changes throughout the year because of varying condenser temperature. The next curve (labeled BOS) shows the
net electrical output from the plant when the thermal energy requirements are satisfied by the bleed-off system. Items that are not included in this curve, but that must be charged against the BOS are (1) the decrease in electrical generation because of heat transfer losses from the piping and (2) the electricity required for pumping. Yearly average values of these items are noted on the LWR plots, and are denoted by $\hat{E}_{\text{loss-Q}}$ and $\hat{E}_{\text{pump}}$ respectively. The lowest curve (labeled AES) shows the net electricity that would be available for purposes other than supplying the thermal energy requirements of the city if the heating and water heating were accomplished by electrical resistance heating and the air conditioning was accomplished with an electrically driven unit with a cooling COP of 4.0.

Each of Figures 3-24 to 3-26 reveal that, for monthly average values, the steam bleed-off system decreases the net electrical energy that is available for other purposes much less than the all-electric system. If daily load profiles are considered, the steam-bleed-off system results in a maximum decrease of net electrical output of approximately 77 MW for LWR plants, depending on the condenser temperature, while the all-electric system would result in a maximum decrease of electrical output of approximately 470 MW—a considerable difference.

Figure 3-27 shows the change in the thermal-energy disposal requirements at the power plant as a result of urban utilization in the city of 129,000 for the bleed-off system. The changes
amount to approximate yearly decreases of 5.2%, 7.0%, and 8.6% respectively for the LWR, FBR, and FFP type plants. Although these are the values for a given type plant, the actual decrease of thermal pollution is greater because the urban energy requirements are satisfied in an efficient manner. To visualize this, consider now, an LWR type plant and compare the thermal disposal requirements with the bleed-off system and the all-electric system considered above when providing the same net electrical output in addition to satisfying the urban thermal-energy requirements. For the city of 129,000 and the Portland climate, the January average thermal-energy disposal rate would be 2037 MW with the bleed-off system, and 2507 MW with the all-electric system or a 23% difference. Similarly, for August the thermal-energy disposal rate is 2107 MW with the bleed-off system and 2309 MW with the all-electric system or a 9.5% difference.

Figures 3-28 through 3-32 for the city of 21,500 in a Portland climate parallel the Figures 3-23 through 3-27 presented above.

Heat Pump System

For BOS the space and hot water requirements directly give the demand on the power plant to satisfy them. But the HPS there is an electrical demand in addition to the thermal energy demand from the power plant. Both the electrical and thermal energy demand must be calculated in components of space heating, low-temperature water heating, high-temperature water heating and space cooling. Each component of the load requires modification with the corresponding individual unit performance values, similar
to the case in the calculation of heating needed for a cooling load in BOS. For the HPS proposed here, there are two further modifications on loads.

First, when cooling and heating loads exist simultaneously there is the possibility of achieving part of the heating with the energy rejected from the cooling. In the HPS possibilities exist for heating domestic hot water in apartments and offices, and for space heating in offices since most of them require some cooling year around. Continuous load profiles would be needed to determine load overlappings accurately, but an approximate method will be sufficient and is used here. For an average day of each month, hourly domestic hot water usage in apartments is obtained from the profiles given in Figures 3-7 and 3-8. For other buildings hot water usage is assumed uniform during the day. The total monthly cooling load is distributed to the hours of an average day according to the local sol-air temperature calculated from actual weather data using the methods discussed by ASHRAE [64]. The hot water and space cooling profiles of each month are then compared and fractions of water that could be heated free (i.e., from the cooling energy rejection) is estimated, somewhat conservatively. Free space heating in non-residential buildings is estimated to be 75% of the cooling rejection that remains after the contribution to the water heating, or half of the space heating load, whichever is smaller.
To indicate the resulting range for the Portland climate, in January, all free hot water heating occurs in the non-residential buildings and amounts to roughly half of the office building hot water requirement, or 6% of the city hot water heating requirements. Offices maintain about this amount of free water heating all year. In July, the total free water heating for the city reaches a maximum of about 60% of the water heating requirement. Free space heating is 10% of the city heating in summer and 4% in winter.

The second modification is for the heat rejection of cooling that must be handled by the power plant cooling system. There is a reduction from the gross rejection because some of it is utilized for hot water or office space heating. A further reduction is possible by introducing local cooling towers in application sites. For the HPS evaluated here, it was assumed there would be local cooling towers only in the non-residential buildings, and their capacity would be limited to satisfy the average load for the month of May. Use of the local towers improves the overall air conditioning COP by providing a lower sink temperature, reduces the rejection at the power plant site and provides an emergency back-up system. Their use might possibly be extended to apartment complexes.

The electricity requirements of the HPS for these assumptions are presented in Figure 3-33 for the 129,000 city in the Portland climate. Contributions to heating, water heating through the heat
pump, electrical resistance water heating, and space cooling are illustrated.

Table 3-8 presents the monthly average circulation requirements of the plant-city piping loop for the HPS with the city of 129,000 in the Portland climate. The flow rate must provide for both the energy supply and rejection requirements. Note that the average cooling requirements are substantially less than the average energy requirements from the loop; the main reasons being the local cooling towers of non-residential buildings and the free water heating.

Figures 3-34, 3-35 and 3-36 show how the heat pump system compares with the bleed-off system and the all-electric system with regard to net electricity available from the power plant for other uses. These three figures are for the city of 129,000 in the Portland area. As these figures indicate, the relative position of the heat pump system increases as the efficiency of the power plant increases. Figure 3-34 shows the BOS to have the greater net electricity output for the LWR type plant while Figure 3-36 shows the HPS to have the greater net electricity output for the FFP type plant. This result would be predicted from Figure 2-4a.

Figure 3-37 shows the thermal energy disposal requirements of the given size power plant for each of the three systems. The heat pump system has slightly greater thermal energy disposal requirements than the bleed-off system during the winter and sub-
stantially more, at the power plant, during the summer. When comparing with the all-electric system, recall that for equal net electricity output, the thermal energy disposal requirements are greater than for the given size power plant considered here.

Figures 3-38 through 3-42 and Table 3-9 for the city of 21,500, with a Portland climate, parallel the material presented above in Table 3-8 and Figures 3-33 through 3-37.

In addition, BOS will also have high thermal energy disposal requirements at the point of consumption due to the cooling towers required for the absorption units. Localized cooling towers could also be used in the HPS, thereby keeping the thermal energy disposed at the power plant the same as if no urban utilization was considered during the summer. Here, use of local cooling towers with the heat pump system has been confined to units for commercial buildings with a limited capacity sufficient for the average load during the month of May.
TABLE 3-8. Monthly Average Circulation Requirements of Plant-Town Loop for the Heat Pump System with the City of 129,000 in the Portland Climate.

<table>
<thead>
<tr>
<th>Month</th>
<th>Net Thermal Energy Requirement From the Loop (MW)</th>
<th>Net Cooling Energy Rejection to the Loop (MW)</th>
<th>Loop Flow Rate 1/ (gal/sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>J</td>
<td>124.47</td>
<td>0</td>
<td>360</td>
</tr>
<tr>
<td>F</td>
<td>104.57</td>
<td>0</td>
<td>302</td>
</tr>
<tr>
<td>M</td>
<td>88.33</td>
<td>0</td>
<td>255</td>
</tr>
<tr>
<td>A</td>
<td>69.99</td>
<td>4.54</td>
<td>215</td>
</tr>
<tr>
<td>M</td>
<td>48.98</td>
<td>10.65</td>
<td>172</td>
</tr>
<tr>
<td>J</td>
<td>24.64</td>
<td>12.49</td>
<td>107</td>
</tr>
<tr>
<td>J</td>
<td>13.04</td>
<td>43.79</td>
<td>164</td>
</tr>
<tr>
<td>A</td>
<td>13.78</td>
<td>43.28</td>
<td>165</td>
</tr>
<tr>
<td>S</td>
<td>22.91</td>
<td>28.39</td>
<td>148</td>
</tr>
<tr>
<td>O</td>
<td>58.84</td>
<td>5.72</td>
<td>187</td>
</tr>
<tr>
<td>N</td>
<td>89.91</td>
<td>0</td>
<td>260</td>
</tr>
<tr>
<td>D</td>
<td>111.47</td>
<td>0</td>
<td>322</td>
</tr>
<tr>
<td></td>
<td><strong>Yearly Average</strong></td>
<td><strong>64.24</strong></td>
<td><strong>12.41</strong></td>
</tr>
</tbody>
</table>

1/ For a loop water temperature change of ± 40°F.
### TABLE 3-9. Monthly Average Circulation Requirements of Plant-Town Loop for the Heat Pump System with the City of 21,500 in the Portland Climate.

<table>
<thead>
<tr>
<th>Month</th>
<th>Net Thermal Energy Requirement From the Loop (MW)</th>
<th>Net Cooling Energy Energy Rejection to the Loop (MW)</th>
<th>Loop Flow Rate(^1) (gal/sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>J</td>
<td>20.3</td>
<td>0</td>
<td>58.7</td>
</tr>
<tr>
<td>F</td>
<td>16.9</td>
<td>0</td>
<td>48.8</td>
</tr>
<tr>
<td>M</td>
<td>14.4</td>
<td>0</td>
<td>41.5</td>
</tr>
<tr>
<td>A</td>
<td>11.5</td>
<td>0.73</td>
<td>35.3</td>
</tr>
<tr>
<td>M</td>
<td>8.0</td>
<td>1.64</td>
<td>28.0</td>
</tr>
<tr>
<td>J</td>
<td>4.0</td>
<td>1.99</td>
<td>17.4</td>
</tr>
<tr>
<td>J</td>
<td>2.1</td>
<td>7.12</td>
<td>26.7</td>
</tr>
<tr>
<td>A</td>
<td>2.3</td>
<td>7.01</td>
<td>26.8</td>
</tr>
<tr>
<td>S</td>
<td>3.8</td>
<td>4.60</td>
<td>24.1</td>
</tr>
<tr>
<td>O</td>
<td>9.7</td>
<td>0.88</td>
<td>30.6</td>
</tr>
<tr>
<td>N</td>
<td>14.7</td>
<td>0</td>
<td>42.4</td>
</tr>
<tr>
<td>D</td>
<td>18.1</td>
<td>0</td>
<td>52.4</td>
</tr>
<tr>
<td></td>
<td>Yearly Average</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>10.5</td>
<td>2.00</td>
<td>36.0</td>
</tr>
</tbody>
</table>

\(^1\) For a loop water temperature change of \(\pm 40^\circ F\).
3-6. Other Climates

The discussion in this chapter so far has been restricted, climate wise, to Portland. By applying identical procedures, the energy requirements and system performances for the same cities located in Astoria or Pendleton, Oregon climates have also been determined and presented in detail in the final report [58] of the research effort which included the work reported in this chapter. For brevity, this section will summarize the specifications and give the results for the 129,000 city only.

The characteristics of Astoria and Pendleton climates are indicated by the major design parameters listed in Table 3-10. As can be seen, Astoria has a plain climate that requires no cooling for apartments; but heating, sizeable even during the summer, adds up to exceed the yearly load of either of the other locations. Also, the use of once-through cooling with ocean water gives a flat and fairly low condenser temperature profile. Pendleton has the severest climate variation among the three; and with a reservoir as the main thermal sink, the condenser temperature becomes very low in winter. The Figures 3-43 and 3-44 plot the various load components for the 129,000 city in Astoria and Pendleton respectively.

Electrical outputs and condenser cooling requirements for the systems and the power plants considered are presented in Figures 3-45 through 3-48 for Astoria, and 3-49 through 3-52 for Pendleton. Comparing them with the Portland results (Figures 3-34 through 3-37) reveals only minor differences. The yearly averaged electrical
output from the HPS as compared to that from the BOS improves slightly for Astoria, and a little more in Pendleton where they coincide for FBR and the HPS is slightly better with FFP. The relative improvement of the HPS summer performance in going from Portland to Pendleton is due to the lowered temperatures of the power plant condenser and of the loop water, and the increased cooling load. Each of these changes favor the HPS as the analysis of the actual systems in Chapter 2 indicated (see Figure 2-5c). Because of the sharply decreased cooling load in Astoria, the HPS improvement there is not big despite the lowest water temperatures. In winter, Astoria and Pendleton again present lower condenser temperatures and sometimes increased heating loads, all of which work in general against the HPS, slightly dropping its relative performance. However, with FFP, the HPS remains superior to the BOS at all condenser temperatures, and the increased loads amplify this superiority. Again, these tendencies are in agreement with the predictions from the actual system analysis of Chapter 2. As in Portland, artificially keeping the condenser temperatures high for the HPS during winter did not seem economically worthwhile.
## TABLE 3-10. Design and Climatic Conditions of the Three Locations

<table>
<thead>
<tr>
<th>Portland</th>
<th>Astoria</th>
<th>Pendleton</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>25</td>
<td>34</td>
</tr>
<tr>
<td>22</td>
<td>17</td>
<td>25</td>
</tr>
<tr>
<td>60</td>
<td>55</td>
<td>72</td>
</tr>
</tbody>
</table>

- **25** - Apartment (A) Heating Design (Btu/hr ft²)
- **22** - Office (SO) Heating Design (Btu/hr ft²)
- **60** - Heating Design Temperature Difference (°F)
- **4635** - Yearly Heating Degree Days
- **825** - January Heating Degree Days
- **25** - July Heating Degree Days
- **0.71** - Apartment (A) "C" Value
- **0.58** - Office (SO) "C" Value
- **0.2310** - January fraction of office heating
- **0.0090** - July fraction of office heating
- **600** - Apartment (A) Cooling Design (ft²/ton)
- **375** - Office (SO) Cooling Design (ft²/ton)
- **449** - Yearly Cooling Degree Days
- **35** - May Cooling Degree Days
- **122** - August Cooling Degree Days
- **0.05** - Apartment (A) Cooling Load Factor
- **0.18** - Office (SO) Cooling Load Factor
- **0.0410** - January Fraction of Office Cooling
- **0.1220** - July Fraction of Office Cooling
- **0.04** - January Free Space Heating Fraction
- **0.15** - July Free Space Heating Fraction
- **0.06** - January Free Water Heating Fraction
- **0.60** - July Free Water Heating Fraction
- **67.6** - January Plant Cooling Water T (°F)
- **82** - July Plant Cooling Water T (°F)
- **37.3** - Cooling Water Temperature Increase (constant)
- **42** - January City Water Temperature (°F)
- **55** - August City Water Temperature (°F)
Using the engineering and performance data produced in the preceding, and the owning and operating costs furnished for the conventional alternatives, Dr. W. E. Schmisseur\textsuperscript{23} evaluated the economic potential of the BOS and the HPS systems. The method, assumptions, and the results obtained for one plant (LWR) and one location (Portland) are published as a part of a research report\textsuperscript{[58]}. They are summarized in this section to provide initial indications on this salient question.

The primary objectives of the economic study are to determine: (i) the maximum charge that the residents will be economically justified to pay for the loop water supplied in the BOS or the HPS systems, and (ii) the maximum distance that the suppliers will be economically justified to transmit the loop water at charges found above.

In fulfilling the first objective, the primary calculations are to evaluate the total cost of meeting space heating/cooling and domestic water heating demands of the 129,000 and the 21,500 cities by several alternative conventional systems - central gas, central oil, utilized electric, and utilized gas. In doing this, the thermal loads are converted to various energy resource demands by using average equipment performance estimations of each system. The analysis includes expected cost profiles of various energy resources, expected inflation rates on construction, maintenance and operation

\textsuperscript{23}Research Associate, Department of Agricultural Economics, Oregon State University.
costs, expected useful lives of the systems, and the interest costs up to the year 2016. For the Portland location, the unitized gas system is found to be the least-cost conventional system with a slight edge over the others. The cost of this system is assumed to establish the maximum cost that the owners and renters will be justified to pay for the BOS or the HPS; and the corresponding maximum loop water charges are found as a residual from this amount after all other owning and operating costs of the BOS or the HPS systems are paid. For the BOS system the loop water charge rises from $2.2/MBtu in 1977 to $48.1/MBtu in 2016. For the HPS, which uses the loop as the source and the sink, this maximum cost is $39.5/MLbm in 1977, increasing to $2083/MLbm by 2016.

In obtaining the second objective, various costs for the supplier are established. These costs include the capital investment, maintenance and operation of a distribution network; electrical revenue losses due to bleed-off in the BOS; property taxes and interests. The distribution network construction costs are based on the data used in Reference 57 which reflects work in new rather than established areas. Construction costs are spread over four years, starting with 1974. Balancing the costs thus found with the revenues that will be gained at the above loop water charges indicates the maximum distance it is economically justified to transmit the water. For the BOS, this distance is 2.5 miles for the 21,500 city and 8.5 miles for the 129,000 city. At these distances, the rate of pre-income tax return is about 15%; it will be less for
the larger distances, and will be in excess of 15% for shorter distances. Estimates for other cities with population below 129,000 show that the distance can be taken as directly proportional to the service population. Similar analysis for the HPS indicates that the rate of return will be somewhat below 15% even at the minimum expected distance of 3.5 miles for the 129,000 city.

These findings are preliminary in nature. They are based on the conceptual engineering designs, and initial estimates of capital investment and operation cost. Nevertheless they indicate that the BOS system could be operated as economically as the least-cost conventional alternative. For the case examined, the economic outlook of the HPS is not as bright. However, that system still should be kept in consideration. The performances of the HPS improves with favorable loop water profile and more efficient plants. For example in Pendleton with a FFP, the remaining electricity with the HPS exceeds that with the BOS - while the reverse is true for the LWR in Portland for which the economics have been evaluated. Thus, the economic position of the HPS would improve considerably in going from Portland with LWR to Pendleton with FFP. Also, in the analysis, the HPS is perhaps charged too heavily for the electricity used by the heat pumps. The bus-bar rate is applied in evaluating the revenue losses due to bleed-off in the BOS, however the HPS is charged at the consumer rates which remain much higher than the bus-bar rates despite the rapidly increasing fuel costs. If power plants closer to the cities
could be built the transmission of both water and electricity would be less costly, favorably altering the economic picture of both systems. By taking advantage of this reduced transmission costs, the utilities could supply the electricity to nearby towns at a reduced charge (approaching the bus-bar rates), and thus effectively encourage the acceptance of the HPS as well as the power plant itself. Another direction for improved economics could be integrating the city cold water distribution with the HPS loop water distribution. Unfortunately, the funds for the project did not allow the investigation of such proposals.

In addition, natural gas and oil prices have increased recently at a sharper rate than assumed in the economic analysis. As a result, the new systems now can compete with the alternatives at much greater plant-city distances. In fact for the 129,000 city, the revised distances are 21.5 miles for the BOS and 11.5 miles for the HPS.

Further, the fuels of conventional alternatives may simply be unavailable even at the high costs. Indication of such a situation is seen in recent local bans on connecting new customers to natural gas sources. Then, the BOS and the HPS would have fewer alternatives to compete with. Actually, the BOS and the HPS could be operated as another public service even in cases where they are poorly competitive with other systems. The reason to do so may be based on the positive sides of these systems, such as savings in energy, reductions in pollution, leveling of peak demands. In the economic analysis, hardly any direct credit was granted for these benefits.
Fig. 3-1. Layout for city with 200,000 population. High density residential area of 129,000 people and the downtown area are served by the district heating system. [57]

Fig. 3-2. Detail of college and downtown area for city with a population of 129,000 served people. [57]
Fig. 3-3. Typical residential mile square in apartment area of city of 129,000 served people. The symbols are described in Table 3-1.

Fig 3-4. Typical apartment-block A. Each apartment building is to be 3 stories in city of 129,000 served population, 2 stories in city of 21,500 served population.
(A geographical form (ocean, river, highway, etc.) influential on the city setting)

Fig 3-5. Layout for city of 21,500 served population.

Fig 3-6. Detailed layout for the area in the city with 21,500 served residents. The symbols are described in Table 3-1.

Fig 3-8. Hourly profile of apartment hot water demand on an average day. [31]
Fig. 3-9. Averaged and smoothed rate profiles for monthly distribution of annual heating and cooling loads for non-residential buildings in Portland, Oregon. Data were based on information supplied by the Portland General Electric Company.
Fig. 3-10. Monthly average energy requirements for water heating, air conditioning, and space heating for the Portland climate and city with 129,000 served people.

Fig. 3-11. Monthly average energy requirements for water heating, air conditioning, and space heating for the Portland climate and a city of 21,500 served people.
Fig 3-12. Pipelines for a typical square mile of the residential area in a city with 129,000 served people. The numbers indicate pipe sizes. All unmarked extensions are 3 inch extensions to apartment blocks. (Portland climate)

Fig 5-13. Pipelines for the downtown and college area in the city with 129,000 served people. The numbers indicate pipe sizes. (Portland climate)
Fig. 3-14. Mains and branch mains for the piping system for the city with 129,000 served people. The numbers indicate pipe sizes. (Portland climate)

Fig. 3-15. Pipelines for the 21,500 served population city. Numbers indicate pipe sizes. All unmarked extensions are 3 inch branches for apartment block A.
Fig. 3-16a. Schematic diagram of the heat pump operation during the space heating cycle.

Fig. 3-16b. Schematic diagram of the heat pump operation during the space cooling cycle.
Fig. 3-17a. Coefficient of performance for the heat pump unit.

Fig. 3-17b. Performance of the heat pump unit, operating as an air conditioner, as a function of power plant condenser temperature and loop temperature change.
Fig. 3-18. The monthly average inlet and outlet temperatures for a natural draft cooling tower operating at a 1100 MW power plant in Portland, Oregon with a constant temperature drop of 37.3°F. The solid line shows the temperature of the loop water received by the heat pumps of the HPS.
Fig. 3-19. Piping system for a typical square mile area in the city with 129,000 served population in a Portland climate using the heat pump system with a temperature change of 40°F for the loop water.

Fig. 3-20. Pipelines in the downtown and college area of the 129,000 served population city in a Portland location for the heat pump system with a temperature change of 40°F for the loop water.
Fig. 3-21. Mains and branch mains of the piping system for the 129,000 served population in a Portland climate using the heat pump system with a temperature change of 40°F for the loop water.

Fig. 3-22. Pipelines for 21,500 served population city in a Portland location for the heat pump system with a temperature change of 40°F for the loop water. Numbers indicate pipe sizes. All unmarked extensions are 4" lines for apartment block A.
Fig. 3-23. Total monthly average energy requirements for space heating, air conditioning and hot water heating for 129,000 people living in a city in a Portland climate, using the steam-bleed-off system.

Fig. 3-24. Monthly average electricity output by a nominal 1100 MWe LWR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Portland climate. Meaning of symbols is explained in the text.
Fig. 3-25. Monthly average electricity output by a nominal 1100 MWe FBR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Portland climate. Meaning of symbols is explained in the text.

Fig. 3-26. Monthly average electricity output by a nominal 1100 MWe FPP power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Portland climate. Meaning of symbols is explained in the text.
Fig. 3-27. Cooling requirements of nominal 1100 MWe LWR, FBR or FFP power plants serving the urban energy requirements of 129,000 people living in a Portland climate. Meaning of symbols is explained in the text.

Fig. 3-28. Total monthly average energy requirements for space heating, air conditioning, and hot water heating for 21,500 people living in a city in a Portland climate, using the steam-bleed-off system.
Fig. 3-29. Monthly average electricity output by a nominal 1100 MWe LWR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 21,500 in a Portland climate. Meaning of symbols is explained in the text.

Fig. 3-30. Monthly average electricity output by a nominal 1100 MWe FBR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 21,500 in a Portland climate. Meaning of symbols is explained in the text.
Fig. 3-31. Monthly average electricity output by a nominal 1100 MWe FFP power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 21,500 in a Portland climate. Meaning of symbols is explained in the text.

Fig. 3-32. Cooling requirements of nominal 1100 MWe LWR, FBR or FFP power plants serving the urban energy requirements of 21,500 people living in a Portland climate. Meaning of symbols is explained in the text.
Fig 3-33. Total monthly average electricity requirements for space heating, air conditioning, and hot water heating for 129,000 people living in a city in a Portland climate, using the heat pump system.

Fig. 3-34. Monthly average electricity output by a nominal 1100 MWe LWR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Portland climate. Meaning of symbols is explained in the text.
Fig. 3-35. Monthly average electricity output by a nominal 1100 MWe FBR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Portland climate. Meaning of symbols is explained in the text.

Fig. 3-36. Monthly average electricity output by a nominal 1100 MWe FFP power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Portland climate. Meaning of symbols is explained in the text.
Fig. 3-37. Cooling requirements of nominal 1100 MW(e) LWR, FBR or FFP power plants serving the urban energy requirements of 129,000 people living in a Portland climate. Meaning of symbols is explained in the text.
Fig. 3-38. Total monthly average electricity requirements for the city of 21,500 population in a Portland climate with the heat pump system.

Fig. 3-39. Monthly average electricity output by a nominal 1100 MWe LWR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 21,500 in a Portland climate. Meaning of symbols is explained in the text.
Fig. 3-40. Monthly average electricity output by a nominal 1100 MWe FBR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 21,500 in a Portland climate. Meaning of symbols is explained in the text.

Fig. 3-41. Monthly average electricity output by a nominal 1100 MWe FPP power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 21,500 in a Portland climate. Meaning of symbols is explained in the text.
Fig. 3-42. Cooling requirements of nominal 1100 MWe LWR, FBR or FFP power plants serving the urban energy requirements of 21,500 people living in a Portland climate. Meaning of symbols is explained in the text.
Fig. 3-43. Total monthly average electricity requirement for space heating, air conditioning, and hot water heating for 129,000 people in an Astoria climate.

Fig. 3-44. Total monthly average energy requirements for space heating, air conditioning and hot water heating for 129,000 people living in a city in a Pendleton climate.
Fig. 3-45. Monthly average electricity output by a nominal 1100 MWe LWR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in an Astoria climate. Meaning of symbols is explained in the text.

Fig. 3-46. Monthly average electricity output by a nominal 1100 MWe FBR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in an Astoria climate. Meaning of symbols is explained in the text.
Fig. 3-47. Monthly average electricity output by a nominal 1100 MWe FFP power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in an Astoria climate. Meaning of symbols is explained in the text.

Fig. 3-48. Cooling requirements of nominal 1100 MWe LNR, FBR, or FFP power plants serving the urban energy requirements of 129,000 people living in an Astoria climate. Meaning of symbols is explained in the text.
Fig. 3-49. Monthly average electricity output by a nominal 1100 MWe LWR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Pendleton climate. Meaning of symbols is explained in the text.

Fig. 3-50. Monthly average electricity output by a nominal 1100 MWe FBR power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Pendleton climate. Meaning of symbols is explained in the text.
Fig. 3-51. Monthly average electricity output by a nominal 1100 MWe FFP power plant, serving the space heating, air conditioning, and hot water heating energy needs for a population of 129,000 in a Pendleton climate. Meaning of symbols is explained in the text.

Fig. 3-52. Cooling requirements of nominal 1100 MWe LWR, FBR or FFP power plants serving the urban energy requirements of 129,000 people living in a Pendleton climate. Meaning of symbols is explained in the text.
CHAPTER IV

DIGITAL SIMULATION OF THE HPS

In the preceding chapter, two basic systems were examined and their monthly averaged performances compared. That analysis involved rather broad assumptions for the behavior of the major components, and although it is expected that the inaccuracy involved is not severe enough to significantly alter the conclusions, it remains as a goal to refine the analysis so that more detailed and definite engineering (and economic) performance estimates may be made. To this end, an engineering design of the heat pump components shown in Figure 3-16 is developed. Balancing these components under varying operation conditions will permit calculation of almost momentary, say hourly, values of the major variables. Then, a continuous simulation of the system is also attainable simply by repeating the procedure in time steps for any desired duration.

This chapter presents the engineering design of heat pumps for the city apartments, the development of a computer simulation of the HPS, and the simulation-results for a Portland-climate.

4-1. Overall Scope and Goals

The simulation effort is directed only at the heat pump with the following reasoning. The power plant performance here is simply a function of the turbine exhaust saturation temperature, so its performance can be evaluated by specification of the cooling tower
temperature profile. The energy storage capacity of the system is small with respect to energy use, so the response of the heat exchangers and distribution network can be assumed instantaneous. This leaves only the absorption unit of the BOS and the heat pump assembly of the HPS for further investigation. However, the temperature of the loop water that the absorption units receive is kept constant; hence, their performance remains close to the fixed value used, with only minor fluctuations due to local cooling tower performance variations. Further such absorption units for space comfort are already in practice, and the BOS itself has been extensively studied previously [57], and applied with minor differences in many district heating (and cooling) projects. Therefore, there is little urge to further consider the absorption units.

On the other hand several factors show need for simulating the heat pump units:

i) The performance of the heat pump strongly depends on the source and sink temperatures. Thus, in the HPS, the continually-changing loop water temperature in turn affects most parameters of interest - condenser and evaporator temperatures (pressures), maximum refrigerant temperature (pressure), electricity demand, operation duration, domestic hot water temperature, and the like.

ii) The loop water provides sink/source temperatures uncommon to residential heat pumps. Thus, present operation experience for this range is limited.
iii) Because the HPS does not affect the internal operations of the power plant, that system has been received very favorably by the utility planners even when its performance is not heavily superior.

These points encourage the further design and simulation of the HPS with the following specific advantages to be gained:

i) The design of the heat pump components must be further specified including the size and main construction features. This design effort will bring to light component balancing difficulties, indicate possible improvements in performance, and allow better cost data to be determined.

ii) The design, as well as the simulation results will furnish a useful background for a possible field simulation that may be undertaken for further information and demonstration.

iii) The results of the simulation will add confidence in or allow modified interpretations of the previous conclusions, and provide information to check and alter the assumptions previously used, including:

1. The loop temperature drop during heating, and the rise during cooling were taken to be a common constant, selected once for each location, that is

   \[ T_{\text{loop,out}} = T_{\text{loop,in}} \pm \Delta T. \text{(constant } \Delta T) \]

   For Portland, \( \Delta T = 40^\circ \text{F} \).

2. In all heating applications, the heat-pump condenser temperature was taken to be constant at 110°F, and the evaporator temperature was related to the loop outlet temperature with a constant -5°F approach, that is

   \[ T_{\text{con}} = 110^\circ \text{F}, \ T_{\text{ev}} = T_{\text{loop,out}} - 5^\circ \text{F} = T_{\text{loop,in}} - 45^\circ \text{F}. \]
3. In all cooling applications, the heat-pump evaporator temperature was taken to be constant at 50°F, and the condenser temperature was related to the loop outlet temperature with a constant +5°F approach, that is

\[ T_{ev} = 50°F, \quad T_{con} = T_{loop,out} + 5°F = T_{loop,in} + 45°F. \]

4. Domestic water was assumed to be heated by the refrigerant to the main demand temperature, requiring electrical resistance heating only to boost the temperature of the water for the clothes and dish washers.

5. A monthly average profile was used to estimate the free water heating during cooling.

iv) A simulation covering all yearly conditions will develop the extremum of the parameters which must be acceptable in a practical operation.

The responses to these points will to a degree depend on the specific system design. The thermodynamic best would be an impractical design and general constraints are too complex to explicitly seek the optimum. Rather, here the aim is to have a simple yet operable design based on and improved by basic thermoscientific principles and engineering experience.

4-2. Concepts and Design of the Components

The heat pumps of the HPS are expected to supply comfort heating/cooling and water heating by using the loop water as the thermal source in heating and the thermal sink in cooling as the schematics in Figure 3-16 showed. Basically, a domestic hot water tank is added to the essential components of a water-to-air heat pump - a compressor and throttling valve, separated by air and water (loop) heat exchangers,
the reversing valve and other control parts. The main hot water
tank is directly connected to the compressor outlet and always
receives some energy from the hot refrigerant vapor. During water
heating the tank is the condenser and the air unit is bypassed.
For space heating, the air unit is the main condenser but a small
section of the tank becomes a desuperheater in an effort of keeping
the tank temperature up. In both heating modes, the loop unit is
the evaporator. In the cooling mode, the reversing valve switches
the roles of the air and the loop units - the former becoming the
evaporator and the latter in combination with the tank working as
the condenser. In this mode, the tank becomes a sufficient condenser
by itself if it is being refilled, and during these periods the loop
may be turned off. Then the heat pump is operating at its ultimate,
simultaneously serving its source and sink.

While this setup remains valid, the actual design of the com-
ponents will depend on the profiles of the loads which will be
influenced by the building design and the climate. A complete
redesign should in general be necessary for each new location and
building type due to such variables as the ratio of peak heating and
cooling loads, and the desired loop temperature change. A service
size must also be specified, but it is not critical since its change
will alter all components by roughly the same factor, the performance
remaining fixed. Here a floor of the apartment buildings in a
Portland, Oregon climate is taken as the basic unit.24

24 Hence there would be three independent units in each of the
apartment buildings of the 129,000 city. Of course, the units might
serve a corresponding vertical section. The simulation program will be
kept general enough to accept and evaluate the new designs when and if
they are available.
Since the greatest design load is space heating, the design is based on that mode with the following specifications:

- Maximum space heating demand: 210,000 Btu/hr.
- Incoming air temperature: 72°F
- Return air temperature: 105°F
- Heat pump condenser temperature: 110°F
- Heat pump evaporator temperature: 55°F
- Minimum incoming loop temperature: 100°F
- Loop water temperature drop: 40°F

The design also has to satisfy the hot water and space cooling requirements with the following conditions:

- Maximum hot water use: 985 lb/hr.
- Range of city water temperature: 36-60 °F
- Water tank temperature: Average 130°F (minimum 125°F, maximum 150°F)
- Maximum cooling demand: 168,000 Btu/hr.
- Incoming loop water temperature range: 78-118 °F

The detail of the selected design is now presented.

**Refrigerant and Cycle:** As usual, a number of common refrigerants could be used, but for this work R-22 is selected even though it is more often used in lower temperature applications. The major factor in this selection is to utilize its relatively large superheating horn in boosting the tank temperature. The critical temperature of R-22, 205°F, is considered sufficiently high. A simple cycle with 10°F subcooling and 20°F superheating, with no pressure drops in the condenser or evaporator is assumed.
Compressor: It is easy to find a compressor capable of handling a heating capacity of 210,000 Btu/hr at $T_{\text{con}} = 110^\circ$F. Estimating its performance at other conditions is however a major task, especially for the wide range occurring in this application. Basically there are two options: i) Utilize an analytic generalization based on the actual data or separate theoretical simulations, ii) Utilize the plots of the manufacturers' data. Both methods are considered and as explained below, a combination is selected for this work.

Even though some rather extensive simulation work has been done on compressors (see for example the discussion and the sources given in References 66 and 67), the success has been limited and the partial results have not yet been synthesized to a general model [68,69]. ASHRAE [70] has proposed a set of equations for the capacity and power of reciprocating compressors, based mostly on the data from several units. (This source actually only suggests the form of the polynomials, a set of values for the coefficients are provided mainly as an illustration.) The COP values calculated from these equations show that they can only be valid within a narrow variable range. Outside this range, the behavior quickly becomes absurd. Still another analytical attempt is proposed in Reference 71. Using the commonly defined volumetric and compression efficiencies, this method yields the performance of a compressor with the specification of two adjustment constants. However, the values of these constants exhibited significant variation when calculated from known data sets (some of which were supplied by the author of Reference 71). Various attempts of modifying the last method resulted in only minor improvements,
indicating a basic deficiency in that model.

Although it is possible to find some data from manufacturers, this type of data has its weaknesses. First, noticeable variation among the data sets is to be expected. More significantly, the energy sources ordinarily available for space heating result in much lower evaporator temperatures. Therefore, some extrapolation is needed with all the data sets that are collected. This brings in the usual dangers of extrapolation and it will reflect operation of equipment outside its intended design region. Equipment specially designed for the given conditions should in general deliver a superior performance.

A set of data presented by ASHRAE [61] offers a partial solution to these problems since it is a composite of data from several compressors. An acceptable approach for this work then is to use the suggested forms of Reference 70 to represent the preferred data of Reference 61. In this merger, of course the latter has a greater contribution. The resulting equations for the refrigeration capacity ($C_\text{r}$, Btu/hr) and power requirement ($W$, Btu/hr) as a function of the heat-pump condenser and evaporator temperatures ($T_{\text{con}}$ and $T_{\text{ev}}$, °F) are

$$C_\text{r} = 3.1 T_{\text{con}} + 2.3 T_{\text{ev}}$$

Note that the Equations (2-25) and (2-26) were also based on the same data, but since, there, only one temperature varied at a time, simpler special COP equations were developed rather than reducing the above equations.
\begin{equation}
3.1 \ c_c = 140000 + 21500 \ T_{ev}^2 - 65 T_{ev} + 2275 \ T_{con}^2 - 13.75 \ T_{con}^2 - 231 \ T_{ev} \ T_{con} \\
+ 2.125 \ T_{ev}^2 T_{con} + 0.5625 \ T_{ev} T_{con}^2 - 0.00625 \ T_{ev}^2 T_{con}^2 \ (4-1)
\end{equation}

\begin{equation}
2.3 \ \dot{W} = 25,500 - 375 \ T_{ev}^2 - 2.5 T_{ev}^2 + 712.5 \ T_{con} \\
- 3.125 \ T_{con}^2 + 10.375 \ T_{ev} T_{con} \\
- 0.1375 \ T_{ev} T_{con}^2 + 0.04375 \ T_{ev} T_{con}^2 \\
+ 0.000625 \ T_{ev} T_{con}^2 \ (4-2)
\end{equation}

In the expected range of the operation, the COP from these equations closely duplicate those calculated from the data of Reference 61.

**Air Unit:** This is essentially a simple heat exchanger. The refrigerant is in finned tubes over which air is circulated by a two speed fan. Basic design calculations are:

- **Air mass flow rate**, supplied by the normal (low) fan speed is:

  \[ \dot{m}_a = \frac{\dot{Q}_h}{\Delta h_a} = \frac{210,000}{8.4} = 25,000 \ lb/hr. \]

- **Refrigerant mass flow rate**:

  \[ \dot{m}_r = \frac{\dot{Q}_h}{\Delta h_r} = \frac{210,000}{72} = 2,920 \ lb/hr. \]

Then

\[ UA = \frac{\dot{Q}}{\Delta T} \ln \frac{\Delta T_1}{\Delta T_2} = \frac{210,000}{105-70} \ln \frac{110-70}{110-105} = 12,500 \ \text{Btu/hr} \circ F, \]

and

\[ NTU = \frac{UA}{\dot{m}_c} = \frac{12,500}{(25,000 \cdot 0.2405)} = 2.07 \]
Surface type could be (7.75-5/8 T) of Kays and London [72].

A surface area of 10 ft² means, the air velocity is 560 ft/min and the Reynolds Number is \(1.3 \times 10^3\). Then, \(h_o = 12.4 \text{ Btu/hr-ft}^2\text{oF}\) and the fin efficiency is 0.8. The \(h_i\) for the condensing R-22 is found using Chato's equation [73]. On the third iteration, \(\Delta T = 12\text{oF}\), \(h_m = 240 \text{ Btu/hr °F}\). With these, \(U_o = 5.2 \text{ Btu/hr-ft}^2\text{oF}\), and \(A_o = 2,400 \text{ ft}^2\). Then the number of rows required is 9.7, or 10, and the depth of the unit is 17.5 inch. A thinner unit may be desirable, but is not considered here, since the pressure drop for the above is less than 0.18 in.H₂O. For a refrigerant velocity of 3.9 ft/sec, the number of tubes required is 24, and the width of the front face is 3.0 ft. Hence, the unit dimensions in feet are 3.33 x 3.0 x 1.46.

During cooling, this unit performs as an evaporator. When the fan speed is the same as in the heating mode and the effects of any water condensation neglected, the air-side coefficient may be assumed unchanged. The refrigerant-side coefficient is estimated (from Equation 1, p. 52 of Reference 64) as \(h_i = 138 \text{ Btu/ft}^2\text{hr °F}\). Then the new values are \(U_o = 4.0 \text{ Btu/hr ft}^2\), \(U_oA_o = 9,600 \text{ Btu/hr}; \text{ hence } U_oA_o\)\text{ev}/\(U_oA_o\)\text{cond} = 0.77. This ratio rises to 0.836 when the air flow rate is increased by 50 percent during cooling.

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26 The specifications of the first edition (1955) contained several mistakes, which are apparently corrected in this 2nd edition.
Loop Unit: The loop unit is a multiple pass tube-in-shell heat exchanger. Considering that it will often perform as the evaporator, a horizontal unit with the water being on the tube-side is selected. From design temperatures and heat transfer rate the required UA value for this unit is calculated to be 11,500 Btu/hr °F. The forced convection coefficient of the water-side is established [74] at 820 Btu/hr ft² °F for a mean loop-water temperature of 100°F. When the flow rate of the loop water remains constant, the essential parameter that will change is its temperature, and the effect on $h_i$ is included with the rule that $h_i \sim T_{\text{loop}}^{0.54} (°F)$ [74]. The mean heat-transfer coefficient for evaporating R-22 is more difficult to predict due to the dynamic situation. With several iterations guided by Figure 11 on page 46, of Reference 64, it is established at about 170 Btu/hr ft² °F. With these, $U_o = 110$ Btu/hr ft² °F, and hence $A_o = 104$ ft². For 28 tubes in a one-foot shell, the length of the unit is 13.5 ft., and the pressure drop for the water-side is less than one psi.

During cooling periods, the water-side is not altered much, except again in temperature; but the condensing refrigerant coefficient is substantially higher than in evaporation and calculated (from [75]) as 260 Btu/hr ft² °F.

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27 Also at times the flow rate is increased by 50 percent; the higher water speed then increases the heat transfer coefficient by 38 percent.
**Water tank:** The design of this unit has undergone several iterations in an attempt to maximize the free water heating fractions, and the contribution of the heat pump in general. As indicated in Figure 3-16, the water which the heat pump heats is stored in a main tank for general use; and a smaller tank with resistance heating provides hotter water for clothes washers and dishwashers. In order to keep the main tank temperature high and prevent a very intermittent operation, cold water is not added to the tank with each hot water usage, but rather the level is permitted to drop to a low at which time the tank is filled with one continuous operation. Pressurized water from such a non-full tank could be supplied with a circulation pump. At many locations, however, the city water pressure is high enough to eliminate this pump and operate with nitrogen (or air) stored in the tank. When the service pressure is 2/3 of the supply pressure, the gas storage volume is twice that for the water. The stored gas could either be separated with a diaphragm or have direct contact with the water, and even form an insulation layer. In any case, the performance of the unit is essentially independent of these selections.

The main tank has a natural convection desuperheating coil lined at the bottom of a storage section, and, below, a horizontal forced convection shell-and-tube heat exchanger that forms a separate compartment except for a small connection at one side. The refrigerant vapor leaving the compressor first flows in the upper coil which is always in the circuit. If the tank is being filled, the refrigerant then passes through the bottom tubes where it condenses while heating the city water which flows around the tubes in a countercurrent manner.
toward the connection to the storage. At other times, a 3-way valve allows the refrigerant to bypass this section and the reversing valve directs it to the condenser. The water temperature is prevented from overshooting during a long space application by raising the outlet pipe about six inches above the tank bottom and thus providing a permanent storage area. Of course, the tank filling starts when (or before) the level is lowered to six inches.

Thirty percent of the daily use is for washing, and requires 160°F water which is provided from a small separate tank whose inlet is connected to the main tank. The 160°F tank is kept full and its temperature is boosted by resistive heating. Since the hourly use profile for washing is not fixed, the 160°F tank is not included in the simulation, except an estimate of its daily energy requirement is calculated using the average temperature of the main tank.

In designing the heat exchangers of the tank, reduced compressor speeds are considered, but the full speed is selected for better performance and simpler control. The city water flow rate of 3150 lb/hr (6.32 GPM) is selected on the basis of the spring conditions when the interaction with space operations is nil. In the summer and fall even a greater flow rate would be acceptable due to higher inlet temperatures. In the winter when the starting point is low, this flow rate will result in colder water, but the temperature should be boosted up enough during long space heatings. The tank stores just enough water for the peak hour in a 8 ft³ (2x2x2) section. Gas storage for the pressure regulation adds 16 ft³ to the tank size. An empty tank takes 9.45 minutes to fill-up.
It turns out that the effect of desuperheating is less than expected. Even though the refrigerant enters at a rather high temperature, the vapor next to the walls is quickly cooled to the condensation temperature. Using the ratios of the heat transfer resistances, it is estimated that condensation on the wall at 120°F starts when the average R-22 temperature is still up at about 159°F. For an entering temperature of 170°F, this represents only 6.5% of possible energy transfer and achieves a temperature increase of merely 2°F for the water while the tank is being filled. The natural convection heating however becomes significant with the longer duration space heating or cooling applications.

Ten 0.8 inch ID tubes laid on the bottom surface of the tank form the desuperheater. With \( h_i = 37.6 \text{ Btu/hr ft}^2 \text{oF} \) on the refrigerant side, and \( h_o = 160 \text{ Btu/hr ft}^2 \text{oF} \) for the free convection to tank water, the \( U_o \) is 25.6 Btu/hr ft\(^2\). Then the surface area needed is 6.6 ft\(^2\), or 3 ft of each of the ten pipes. The performance of this unit during space operations will be very close to its performance during the water heating. During filling, about 93.5 percent of the energy is added to the water in the forced convection compartment. With average coefficients of \( h_i = 250 \text{ and } h_o = 500 \text{ Btu/hr ft}^2 \text{oF} \), the \( U_o = 167 \text{ Btu/hr ft}^2 \) and \( A_o = 65.5 \text{ ft}^2 \). Using unfinned 0.6 inch ID tubes requires 37 ft of each of the ten lines, or 19 passes for the two feet long water heater.
Controls: The heat pump unit is fully controlled automatically throughout the year. The reversing valve sets the refrigerant flow for heating or cooling mode. There are two 3-way bypass valves, one for the space unit, the other for the forced convection part of the water tank. The compressor has an off/on control and the air fan has off/normal, or high speed modes. The inlet valve on the water tank is of the off/on type, but the loop water unit has off/normal, and high flow settings. The conditioning-air fan speed and the loop water valve(s) are controlled by a pair of sensors that check the air and loop water temperatures. The normal (lower) settings are switched to their higher levels when the temperatures exceed certain predetermined values indicating high cooling demand. There are three level sensors in the main tank. The lowest, about six inches above the bottom, initiates the tank fill-up whenever the water drops to this level. The top one, at two feet above the lower one, stops the filling. The middle level (which can be eliminated by combining it with either of the other two) acts as the low-level during a space cooling run. There are temperature sensors to operate the resistance heaters in both the main and 160°F tanks. Table 4-1 summarizes the position of the controllers in various demand modes.
TABLE 4-1. The position of the controlling elements during the various modes.

<table>
<thead>
<tr>
<th></th>
<th>Space Heating</th>
<th>Water Heating</th>
<th>Space cooling with Tank Filling</th>
<th>Nat. Conv.</th>
<th>No Operation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reversing valve</td>
<td>Heat</td>
<td>Heat</td>
<td>Cool</td>
<td>Cool</td>
<td>Either</td>
</tr>
<tr>
<td>Loop water valve</td>
<td>Open</td>
<td>Open</td>
<td>Close</td>
<td>Open²/</td>
<td>Off</td>
</tr>
<tr>
<td>City water valve</td>
<td>Close</td>
<td>Open</td>
<td>Open</td>
<td>Close</td>
<td>Off</td>
</tr>
<tr>
<td>Tank 3-way valve</td>
<td>Bypass</td>
<td>No-bypass</td>
<td>No-bypass</td>
<td>Bypass</td>
<td>Either</td>
</tr>
<tr>
<td>Air 3-way valve</td>
<td>No Bypass</td>
<td>Bypass</td>
<td>No-bypass</td>
<td>No-bypass</td>
<td>Either</td>
</tr>
<tr>
<td>Air fan</td>
<td>On</td>
<td>Off</td>
<td>On²/</td>
<td>On²/</td>
<td>Off</td>
</tr>
<tr>
<td>Compressor</td>
<td>On</td>
<td>On</td>
<td>On</td>
<td>On</td>
<td>Off</td>
</tr>
</tbody>
</table>

¹/ Normal or high flow setting according to the cooling load size.
²/ Normal or high flow setting according to the loop heat temperature.

4-3. Simulation of the HPS

The simulation of the heat pump unit that was designed in the last section is carried out by a computer program discussed later in this section. Before the actual simulation however, additional input parameters values must be specified and operational decisions made. Those that are selected for this work are discussed next.

Data for the simulation

A truly continuous simulation is of course not feasible with a digital simulation; rather a time unit is to be selected such that it is long enough for the calculations to apply on the average, and yet short enough to approximate a continuous operation. The period should be much less than a day; and while data for a few time-dependent
variables are available on an hourly basis, there are no true data for periods shorter than an hour. Thus, one-hour is selected as the simulation time-step. Quantities for which genuine (hourly) information is not available are dealt with by compromising assumptions discussed below.

Climatic data: Hourly dry bulb temperature of the outside air was extracted from a U.S. Weather Bureau tape for 1952. This year was selected for its "average" appearance in a check in regard to monthly degree days. Among the climatic data needed for this work, only the outside air temperatures (and the degree hours) are truly hourly. The arriving loop and city-water temperatures are available from Chapter 3 as monthly averages, and are applied as a fixed value to all hours of a month. It would be possible to fit these data to smoothly varying curves and later calculate values for individual hours. Generation of such an artificial set however is not viewed as an improvement and is not adopted here. The hourly variations of these temperatures are actually quite small, and the only really bothersome point in using the averaged values is the sudden jumps between the months. Even that may not be very objectionable however as long as the simulation is not limited to a short period about these change-hours.
Load data: The inlet temperature of the conditioned air is taken as a constant at 72°F for all times. This assumes a constant inside temperature, no mixing of outside air at the unit and infiltration being a part of the load.

Daily hot water use in gallons and the usage fractions for each hour were used previously in Chapter 3. Here that profile is applied to all days, without any adjustments for the season or special days. For apartments this should be tolerable. The hourly space heating and cooling loads however are not available and difficult to specify. Suggested methods for a fully specified building and occupancy exist, but the time and effort their detail demands are forbidding for fast and average load estimates wanted in this work. Therefore, an approximate but very fast and simple load estimation procedure is specified as an alternative.

It is assumed here that the space heating loads have three major components: i) the net energy contribution from internal activities, ii) infiltration, and iii) the energy losses by convection and conduction through confining surfaces. Further, the internal contribution is supposed to remain at a roughly constant level while the other two components are both proportional to the difference of inside and outside temperatures. Then the hourly heating load may be directly connected to the outside air temperature in terms of the "degree hours" which not only indicate the temperature gradient but also account for the net internal contribution by using a base temperature value (65°F) that is lower than the actual inside temperature. The load
per degree hour is found from the division of the yearly use by the yearly degree-hour sum; then, for any hour, multiplying this ratio by the degree-hours of that hour gives an approximate but representative estimation of the hour's heating load. At the design temperature, this distribution would put the load 71 percent of the design load, because the yearly use estimation was done using the modified degree day method with $C = .71$. This difference is tolerable because the hourly load is an average-type load for that temperature, while the design load is the maximum; further, there are only few hours with near-design temperature. Estimating all hourly loads based on the design load and design temperature would be a poorer way, and result in too high a yearly use.

The direct influence of the outside air temperature on the hourly cooling load is even weaker than it is in heating because of the additional load components due to direct and diffuse radiation as well as the amplified role of the energy storage capacity of the building contents. Nevertheless, the use of the degree hour approach is extended to this mode due to the lack of a method that is better but still reasonably manageable. However, in this case, the design load and design degree hours seemed to be a better basis than the corresponding yearly use and degree hours since the yearly use was only roughly estimated. Hence, with a design load of 14 tons at 100°F outside, the cooling load per degree hour is 4900 Btu. The base temperature for the cooling degree hours is taken at 65°F as in the heating.
Operation of the Unit

The typical situations in which operational procedures are established are discussed in the following paragraphs. In making these decisions before the simulation, it was challenging to keep the convenience assumptions realistic, yet retain an efficient and functional program.

At times, the space and hot water loads could coincide, and it would be possible to meet them simultaneously. When the space needs cooling, it is in fact desirable to do this. To meet both heating requirements together, however, would inflate the heat pump size. Since the water heating is grouped as ten-minute operations, serving the heating demands one at a time seems preferable. The water heating is given priority, since its duration is shorter and its absence more noticeable. Also, since the hourly-measured outside air temperature is used here to determine the space load, there can only be one type of space load for any hour.

A more subtle point arises because of the interaction between the space and hot water loads. Even if the correct loads for an hour are known, their profile within the hour could not be specified. It will be assumed that i) all hot water use precedes any space load of that hour, ii) the tank is always filled-up exactly in 9.45 minutes. (i.e., no water usage during the filling), and iii) no tank middle level is used (that is, water is added to the tank at all cooling hours until the tank is full or the cooling ends).

For a single hour, these assumptions may appear to be quite rough. Yet, an hour is not an isolated period; and the contribution
of the preceding hour is expected to be such that the above assumptions will in effect be quite applicable on the average. Then, for a day or longer period, the totals and the variables that remain essentially the same should have accurate values. Variables such as the tank level or temperature which fluctuate from hour to hour preserve most of their meaning under a slightly modified view: With a simulation run for a period longer than a few hours, these variables indicate the nature and frequency of various types of hours that would occur in an actual operation during the same period.

**Computer simulation program**

Now the structure and execution of the FORTRAN simulation program can be discussed. The actual program statements and three separate lists describing the input, output, and other major program variables are contained in Appendix D. Some of the program and design information presented below (or previously) are also included as comment statements in the program listing.

**Structure:** The simulation program consists of a main program (SOS), four subroutines,(DWH,T, SPHT, SPCL, HPU), and two functions (H,P). Figure 4-1 indicates the component hierarchy. The main program first collects the initial data, and calculates several universal variables. Then, for each hour, the main program finds the loads, calls the subroutines as required and combines their result for output. Each basic mode has its own subroutine: DWH,T for domestic water heating, SPHT for space heating, and SPCL for space cooling. These subroutines contain design information, and handle decisions and calculations
special to the mode. Balancing the compressor with the heat transfer components is handled as a separate, fourth subroutine (HPU), to avoid its duplication at each mode. The two functions estimate R-22 properties at various stages; P is for saturation pressure as a function of temperature, and H is for saturated liquid and saturated or superheated vapor enthalpies as a function of temperature and pressure. The saturated liquid properties are found from specially developed equations of good consistancy with the tables (multiple correlation coefficients are better than 0.99999). The enthalpy of the vapor is found from equations given in Reference 76. These equations are included in the program listing.

Fig. 4-1. Hierarchy of the subprograms in the HPS simulation program.
Execution: The hourly simulation is done a day at a time. Days can be input in any order, and the simulation stops when the data is exhausted. For runs with consecutive days, the tank level and temperature information are carried through, and a summary list is formed by calculating seasonal averages of fluctuating variables such as the free water heating fraction, tank temperature and gross electricity demand. In other runs, the tank starts half-full at $130^\circ F$, and naturally the days are not averaged.

Hot water is checked first. The use, which is specified at $130^\circ F$, is adjusted to the actual tank temperature. When the tank contains sufficient water for the city, the program updates and jumps to the space operation. Otherwise, the available water is used up, the tank is refilled, the remaining demand is met, and the case is updated. Although rare, the program can properly deal with a second fill-up in the same hour, when necessary. For a given design and loop inlet temperature, the performance for the water heating depends on the inlet temperature of the city water. Thus, when this temperature is the same for all hours of a month, computation time is saved with the specification of $IW=0$ which causes the water heating calculations to be done only once for every new month; then the results are stored and used throughout the month as needed.

Next, the space load is checked and when a load exists, the appropriate subroutines are called to meet it. Then the daily totals

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$28$ Actually, the gross hot water use rate was specified for an average temperature of $140^\circ F$. Here, since 30 percent of the hot water is provided at $160^\circ F$, the water in the main tank needs to be only at about $130^\circ F$. 

and averages are updated, and the hourly information is printed to complete the hour. Daily information is printed after each 24-hour period.

The mode subroutines are basically quite similar. The flow rates and UA values for loop and air or tank unit is established here. The water-side heat transfer coefficient for the loop unit is calculated as a function of its temperature in the main program since it is common to all modes. Other coefficients and corresponding UA values are calculated or specified in the subroutines. Actually, the program starts out with the design values, and when any of them changes, the new value is found by first calculating the ratio of it to the design value and then using it as a multiplicative correction factor on the design value. In both heating modes, the operation type and the flow rates are fixed; hence the subroutines DWHT and SPHT are fairly simple. However, in the space cooling first the role of air and loop units change. Also, for better performance, the loop flow rate is increased 50 percent when its temperature is high, and for higher capacity, the air flow rate is stepped up 50 percent when the outside air temperature (thus, the load) is high. Further, now the space-tank interaction is more involved. At the start, the loop is turned off and the tank is filled. The natural-convection heating for this period is calculated with average capacity and temperature. Usually space cooling load is long enough to last until the tank is full, and during the remaining load, the loop unit must be used as the condenser. However, there still is natural convection to the tank which is calculated in six-minute time-steps to
update the tank temperature. At the end, time averaged performance parameters for the hour are calculated.

Once the UA values and other auxiliary parameters are calculated, the mode subroutines call the HPU to balance the compressor with the heat exchangers. This is done by iterating for condenser and evaporator temperatures. A guess for their values gives the compressor capacity and work requirement; these and heat transfer units in turn dictate a new set of values for the temperatures. It is found by experimentation that the optimum convergence speed is obtained when the values of the next attempt are formed from 70 percent of the last estimation and 30 percent of the previous guess. Including much different than 30 percent of the old guess slows or prevents the convergence, and if the last estimation is directly used as the next guess, the iteration almost always diverges. The number of iterations per hour is also reduced by starting with the solutions of a previous hour, if they are available. In this way, slightly fluctuating results are obtained in the first or second iteration.

After the balancing, HPU calculates R-22 properties for the estimation of the natural convection to the tank. During space heating, the unit actually should run a little longer than would be estimated from the performance and the load, because not all of the energy goes to the space. In early runs, an iteration is used to find the effect of the desuperheating occurring at the tank; but it is found that the extra time is usually very close to 2 percent, and this constant fraction is adopted for all space heating runs to gain
some computation efficiency. During space cooling, the loop unit may be in or out. The mode with the loop on is signaled to the HPU by sending a negative value (-99) for COPA (thus avoiding a new variable). For all modes, the HPU calculates the COP values before the command is returned to the mode subroutine.

4-4. Simulation Results

The results of the computer simulation-program runs can be viewed in two parts: i) The approval or modification of the previous design and assumptions based on the information from trial runs, and ii) numerical results giving the year around performance estimation of the modified system. Some of the changes made in the unit operation and the computer logic were already pointed out. Here, several major system adjustments are discussed in detail; and then the numerical results are presented and examined. Before the results are presented, however, the actual simulation periods within a year that are used to obtain the results are reviewed in the next paragraph.

The output values of some variables are unchanged within each month because they are determined simply by the design and monthly-valid input; therefore simulations with one random-day from each month are sufficient to determine the condenser and evaporator temperatures, the COPs, and the output temperatures in both heating and various cooling modes that may occur. For heating-only months, the monthly averaged values of certain other variables (such as the water tank temperature, electrical demands of the compressor and
resistance heaters, unit operating duration, and the thermal pollution decrease) can also be obtained with one-day simulations if, instead of a random or artificial day, a day composed of monthly-averaged hourly temperatures is used. For months with cooling periods, however, a complete simulation is essential to find values of cooling-load related variables (such as the averages of the cooling COP the free water-heating fraction, the tank temperature, operating duration, and thermal pollution decrease in months with cooling). Otherwise, the cooling and heating loads occurring at different days would erroneously be cancelled from one another with the use of a composed, monthly-averaged day; or, if a random-day is used, the results would be unreliable due to the possibility of enormous variation within a month. For this work, the check-runs are done by using typical days selected one from each month, and final numerical results are taken from a full simulation with data from the year 1952.

**System Design and Operation Adjustments**

The basic question here of course was whether the design will have acceptable performance without major modifications. Early results indicated an essentially positive reply; only in cooling modes certain flow rate adjustments (that were built into the program as already mentioned in the "execution" discussion) are applied at times to improve the performance.

One modification was to increase the air flow rate during high cooling loads. The high air flow rate increases the cooling capacity at the price of slightly reduced performance due to higher $T_{con}$ and $T_{ev}$. 
Nevertheless, near-peak cooling loads are met with this mode rather than with an increased size which would give an oversized unit for most of the year. It is found that the higher air flow rate is needed when outside temperature exceeds 94°F (load exceeds 145,000 Btu/hr.). With the increased air flow rate, loads occurring with outside temperatures up to 98°F may be met; any excess cooling load is carried to the next hour.

Also, to improve the performance, the loop flow rate is increased by 50 percent during cooling with incoming loop temperatures over 100°F. This will occur during only the few cooling hours in a mainly heating season (in April, May, and October), because June 1 through September 30 is taken as the cooling season, and the loop then carries the cooler water from the tower rather than the condenser water. During summer, most of the heating is of water and the lower loop temperatures are acceptable since the city water is warmer in those months, and, also, significant free water-heating occurs with longer cooling loads.

In regard to water heating during the cooling hours, the results showed that while being refilled the main tank can alone perform as the condenser. At the outset of cooling, then, city water to the main tank is turned on and the loop flow is kept off. This cooling mode exists until the tank is full or the cooling load is met. In addition to the sizeable benefit of free water heating in this manner, this mode presents another advantage. Because the city water is colder than the loop water, the cooling COP for this mode is higher (2 percent in August, but over 40 percent in April) than in the other cooling
modes. The tank filling rate is quite large, so the water is heated to only about 110°F. Actual tank temperature will be higher though since water is mixed with warmer tank content and natural convection heating continues for the balance of the cooling load. In order to accelerate this convective heating, circulating the tank water for a brief period after the fill-ups is considered; but the simulations with such circulation showed benefits that were too small to warrant additional controls and pumping.

**Numerical Results**

Table 4-2 lists the numerical results for the variables that were examined in regard to modification or acceptance of the system. At each month, two values are given for each variable, the upper number is the result from the simulation, the lower one is taken from the analysis of Chapter 3. In general, examination of the simulation entries indicates an acceptable operation for the system, and comparing the results of the two analyses indicates overall confirmation of the previous analysis but also points out several correction areas. Specific results are examined in the following discussion.

**Loop water temperature.** The arriving loop water temperatures, which are common to both analyses, are listed here, for easy reference, along with the temperature drop during heating (-ΔTₜ₁), and the rise during cooling (ΔTₜ₂). The comparison of the pairs shows that the winter values are almost the same while in the warmer half of the year, the previous assumptions are too high both in heating,
TABLE 4-2. Simulation results for the system-check variables. (For each month the upper line gives the simulation results, and the lower line is from Chapter 3 analysis.)

<table>
<thead>
<tr>
<th>MONTH</th>
<th>LOOP TEMPERATURE</th>
<th>SPACE COOLING</th>
<th>SPACE HEATING</th>
<th>WATER HEATING</th>
<th>TANK TEMPERATURE</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$T_{in}$</td>
<td>$-\Delta T_h$</td>
<td>$\Delta T_c$</td>
<td>$T_{con}$</td>
<td>$T_{ev}$</td>
</tr>
<tr>
<td>J</td>
<td>105</td>
<td>40</td>
<td>22</td>
<td>40</td>
<td>121</td>
</tr>
<tr>
<td>F</td>
<td>106</td>
<td>41</td>
<td>40</td>
<td></td>
<td>122</td>
</tr>
<tr>
<td>M</td>
<td>108</td>
<td>41</td>
<td>40</td>
<td></td>
<td>123</td>
</tr>
<tr>
<td>A</td>
<td>111</td>
<td>42</td>
<td>40</td>
<td></td>
<td>122</td>
</tr>
<tr>
<td>M</td>
<td>113</td>
<td>44</td>
<td>40</td>
<td></td>
<td>125</td>
</tr>
<tr>
<td>J</td>
<td>79</td>
<td>31</td>
<td>40</td>
<td></td>
<td>116</td>
</tr>
<tr>
<td>J</td>
<td>82</td>
<td>32</td>
<td>40</td>
<td></td>
<td>119</td>
</tr>
<tr>
<td>A</td>
<td>82</td>
<td>32</td>
<td>40</td>
<td></td>
<td>120</td>
</tr>
<tr>
<td>S</td>
<td>78</td>
<td>31</td>
<td>40</td>
<td></td>
<td>118</td>
</tr>
<tr>
<td>O</td>
<td>112</td>
<td>43</td>
<td>40</td>
<td></td>
<td>123</td>
</tr>
<tr>
<td>N</td>
<td>108</td>
<td>40</td>
<td>40</td>
<td></td>
<td>123</td>
</tr>
<tr>
<td>D</td>
<td>106</td>
<td>41</td>
<td>40</td>
<td></td>
<td>122</td>
</tr>
</tbody>
</table>
and to a lesser degree, in cooling. The very low $\Delta T_c$ values found in the simulation for the transition months (April, May, and October) result from the flow rate increases applied in these months.

**Space cooling.** In simulation, the evaporator temperatures in general and the condenser temperatures in the transition months turned out to be lower than what were assumed earlier. The resulting COPs, however, differ only in the transition months when they are higher than the previous estimates. It should be noted that the listed simulation results are approximate monthly averages. Due to the varying durations of the several modes that can exist at each cooling hour, the (averaged) hourly values fluctuate (up to 40 percent in the transition months, as indicated previously) from hour to hour.

**Space heating.** The results show that the evaporator temperatures in summer, and the condenser temperatures in winter were assumed too low. As a result, the previous COP values are too high all year except for four summer months when their prediction is too low.

**Water heating.** This mode is very similar to space heating.

**Tank temperature.** Except in summer, the average tank temperatures are much higher than expected. For October and November, it exceeds $140^\circ F$, and the yearly average from simulation is $132^\circ F$. The monthly minimum and maximum temperatures are also shown for approval. Both extremes occur only a few hours in each month; mostly, there is only slight fluctuation around the average. The minimums occur just after the tank is refilled. Usually, the temperature increases above this minimum by natural convection if any space load exists, or by
resistance heating if the minimum is below the allowed-minimum of 120°F. The maximums occur almost always at 5 or 6 a.m. on days with a low tank level and long space heating. Summer afternoons with very long space cooling also have fairly hot tank temperatures. Dead storage height. The 6-inch storage area that was provided could actually be much less in summer because the tank is filled in all cooling hours, and the tank temperatures after filling are too low to overshoot even in the coldest summer mornings if the water level is an inch or more. Too much water is not desirable in summer because it unnecessarily lowers the tank temperature and thus increases the resistance heating. In winter, on the other hand, lowering the storage to 4 inches increases the maximum water temperature 2 or 3°F, and lowering it to 1 inch is unacceptable. Assuming a fixed level is to be maintained all year, the storage height is compromised at 4 inches. If the maximums given in Table 4-2 are considered too high, either the storage could be increased back to 6 inches or more, or a new control that will add some cold water whenever the tank temperature is too high could be incorporated into the design. In this work, neither is considered essential, because the high tank temperatures occur infrequently and heat losses from pipes should lower the delivery temperature and give enough warning time.

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29 For Table 4-2, the temperatures before the first filling at each month are ignored since they are influenced by the arbitrary starting conditions.
The results giving the final performance of the system are collected in Table 4-3. Again the upper lines are from the simulation while the lower lines are from a repeat of the Chapter 3 analysis for only the apartments in the 129,000 city. Overall, the predictions of the two analyses appear close. A brief look at the individual items follows.

**Free water heating fraction.** As anticipated in the previous analysis, the potential of free water heating is substantial. The simulation results are somewhat lower than the rough estimates made in Chapter 3. The wide difference in June is mostly due to low cooling load for this month is 1952.30

**Resistive water heating.** In both cases, a very large portion of this load comes from the 160°F water. In Chapter 3, the minimum (and the average) tank temperature was assumed at 120°F. As seen in Table 4-2, the minimum actually falls below 120°F only in Spring; but the average tank temperature found from simulation is 12°F higher, and the gross energy requirements for resistance heaters are therefore lower in the simulation. Nevertheless the yearly resistance heating averages 751 W which is about 16 percent of the gross electricity use of the system.

**Thermal pollution.** For both the winter months and yearly average, the decrease in thermal pollution, implied by the negative numbers, are estimated to be almost the same by both analyses. At other times

30. The monthly total cooling degree hours for June is 859 in 1952. The ten-year average used in Chapter 3 was 1,300.
TABLE 4-3. Simulation results for the system-performance variables.
(For each month, the upper line gives the simulation results, and the lower line is from Chapter 3 analysis.)

| Month | Free Resistive Resistive Thermal Electricity Use |
|-------|-----------------------------|-----------------------------|-----------------------------|
|       | W. Heating (BTU/HR) | W. Heating (BTU/HR) | Pollution (BTU/H) | Per Unit (KW) | 129,000 City (MW) |
| J     | .0 | 2340 | -80016 | 6.56 | 30.21 |
|       | .0 | 2859 | -75082 | 5.63 | 25.93 |
| F     | .0 | 2426 | -68048 | 5.68 | 26.18 |
|       | .0 | 2859 | -68104 | 5.11 | 23.55 |
| M     | .0 | 2291 | -60124 | 5.07 | 23.37 |
|       | .0 | 2859 | -59750 | 4.50 | 20.73 |
| A     | .062 | 2230 | -43435 | 4.13 | 19.03 |
|       | .0 | 2837 | -43951 | 3.96 | 18.26 |
| M     | .121 | 2210 | -30380 | 3.92 | 18.05 |
|       | .140 | 2387 | -25775 | 3.84 | 17.70 |
| J     | .140 | 4384 | -23943 | 4.12 | 18.98 |
|       | .400 | 2822 | -10295 | 3.56 | 16.40 |
| J     | .412 | 3523 | +8545 | 4.82 | 22.20 |
|       | .550 | 2815 | +18317 | 4.61 | 21.23 |
| A     | .385 | 3084 | -2626 | 4.30 | 19.83 |
|       | .500 | 2785 | +15080 | 4.33 | 19.96 |
| S     | .362 | 2900 | -539 | 4.38 | 20.19 |
|       | .400 | 2696 | -1141 | 3.92 | 18.06 |
| O     | .017 | 1457 | -23853 | 3.77 | 17.35 |
|       | .0 | 2778 | -35915 | 3.56 | 16.41 |
| N     | .0 | 1713 | -67077 | 5.55 | 25.59 |
|       | .0 | 2822 | -59965 | 4.50 | 20.74 |
| D     | .0 | 2215 | -67562 | 5.64 | 26.01 |
|       | .0 | 2859 | -69090 | 5.17 | 23.82 |
| Yearly Average | 2564 | -38255 | 4.83 | 22.25 |
|       | 2819 | -34639 | 4.39 | 20.23 |
the estimates differ, depending on the actual magnitude and distribution of space cooling load.

**Electricity use.** Comparing the gross electrical use per unit indicates about 10 percent higher requirement in the simulation. (Only in August the simulation requires less.) The difference increases in going from summer to winter months as would be expected from the COP comparisons via Table 4-2. For another view, this basic-unit performance is extended in the last column of Table 4-3 to the full-size operation in the 129,000 city. In doing this, identical performance is assumed in all 4608 (192 A-blocks with 24 units per A-block) heat pump units.

**Yearly space loads.** The comparisons made with the results given in Table 4-3 actually need some qualification because the total loads met by the two analyses are not identical. The monthly and yearly load totals are given in Table 4-4 together with the degree hour totals. Again, the upper lines are from the simulation while the lower lines are from Chapter 3. The degree hour values differ because they are from different years, and because Chapter 3 values for heating are converted from heating degree days. Actual heating degree hour sums in summer are much greater than 24 times degree days because many days with above 65°F average temperature do have below 65°F hourly temperatures.

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31 The simulation values are from 1952; the Chapter 3 values are the long-term averages as specified in Table 3-7.
TABLE 4-4. Total space heating and cooling loads and degree hours.
(For each month the upper line gives the simulation results, and the lower line is from Chapter 3 analysis.)

<table>
<thead>
<tr>
<th></th>
<th>Monthly Space Heating Degree Hours</th>
<th>Monthly Space Cooling Degree Hours</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Load (MTBU)</td>
<td>Load</td>
</tr>
<tr>
<td>J</td>
<td>22168 55.09</td>
<td>0</td>
</tr>
<tr>
<td></td>
<td>19800 49.20</td>
<td>0</td>
</tr>
<tr>
<td>F</td>
<td>16450 40.88</td>
<td>0</td>
</tr>
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<td></td>
<td>15456 39.80</td>
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<tr>
<td>M</td>
<td>14743 36.66</td>
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<td></td>
<td>14064 34.93</td>
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</tr>
<tr>
<td>A</td>
<td>9137 22.71</td>
<td>304</td>
</tr>
<tr>
<td></td>
<td>9504 23.63</td>
<td>1.46</td>
</tr>
<tr>
<td>M</td>
<td>6774 16.83</td>
<td>855</td>
</tr>
<tr>
<td></td>
<td>5880 14.60</td>
<td>4.10</td>
</tr>
<tr>
<td>J</td>
<td>4734 11.76</td>
<td>859</td>
</tr>
<tr>
<td></td>
<td>2520 6.24</td>
<td>4.12</td>
</tr>
<tr>
<td>J</td>
<td>2096 5.21</td>
<td>3976</td>
</tr>
<tr>
<td></td>
<td>600 1.49</td>
<td>19.08</td>
</tr>
<tr>
<td>A</td>
<td>2058 5.11</td>
<td>3252</td>
</tr>
<tr>
<td></td>
<td>672 1.65</td>
<td>15.61</td>
</tr>
<tr>
<td>S</td>
<td>3304 8.21</td>
<td>2939</td>
</tr>
<tr>
<td></td>
<td>2736 6.77</td>
<td>14.11</td>
</tr>
<tr>
<td>O</td>
<td>6321 15.71</td>
<td>1158</td>
</tr>
<tr>
<td></td>
<td>8520 20.00</td>
<td>5.56</td>
</tr>
<tr>
<td>N</td>
<td>17619 43.78</td>
<td>0</td>
</tr>
<tr>
<td></td>
<td>14328 35.62</td>
<td>0</td>
</tr>
<tr>
<td>D</td>
<td>17634 43.82</td>
<td>0</td>
</tr>
<tr>
<td></td>
<td>18072 43.80</td>
<td>0</td>
</tr>
<tr>
<td>Total</td>
<td>12308 305.77</td>
<td>13343</td>
</tr>
<tr>
<td></td>
<td>111672 277.73</td>
<td>10773</td>
</tr>
</tbody>
</table>


The heating degree hour sum is 10 percent higher in 1952; the yearly heating load is also 10 percent higher in the simulation, because the simulation heating loads are calculated by using degree hours from 1952 and the yearly heating load per heating degree day estimate from Chapter 3.\(^{32}\) On the other hand, the simulation cooling load is 15 percent less even though the degree hours in 1952 are 24 percent more than the average. If the cooling load were directly proportional to the degree hours, the load in Chapter 3 would have to be 51.71 MBtu, or 42 percent less than the estimation. Part of this discrepancy in cooling loads may be due to the inaccuracy in the proportionality assumption; however, it is apparent that the yearly cooling load estimation of Chapter 3 is too high, even if not as much as 42 percent.\(^{33}\)

It was found in Chapter 3 that for yearly averages, the space heating required 38 percent of the electricity while space cooling required 19 percent. Using these fractions, a year with 10 percent greater space heating load, but 15 percent less cooling load should require a net 1 percent more electricity. Therefore, the effect of meeting different loads for the two analyses is not much in the yearly electricity use comparison, and could be expected to be

\(^{32}\)Clearly the yearly space heating loads would be the same in both analyses if the degree hour sums were equal. \((305.77 \times 111672/\approx 277.7.)\)

\(^{33}\)The cooling load factor used in Chapter 3 for the apartments was 0.05, meaning over 435 hours of cooling operation per year which in fact sounds too long. The load factor should apparently be revised at below 0.04.
negligible in other cases as well.

4-5. **Conclusions**

The simulation results presented in the previous sections demonstrated the technical feasibility of the heat pump operation in conjunction with a thermal-electric power plant. The closeness of the results from the two analyses adds overall reliability to both. The annual performance differed only 10 percent in the two analyses, which is not very large considering the scope and nature of the operation. Since no special efforts were made to optimize the simulated design, it appears likely that the differences existing between the performances from the two analyses can, if desirable, be narrowed and even reversed with the improvement of the design. Improving the heating performance by lowering the condenser temperatures in winter could be one of the first attempts. Nevertheless, the profiles provided by the simulation, in particular for free water-heating fractions, loop temperature changes, and average tank temperatures are much more reliable than their previous tentative values; and this information can be useful even in improving the corresponding assumptions for other climates or designs.

The two flow rate adjustments made to improve the performance need further consideration in the light of the extended simulation. There were less than 10 hours when the outside air temperature exceeded 95°F, and none were over 98°F. The high fan speed and the related controls would not be needed if the inside temperature could be permitted to rise slightly over 72°F for those few hours with near-peak
outside temperature. Also, the increased loop flow rate saved only 13 kWh in the warmest May-day with 6.63 hours of cooling operation spread to 13 hours. As can be found from Table 4-5, the monthly electricity saving is 37.2 kWh while the additional loop water pumping should be about 1 kWh. Therefore, the net saving is only 36 kWh which represents a 1.2 percent saving in the gross electricity use during May. If this saving is considered too small to justify the second setting of the loop value and the coupled controls, they can be taken out of the system without affecting the rest of the operation. Both flow rate adjustments are left in the simulation program as options; they can be simply excluded from future simulation runs.

34 The extra pumping would be about 4.5 kWh if the loop were on in all cooling minutes. However, during most of the cooling, the water tank is being filled and the loop is off. This last fact has a basic role in lowering the benefit from the loop flow rate adjustment which appears more useful from a comparison of the COP values in loop-on mode (minimum values in Table 4-5).

35 It is assumed that, since the increased flow rates will occur during off-peak times, they might be supplied without increasing the size of the distribution network pipes or pumps.
TABLE 4-5. The effects of high loop flow rate operation during space cooling in May 1952.

<table>
<thead>
<tr>
<th></th>
<th>Loop Flow Rate</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Normal</td>
<td>50% Increased</td>
<td></td>
</tr>
<tr>
<td>Max. Cooling $T_{\text{con}}$ (°F)</td>
<td>148</td>
<td>139</td>
<td></td>
</tr>
<tr>
<td>Min. Cooling $T_{\text{con}}$ (°F)</td>
<td>111</td>
<td>111</td>
<td></td>
</tr>
<tr>
<td>Max. Cooling $T_{\text{ev}}$ (°F)</td>
<td>46</td>
<td>45</td>
<td></td>
</tr>
<tr>
<td>Min. Cooling $T_{\text{ev}}$ (°F)</td>
<td>40</td>
<td>40</td>
<td></td>
</tr>
<tr>
<td>Max. Cooling COP</td>
<td>3.43</td>
<td>3.43</td>
<td></td>
</tr>
<tr>
<td>Min. Cooling COP</td>
<td>2.02</td>
<td>2.30</td>
<td></td>
</tr>
<tr>
<td>Loop Temperature rise (°F)</td>
<td>33</td>
<td>22</td>
<td></td>
</tr>
<tr>
<td>Avg. tank temperature (°F)</td>
<td>136.4</td>
<td>136.2</td>
<td></td>
</tr>
<tr>
<td>Free water-ht. fraction</td>
<td>0.124</td>
<td>0.121</td>
<td></td>
</tr>
<tr>
<td>Heat pump operating for cooling (hr.)</td>
<td>31.05</td>
<td>29.87</td>
<td></td>
</tr>
<tr>
<td>Gross electricity use (kwh)</td>
<td>3.972</td>
<td>3.920</td>
<td></td>
</tr>
</tbody>
</table>
CHAPTER V

CLOSURE

5-1. Conclusions Summary

The major topics that were investigated and the conclusions that have been reached in the previous chapters are summarized here.

Chapter 1 defined the low-grade demands and reviewed the cause of wastefulness built into many of the conventional systems in use to meet these demands. The importance of developing more effective systems is pointed out by noting the sizeable share of space heating/cooling and similar needs in the national and domestic allocations of the energy resources. This chapter then presented the concepts and benefits of two systems (BOS and HPS) both of which can, in conjunction with a thermal-electric power plant, meet these needs effectively.

Chapter 2 developed a general evaluation and comparison of the BOS, HPS, and several conventional system performances. With this information, the operation of these systems in widely different applications or conditions can be estimated and compared, and the effects of certain parameter changes can be projected. Overall, it is shown that in the ideal analyses the BOS is favored over the HPS; but that for the actual systems, while either system is superior to all conventional alternatives, the best system is determined by the application specifics. Another way of comparing the actual BOS and HPS was also provided here by obtaining the critical COP values which equalize the overall HPS performance with that of the BOS.
Chapter 3 particularized the BOS and HPS performance analyses to meeting space heating/cooling, and hot water requirements of two specified cities with served populations of 21,500 and 129,000. Using the calculated monthly energy loads, and the system and distribution network designs, the savings in electricity and the reductions in thermal pollution are estimated. The economic evaluations then concluded that the BOS can compete with conventional systems, up to at least 21-mile plant-city distances for the 129,000 city; and that the HPS could also be economically feasible up to at least 11-mile distances.

Chapter 4 attempted to answer some of the uncertainties in the previous HPS evaluation. An engineering design of a typical heat pump unit is developed and its components are simulated to check the operation and to re-estimate the performance throughout a year. The simulation results being, on the average, close to previous results added confidence to both analyses as well as providing some new information.

As a whole, the work here suggests serious considerations of future steps that will culminate in the actual applications of both systems. Certain possible extensions of this work toward that end are discussed next.

5-2. Recommendations

The following recommendations are made to improve and extend the present concepts and information.
The heat pump design used in the simulation can be improved to give better overall performance. In particular, one could redesign the air-unit to lower the condenser temperatures in heating and check the effect of i) using a larger main hot water tank to increase the free water heating fractions, ii) regulating the amount of permanent water storage, iii) regulating the temperature of the loop water, iv) using other refrigerants, and v) altering the size of the heat pump unit. Regarding the last point, actually, one heat pump per household was envisioned at the outset. However, if the heat pumps will serve a larger group, as they did in the simulation here, mixing of the air may be objectionable. Then the heat pump design should be converted to a water-to-water operation.

The methods of estimating the hourly space loads, especially for cooling, needs revision. Examining various load components in representative cases, and introducing the necessary corrective measures may lead to a sufficiently improved method without becoming overly complicated and specific. The revised program then could be used to resimulate the improved (or present) model with climatic data from several other years. Such an extended simulation would yield more reliable results especially for free water heating, thermal pollution and total load values. (With the present model and program, the cost of a 30-day simulation averaged about five dollars at the OSU's CDC-3600 computer.)
Results that depend on the interactions within an hour cannot be improved much in a computer simulation due to inadequate data. To obtain more information regarding these parameters, and check the designs and other results, actual test runs should be conducted for both the BOS and HPS models. While these operations could be done with simulated loop water temperatures and so on, it would be more appropriate to subject the equipment to the actual environment by using them to meet the demands of buildings (visitor center, staff offices, and living quarters) at an operating power plant site.

Both the BOS and the HPS could be established in conjunction with sources other than the power plants. Operations with industrial plants, geothermal fields, and the like should be considered to broaden the application potentials.

The distribution networks of either system are easier to construct in the newly developing communities. The possibility of using one of these systems should encourage new city developments around power plants, and small local plants at large urban development sites. Because insulation is not essential for the HPS network, that system perhaps could be applied in established cities easier. A possibility is using the existing city water lines to deliver water for both domestic and heat pump consumption. In such a case, if the water is warm, a small amount would be cooled for purposes such as drinking. While the city water colder than the condenser water would lower the heating performance, the cooling performance would improve. The basic problems with such an open-loop system would be high cost of water, increased load for sewer handling systems, and possibly the shortage of water.
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60. Glicksman, L. R. "Thermal discharge from power plants." ASME paper No. 72-WA/Ener-2.


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69. Private communication with Prof. W. F. Stoecker, University of Illinois.


APPENDICES
APPENDIX A

SEASONAL VALUE OF $T_0$

As a result of the relationship given in Equation (1-1), the dead state temperature $T_0$ is involved in many performance formulations of any system undergoing heat transfers. However, the real numerical value of $T_0$ could be difficult to specify in many practical calculations. Even when the instantaneous values are known, they must be properly averaged in monthly or seasonal evaluations. To illustrate, assume the system temperature $T$ remains fixed (as would be in space heating) but the heat transfer rate and the dead state temperature vary with time, i.e., $T = \text{constant}$, $\dot{Q} = \dot{Q}(t)$, $T_0 = T_0(t)$.

Then for a duration $\Delta t$, the effective values (denoted by a *) become

$$Q^* = \int_0^{\Delta t} \dot{Q}(t) dt$$

(A1)

and

$$A_Q^* = \int_0^{\Delta t} (1 - \frac{T_0(t)}{T}) \dot{Q}(t) dt.$$  

(A2)

Introducing these into Equation (1-1) yields

$$T_0^* = \int_0^{\Delta t} T_0 \dot{Q} dt/ \int_0^{\Delta t} \dot{Q} dt.$$  

(A3)

That is the effective dead state temperature for a period is obtained by time averaging the $T_0$ with $Q(t)$ as the weight. Two cases with special $Q(t)$ will be considered:
CASE I. \( Q(t) = \text{constant}. \)

This could be applicable to many industrial operations. Here Equation (A3) reduces to

\[
T_0^* = \frac{\int_0^{\Delta t} T_0 \, dt}{\int_0^{\Delta t} dt} = T_{0,m}.
\]

That is the effective \( T_0 \) is the ordinary average of \( T_0 \) for the period.

CASE II. \( Q(t) = K(T - T_0) \), when \( T > T_0 \) (\( K \) is independent of \( t \))

\[
= 0 \quad \text{, otherwise.}
\]

This is similar to relating a space heating load to the degree days, or degree hours. In this case Equation (A3) yields

\[
T_0^* = \frac{T.T_{0,m} - (T_0^2)_{m}}{T - T_{0,m}} \quad (T_0 < T)
\]

Similar developments also apply to cooling. However, accurate functions of heat transfer are in general more complex and difficult to specify. In this work, the maximum averaging period is a month (rather than a year) for which the difference between \( T_{0,m} \) and \( T_0^* \) is estimated (using actual weather data and above equation) to be only several degrees, and is ignored for simplicity and generality.
APPENDIX B

NOTES ON GENERAL PERFORMANCE CONSIDERATIONS

B-1. Limiting Values of $R_h$ and $R_c$

When an energy source rate ($E_F$) is specified, the thermal requirement that could be met from the power plant is limited. A stricter limitation results if energy is to be supplied solely from condenser outlet or the steam bleed off of the power plant. However, since the cycle efficiencies rarely reach as high as 0.5, the limiting loads, and their corresponding $R$ values, are expected to be outside of the range investigated in Chapter 2 (0 to 0.4). Nevertheless, the limiting value expressions for $R_h$ and $R_c$ are now presented.

**Heating --- BOS:** Maximum heating will result when all the steam is bled off. Then the cycle efficiency is $\eta_h = \dot{E}_e/E_{cyc}$, and the thermal energy output is

$$\dot{Q}_{h-max} = \dot{E}_{cyc} - \dot{E}_e = \dot{E}_{cyc}(1 - \eta_h) \quad (B1)$$

so that

$$R_{h-lim} = \frac{\dot{Q}_{h-max}}{\dot{E}_{cyc}} = 1 - \eta_h \quad (B2)$$

In the ideal operation bleed-off is at $T_h$, while for the actual operation selected here it is set at $720^\circ R$ and $770^\circ R$. In either case, any greater demand will necessitate bleeding at higher temperatures, greatly reducing the efficiency.
Heating --- HPS: There are two energy sources for the heat pump (electricity and thermal energy from the loop water); either can limit the system. The energy available from the loop water is

\[ \dot{E}_{\text{loop}} = \dot{E}_{\text{cyc}} - \dot{E}_{e} = \dot{E}_{e}(1 - \eta_{C})/\eta_{C} \] (B3)

with which the heat pump could supply

\[ \dot{Q}_{h-\text{max}} = \dot{E}_{\text{loop}} / \left(1 - \frac{1}{\text{COP}}\right) = \frac{\dot{E}_{e}(1 - \eta_{C}) \text{ COP}}{\eta_{C}(\text{COP} - 1)} \] (B4)

At the changeover from an electrical limit to a thermal limit, the power plant's total electricity generation will just meet the electricity requirement of the heat pump supplying the load given in Equation (B4):

\[ \dot{E}_{e} = \frac{\dot{Q}_{h}}{\text{COP}} = \frac{\dot{E}_{e}(1 - \eta_{C})}{\eta_{C}(\text{COP} - 1)} \] (B5)

Upon cancellation of \( \dot{E}_{e} \) and rearrangement, the upper limit of COP that could result in an electrical limit is found to be

\[ \text{COP} = 1/\eta_{C} \] (B6)

Then under the electrical limit

\[ \text{COP} \leq 1/\eta_{C} \] (B7)

and

\[ \dot{Q}_{h-\text{max}} = \text{COP} \cdot \dot{E}_{e} \] (B8)

Hence

\[ \dot{E}_{\text{h-1im}} = \frac{\dot{Q}_{h-\text{max}}}{\dot{E}_{\text{cyc}}} = \frac{\text{COP} \cdot \dot{E}_{e}}{\dot{E}_{\text{cyc}}} = \text{COP}_{h} \cdot \eta_{C} \] (B8)
which is less than or equal to unity by Equation (B7).

When COP > $1/\eta_h$, the heat pump is limited with the energy of the loop water, and the maximum heat load now is given by Equation (B4). Hence

$$R_{h-max} = \frac{Q_{h-max}}{E_{cy}} = \frac{(1 - \eta_C) \text{ COP}}{(\text{COP} - 1)}$$  \hspace{1cm} (B10)

For the actual heat pump, COP is given as Equation (2-25). In the ideal operation COP = $T_h/(T_h - T_c)$ which yields

$$R_{h-lim} = T_h(1 - \eta_C)/T_c.$$  \hspace{1cm} (B11)

Heating --- AES: If all the electricity generated is used by resistance heaters, they could provide

$$Q_{h-max} = E_{e-max} = E_F \cdot \eta_C.$$  \hspace{1cm} (B12)

This result, as another manifestation of the ineffectiveness of this system, in the strictest limiting $R_h$,

$$R_{h-lim} = \eta_C.$$  \hspace{1cm} (B13)

Furthermore, this represents the absolute limit for this system since it is not possible to increase it with only minor alterations, as was indicated above for the BOS and the HPS.

Heating --- UHS: Assuming the heat source at $T_0$ is vast so that the limitation on $R_h$ is only from available electricity gives

$$R_{h-lim} = \eta_C \cdot \text{COP}.$$  \hspace{1cm} (B14)
Note that $R_h$ could be greater than unity, which indicates that the energy which the heat pumps extract from the surroundings is greater than the energy lost from the power plant to the surroundings.

**Heating --- UFS:** When all of the fuel allocation is directed to the furnaces, the limiting $R_h$ becomes

$$R_{h-lim} = \eta_f.$$

(B15)

Which, as in the AES, is the absolute limit for this system.

**Cooling --- BOS:** With all steam bled off at $T_{ab}$, the absorption unit receives $\dot{E}_e (\frac{1}{\eta_{ab}} - 1)$, and it could supply a maximum cooling load

$$\dot{Q}_{c-max} = \text{COP}_{ab} \cdot \dot{E}_e \left(\frac{1}{\eta_{ab}} - 1\right).$$

(B16)

Hence the limiting load ratio is

$$R_{c-lim} = \frac{\dot{Q}_{c-max}}{\dot{E}_{cyc}} = (1 - \eta_{ab}) \text{COP}_{ab}.$$

(B17)

For the ideal unit, with COP$_{ab}$ as given in Equation (2-18a), Equation (B17) becomes

$$R_{c-lim} = (1 - \eta_{ab}) \frac{T_c(T_{ab} - T_0)}{T_{ab}(T_0 - T_c)}.$$

(B18)

which is increasing with decreased $T_c$ or increased $T_{ab}$, as would be expected.

For the actual unit, COP$_{ab} = 0.7$ so that Equation (B17) becomes

$$R_{c-lim} = 0.7(1 - \eta_{ab}).$$

(B19)
indicating that $\eta_{ab}$ has to be lowered to increase the cooling capacity. Note also that the limit of the actual unit is always less than 0.7, while that of the ideal unit could exceed unity.

Cooling --- HPS: Since the surroundings provide an essentially limitless sink, the limitation here comes only from available electricity. Hence

$$R_{c-lim} = \frac{-\dot{Q}_{c-max}}{\dot{E}_{cyc}} = \frac{\dot{E}_{cyc} \cdot \eta_C \cdot COP}{\dot{E}_{cyc}} = \eta_C \cdot COP \cdot \frac{1}{R_{c-lim}}. \quad (B20)$$

Actual equipment COP$_c$ is given as Equation (2-26). The ideal unit has COP$_c = T_C / (T_C - T_C)$. Note that $R_{c-lim}$ can be larger than 1.0.

Cooling --- UFS: With all the fuel going to the furnaces coupled to the absorption chillers, the result, as in above, reduces to

$$R_{c-lim} = \eta_f \cdot COP_{ab}. \quad (B21)$$

Table B1 shows the limiting R values for the LWR plant coupled with the BOS and HPS as well as the conventional systems considered here. For both the ideal and the actual HPS, the heating limit is from the loop. Note that the corresponding values of BOS and HPS are of the same order, and all values except for the AES are outside the ranges investigated in Chapter 2.
<table>
<thead>
<tr>
<th></th>
<th>BOS</th>
<th>HPS</th>
<th>CONVENTIONAL</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Ideal</td>
<td>Actual</td>
<td>Ideal</td>
</tr>
<tr>
<td>$R_{h\text{-}lim}^{(1)}$</td>
<td>0.65</td>
<td>0.83</td>
<td>0.69</td>
</tr>
<tr>
<td>$R_{c\text{-}lim}^{(2)}$</td>
<td>3.90</td>
<td>0.58</td>
<td>3.00</td>
</tr>
</tbody>
</table>

(1) Heating: $T_h = 105^\circ F$, $T_C = 80^\circ F$, $T_{BO} = 310^\circ F$.

(2) Cooling: $T_c = 55^\circ F$, $T_C = 110^\circ F$, $T_{ab} = 310^\circ F$. 
B-2. Remaining Thermal Waste

The equations for the electricity remaining after the specified thermal demands are met were obtained in the developments of the system effectiveness equations. Since, as seen, the thermal pollution from large plants will seldom vanish, the equations giving the remainder of this output are now presented for completeness.

During heating with ideal BOS, the energy lost to the cooling system, $\dot{E}_L$, can be calculated as

$$\dot{E}_L = \dot{E}_cyc - \sum_i \dot{Q}_{h,i} / (1 - \eta_{h,i})(1 - \eta_c), \quad (B22)$$

or from

$$\dot{E}_L = \dot{E}_cyc - \dot{E}_e - \sum_i \dot{Q}_{h,i}, \quad (B23)$$

both yielding

$$\dot{E}_L = \dot{E}_cyc (1 - \epsilon') - \sum_i \dot{Q}_{h,i} (T_0 / T_{h,i}). \quad (B24)$$

Defining a new ratio, $L_h$, analogous to $R_h$, as

$$L_h = \dot{E}_L / \dot{E}_cyc \quad (B25)$$

results in

$$L_h = 1 - \epsilon' - \sum_i R_{h,i} (T_0 / T_{h,i}). \quad (B26)$$

Similar equations but with $T_{BO}$ in place of $T_{h,i}$ apply to an actual system. As expected, better system effectiveness, greater waste utilization ($R_h$), and lower $T_{h,i}$ reduce the remaining power plant waste.
During heating, HPS utilizes $\Sigma_i \dot{Q}_{h,i} (1-1/COP_i)$, leaving

$$\dot{E}_L = \dot{E}_{cyc} (1-\eta_C) - \Sigma_i \dot{Q}_{h,i} (1-1/COP_i). \quad (B27)$$

This again reduces to Equation (B26) when the definitions of $L_h$ and ideal COP are introduced. The numerical values of $L_h$ for the two systems will differ though, as the values of $\varepsilon'$ are not the same for the BOS and the HPS serving identical demands.

There is no waste heat utilization with the AES and UHS; therefore, as for a single-purpose plant,

$$L_h = 1 - \eta_C. \quad (B28)$$

With the UFS, the thermal waste at the power plant in

$$\dot{E}_L = (\dot{Q}_t - \Sigma_i \frac{\dot{Q}_{h,i}}{\eta_f}) (1-\eta_C) \quad (B29)$$

Hence

$$L_h = (1 - \Sigma_i \frac{R_{h,i}}{\eta_f}) (1-\eta_C). \quad (B30)$$

During cooling, the BOS is like heating of $\dot{Q}_{ab}$ at $T_{BO}$, and the HPS has no effect on the plant when local cooling towers are used.

For the UFS, Equation (B30) is modified to

$$L_c = [1 - \Sigma_i \frac{R_{h,i}}{\eta_f \cdot COP_{ab}}] (1-\eta_C). \quad (B31)$$

Clearly, the reductions in thermal pollution at the power plant site relative to the fuel energy input are obtained by simply taking the difference between Equation (B28) for an uncoupled plant and Equations (B26), (B30), or (B31) for the systems with a reduction.
APPENDIX C

OPTIMUM LOOP TEMPERATURE

The loop water serves the HPS both as a thermal source and a thermal sink. Because water heating, and space cooling in non-residential buildings are year around tasks, frequently the heating and the cooling demands will be served simultaneously, even if independently. For heating, the warmer the $T_{\text{loop, in}}$ the better the performance; so during a mainly heating season, the loop brings directly the cooling water leaving the power plant condenser. For cooling, though, the colder the $T_{\text{loop, in}}$ the better performance; so during a mainly cooling season, the loop brings the cool water directly from a cooling tower or reservoir. At times when neither load dominates, a loop temperature in between these limits would be more desirable than either. Such a medium-temperature water could be obtained by mixing flows from the two sources. Figure C1 gives the optimum loop inlet temperature as a function of the gross cooling and heating loads ratio. However, because the potential benefit from adjusting the loop temperature is not huge, it may not always justify the expense of predicting the ratio and accordingly controlling the mixing. For this reason, the profile is not incorporated into the analysis of this work.
Fig. C1. The optimum loop water temperature as a function of the ratio of the gross cooling and heating loads served by the HPS.
APPENDIX D

HPS SIMULATION PROGRAM

This appendix contains the FORTRAN program listings, and three lists for the input, output, and other major variable names and their brief explanations. The program consists of seven subprograms the general structure and execution of which were discussed in Section 4-3. The comment statements included in the program listing give information about major steps. The input and the output variables are listed in the order of occurrence; the other variables are listed alphabetically. All input variables, explicit or implicit, are listed except those involving the system design given in Section 4-2, and those relating unit conversions. The output list also gives the symbolic headings that are printed on the output pages.
FORTRAN Simulation Program Listing

PROGRAM SOS

C MAIN PROGRAM FOR HOURLY SIMULATION OF THE HEAT PUMP SYSTEM.

DIMENSION TOA(24), HF(24), TCW(12), TLW(12), AD(35)
COMMON ID, HI, CS, T2, MDP, DLP, THI, TMO, CTH, ETA, CUW, X, SW, ESL

CITY WATER TEMP. FOR JAN.-DEC.
DATA (TCW(K), K=1,12) = 42.0, 41.0, 42.0, 46.0, 46.0, 49.0, 51.0, 55.0, 59.0, 56.0, 50.0, 43.0
CITY-PLANT LOOP SUPPLY TEMP. (JUNE-SEPT. FROM PLANT COOLING TOWER OUTLET, CONDENSER OUTLET FOR THE OTHER 8 MONTHS.)
COOLING TOWER (TROJAN) PROFILE IS USED.
DATA (TLW(K), K=1,12) = 14.9, 106.3, 103.3, 110.8, 113.3, 79.0, 82.0, 12.3, 78.0, 112.3, 108.3, 106.3

SHOT WATER USAGE FRACTIONS FOR EACH HOUR (1 AM.-MIDNIGHT) FOR ALL DAYS.
DATA ((HF(K), K=1,24) = 0.016, 0.008, 0.002, 0.002, 0.002, 0.002, 0.008, 0.037, 0.037, 0.037, 0.037, 0.037, 0.037, 0.037, 0.037, 0.037, 0.037, 0.037, 0.037, 0.037, 0.037, 0.037, 0.037, 0.037

DATA (AD=35(0.))
TL0=TAO=TAW=QA=QW=WA=WW=ESL=0.0
MP=MDP=NO3=0
ETW=ETA=50.
CTW=CTA=126.
TT=130.
TAI=72.

SETTING TANK FILL-UP LIMIT (24 INCH), HT. AND CL. LOADS PER DEGREE HOUR.
CAN SERVE 1/3 OF A DRNL-TYPE APARTMENT BUILDING IN PORTLAND, OR, CLIMATE.
WLL=24.
WLP=WLL/2.
CLDH=15800.0/35.
CPG=62.1*0.13368

CAN FIRST READ BASE TEMP. FOR HT. AND CL. (F), HOT WATER USE (GPD), DEAD AREA HEIGHT IN TANK (IN), LOWEST TANK TEMP. DESIRED (F), COMPRESSOR SIZE AS COMPARED TO ONE GIVEN IN SUBR. HPU, CONSECUTIVE DAY SIGNAL (IC=0 FOR NON-CONS. INPUT), AND IW (=0 SIGNALS THAT CALCULATIONS FOR WATER HEATING MODE TO BE DONE ONLY ONCE A MONTH.)
READ(6,2) BTH, BTC, GPD, SW, TWMICS, IC, IW
IF(EOF(6)) GO TO 91
IF (IC.LE.0) GO TO 71
WLP=WLL/2.
TT=130.

CAN START NON-CONSECUTIVE DAYS WITH TANK HALFFULL AT 130F.

99 READ(I,1) (TOA(K), K=1,24), M, MD
IF(EOF(6)) GO TO 91
IF (IC.LE.0) GO TO 71
WLP=WLL/2.
TT=130.

CAN READ HOURLY OUTSIDE AIR TEMP., MONTH AND DAY.
99 READ(I,1) (TOA(K), K=1,24), M, MD
IF(EOF(6)) GO TO 91
IF (IC.LE.0) GO TO 71
WLP=WLL/2.
TT=130.

CAN START NON-CONSECUTIVE DAYS WITH TANK HALF-FULL AT 130F.

20.7 BTU/F/IN. ASSUMES 4 FT*2 TANK BASE, AND CP=1.
71 TMC=(X.P+30)*20.7
TLI=TL4(M) $ THI=TCW(M)
WRITE(1,8) M,MO,TLI
RNH=MLJ=FH=TNH=TNH=TNH=THW=THW=GE=0.
DO 77 <=1,20
77 AD(K)=0.
C
IF (NP.EQ.M*1) GO TO 78
C
CALCULATING WATER SIZE H AS A FUNCTION OF TLI.
HI=820.*((TLI/100.)**0.54)
IF(IW.4E.2) GO TO 78
C
CALLING FOR WATER HEATING PERFORMANCE FOR THE MONTH.
CALL D+H (TLI,TLOF,QWF,WWF)
TWF=TW+ETWF=ETW $ CTWF=CTW
COPWF=QWF/WWF
OLWF=TLI-TPF
DFW=TF-WF
C
HOURLY SIMULATION FOR THE DAY.
C
78 DO 88 J=1,24
OT=TOA(J)
IF (TT.GE.TWM) GO TO 79
RHD=R1D*(TT-TW)*TMC
TT=TWM
79 G=HF(J) * GDP
C
CHECK FIRST HOUR WATER DEMAND OF THIS HOUR, FILL-UP THE TANK IF NEEDED.
C
IF(IW.4E.0) GO TO 19
COPWF=COPWF $ DLW=DLWF $ DH=DWF $ TWF=TWF
QW=QWF $ WW=WWF $ ETW=ETWF $ CTW=CTWF
CAN USE THESE MONTHLY VALUES.
GO TO 18
C
CALCULATE HOURLY WATER HEATING PERFORMANCE IF DESIRED.
19 CALL D+HT (TLI,TLO,QWF,WW)
COPWF=QWF/WW
DLWF=TLI-TLO
DFW=TWF-TWI
C
18 TT=TWF
TYW=9.45
ID=10
WLP=WL-(G-WLP/4)/GF)*(130.-TW)*TT-TW)*0.4
IF(WLP.GE.0.) GO TO 14
CAN FILL THE TANK TWICE IF NEEDED, EACH TAKING 9.45 MINUTES.
TYW=18.9
WLP=WL*10
QW=QWF*TF
\[
WW = \text{WH} \times TF \\
\text{WH} = \text{WH} / 3413, \\
T_{MC} = (\text{WLP} + \text{SH}) \times 20.7, \\
A_D(1) = A_D(1) + \text{GW} \\
A_D(2) = A_D(2) + \text{KWH} \\
\]

CHECK NEXT THE SPACE LOAD. MODIFIED DEGREE HOUR METHOD IS USED FOR BOTH COOLING AND HEATING (MAY READ DIFFERENT BASE TEMP.)

13 DM = BTH + CT \\
IF (DH) 21, 23, 22

23 WA = KWA = SL = TYA = COPA = DLA = DA = TAO = ETA = CTA = EQW = EDT = 0. CAN SKIP SPACE LOAD FOR THIS HOUR.
GO TO 25

21 DM = BTC - CT \\
IF (DH = 0) GO TO 23
SL = CLD1 \times OH4ESL
ESL = 0.
TCDM = TCOH - DH

CALL FOR SPACE COOLING.
CALL S^3GL (TLI, TAI, TLO, TAO, QA, WA, ETA, CTA, SL, TMC, TT, TYA, EQW, COPA) \\
DA = TAI - TAO \\
DLA = TLI - TLO \\
FW = FW \times EW \\
EDT = TT - TMO
GO TO 24

22 SL = HLDH \times DH \times ESL \\
ESL = 0.
T(HM) = T(H) + DH

CALL FOR SPACE HEATING.
CALL S^3HT (TLI, TAI, TLO, TAO, QA, KA, ETA, CTA, SL, TMC, TT, TYA, EQW, COPA) \\
DA = TAO - TAT \\
DLA = TLI - TLO \\
EDI = EQW / TMC \\
TT = TT + EDT

24 WKA = WA / 3413, \\
TYA = TYA \times 60, \\
AD(3) = AD(3) + WKA \\
AD(4) = AD(4) + TYA \\
AD(5) = AD(5) + EQW \\
AD(6) = AD(6) + EDT \\
25 TYM = TYH + TYA \\
TM = WK + WKA \\
QL = -SL + WA + WQ \times HH = EQW \\
AO(7) = AO(7) + TT \\
AO(8) = AO(8) + OL \\
AO(10) = AO(10) + TYH \\
IF (EQ \leq 0.0) GO TO 88

TYA = -ESL \times TYA / SL \\
SL = ESL

OUTPUT THE HOURLY SIMULATION RESULTS.

88 WRITE(1, 9) J, OT, QW, WKK, COPW, CTH, ETW, TWI, TMO, DW, DIL, HLP \\
* , SL, WKA, COPA, CTA, ETA, TAI, TAO, OA, DLA, TYA \\
* , EQW, EDT, TT, TLK, TYM, T0, ID

CALCULATIONS FOR MONTHLY AVERAGES AND TOTALS.
MP = M + 15 MUP = MD + 1
FF = FW / (AD(1) + FW)
AD(9) = AD(2) + AD(3)
DG 76 < 1, 10
KP <= 10
76 AD(KP) = AD(KP) / 24.

RESISTIVE HEATING OF 30% WATER TO 160 F FOR WASHERS.
RHH = (150 - AD(17)) * GPH * CPG
ARWH = RHH + RHLD / 24.
ARWH = RHH / 3413.
GE = ARWH + AD(19)
AD(6) = AD(7) = 0.

C

OUTPUT THE AVERAGES FOR THE DAY.
WRITE(1,5)
WRITE(1,7) (AD(K), K=11,29)

OUTPUT THE TOTALS FOR THE DAY.
WRITE(1,6)
WRITE(1,7) (AD(K), K=11,10)
WRITE(1,4) THCH, TCDH, FF, RHH, THM, RHLD, ARWH, ARWH, GE

CAN GO TO INPUT THE NEXT DAY IF NON-CONS. RUN.
IF (IC.EQ.0) GO TO 99
NO = NO + 1
AD(21) = AD(21) + THDH
AD(22) = AD(22) + TCDH
AD(23) = AD(23) + FF
AD(24) = AD(24) + AD(20)
AD(25) = AD(25) + AD(17)
AD(26) = AD(26) + AD(19)
AD(27) = AD(27) + AD(18)
AD(29) = AD(29) + ARWH

CAN GO TO INPUT THE NEXT DAY.
GO TO 39

C

OUTPUT SUMMARY, IF ANY, AND END.
91 IF (IC.EQ.0) CALL EXIT
92 DO 92 K = 23, 29
93 AD(K) = AD(K) / NO
94 AD(27) = AD(27) / 3413.
95 AD(30) = AD(29) / 3413.
96 AD(31) = AD(27) + AD(30)
97 WRITE (1,10) NO, (AD(K), K=21,31)
98 CALL EXIT

C

1 FORMAT (24F3.0,4X,2I2)
2 FORMAT (5F4.0,F5.2,2I2)
4 FORMAT (I3,F4.0,12(F7.0,2F5.2,F4.0,3E3.0,12X)),
8 FORMAT (#, # MONTH = # , DAY = # , LOC. T (F) = # ,
12 # HR TOA HH-HT WORK COP CON EV T/I T/O DT LP HP#
16 # SP-HT WORK COP CON EV T/I T/O DT LP OP#
20 # TH=PCL INK OP T2 10# 5# F#X
24 # BTU KWH#7X FF F F F F F M BTU F F5X
28 # BTU (WH M F#/
32 FORMAT (I3,F4.0,2(F7.0,2F5.2,F4.0,F3.0,2F4.0,3F3.0,2X),
10 FORMAT ( *1SUMMARY FOR I4# CONSECUTIVE DAYS ...*//
  # TOTAL HEATING DEGREEHOURS =F7.3# TOTAL COOLING DEGREEHOURS =
  # F7.6# AVR. FREE H. MT. FRACTION =F7.3/
  # AVR. OPERATION TIME (M/H) =F7.1# AVR. HEATING DEGREEHOURS =
  # F7.1# AVR. HEAT PUMP WORK (KW) =F7.3/
  # AVR. TANK TEMPERATURE (F) =F7.3# AVR. TANK TEMPERATURE (F) =
  # AVRG. OPERATION TIME (H/H) =F7.1# AVR. TANK TEMPERATURE (F) =
  # AVRG. HEAT PUMP WORK (KW) =F7.3/
  # AVRG. THERMAL POLLUTION =F7.3# BTU/HR. =F7.3# BTU/HR. =F7.3# BTU/HR. =
  # AVRG. GROSS ELECTR. USAGE =F7.3# KW#)

END

SUBROUTINE DHHT (TLI,TLO,Q,W)

DOMESTIC WATER HEATING. #DHHT# PREPARES DESIGN PARAMETERS FOR
THE HEAT PUMP UNIT PERFORMANCE ESTIMATION.

COMMON IDH,HI,CSI2,MPAI,THI,TWO,ET,CUW,X
DATA FOR HE LOOP UNIT AREA (FT*2), FLOW RATE (L3/H), CP, M*CP,
AND WATER UNIT AREA.
DATA (AL=11500.),(FL=5250.),(EPL=1.),(CPF=5250.),(wA=16900.),(WCP=1.)
UL=1./(.0368+1.9/HI)/117.
UL=UL*AL
TUL=1.-EXP(-UAL/CPF)
CUL=CPF*TUL
CITY WATER FLOW RATE (L3/H) TO HEATER IS SET NEXT.
WCPM=WCP*WCP
WU=1.
WUA=WU*MA
WU=1.-EXP(-WUA/WCPM)
CUW=CUW*M*WU

CALLING THE HEAT PUMP UNIT.
CALL HPU(THI,TLI,CUW,CUL,R,ET,CT,0.,0.,0.,0.,0.,0.,0.)

END
SUBROUTINE SPHT (TLI, TAI, TLO, TAO, Q, W, ET, CT, SL, TMC, TT, TYA, EQW, COPA)

SPACE HEATING. * SPHT * PREPARES DESIGN PARAMETERS FOR
THE HEAT PUMP UNIT PERFORMANCE ESTIMATION.

COMMON ID, HI

DATA FOR THE AREA AND CP OF LOOP AND AIR UNITS, AND FLOW RATE OF LOOP WATER.
DATA (AL=11500.), (AA=12500.), (CPL=1.), (CPA=.24),
(FL=5250.), (CPF=5250.)

ID=ID+1
UL=1./(0.0068+1.9/HI)/110.
UAL=UL*AL
TUL=1.-EXP(-UAL/CPF)
CUL=CPF*TUL

CAN SET AIR FLOW RATE (LB/H) HERE.
FA=2500C.
CPM=CPA*FA
UA=1.
UAA=UA*AA
TUA=1.-EXP(-UAA/CPM)
CU=TUA*CPM

CALLING THE HEAT PUMP UNIT.
CALL HPU (TAI, TLI, CU, CUL, R, W, Q, ET, CT, SL, TMC, TT, TYA, EQW, COPA)

W=W*TYA
TAO=TAI+U/CPM
TLO=TLI-R/CPF
RETURN
END
SUBROUTINE SPCL (TLI, TAI, TLO, TAO, Q, W, ET, CT, SL, TMC, TT, TYA, EQW, COPA)

SPACE COOLING. SPCL PREPARES DESIGN PARAMETERS FOR THE HEAT PUMP UNIT PERFORMANCE ESTIMATION.

COMMON ID, HI, CS, T2, MLPL, GLH, TWI, HTA, CT1, ET1, CUW, X, SW, ESL

D DATA FOR AREA, Cp, of loop and air units, and tank fill-up height.
DATA (AL=11500.), (AA=12530.), (CPL=1.), (CPA=.24), (WLL=24.).

ID=ID+3
CAN SET LOOP FLOW RATE (LB/H) HERE.
FL=5250.
HIA=HI
CAN INCREASE LOOP FLOW RATE BY 1.5 WHEN ITS T IS HIGH.
IF(TLI.LT.100.) GO TO 9
HIA=HIA*1.382
FL=7875.
9 UL=1./(.0C49+1.9/HIA)/110.
CPF=C0.*FL
UAL=UL*AL
TUL=1.-EXP(-UAL/CPF)
CUF=CPF*TUL
CAN SET AIR FLOW RATE (LB/H) AND UA RATIO NEXT.
FA=25030.
UA=0.77
CAN INCREASE AIR FLOW RATE BY 1.5 AT HIGH A/C LOAD.
IF(SL.GT.-150060.) GO TO 8
FA=37530.
UA=0.836
8 CPM=C0*FA
UAA=UPAA
TUA=1.EXP(UAA/CPM)
CU=UAA*CPM
COPA=-34.
CALL H'P (TAI, TLI, CU, CUL, R, MLPL, SL, TMC, TT, TYA, EQW, COPA)

CALLING HEAT PUMP UNIT FOR TANK FILL-UP WHILE SPACE COOLING.
COPA IS SET NEGATIVE AS A SIGN OF NO LOOP FLOW DURING FILLING.
CAN FILL TANK AT THE RATE OF 150.3 INCH/HR.

IF(ACT1.GT.TYA) ACT1=TYA
WLI=ACT1*150.3
WLPA=WL+WL
TMC=4*(WLPA+5K)*20.7
EQW1=CQW*(CT1-THI)*ACT1
WTO=THI+EQW1/WLI/20.7
WTA=(WTO+WL+TT*HLP)/WLPA
TPCM=(TMC+TMCA)/2.
THM=(TT+HWA)/2.
X=X*TM/THCM
EQW2=TI2CM*(1.-EXP(-X*ACT1))*T2-THM
TT=WTA+EQW2/TMCA
TLO=TLI
TAO=TAI-FA/CPM
W=ML*ACT1
EQM=EQM1+EQM2
TMC=TMCA
WLPL=WLPA

193
CALLING FOR ONLY NAT. CONV. IN THE TANK DURING THE REMAINING COOLING.

CALL H3J (T1T, T1L, CU, CUL, P, W3, G, ETS, CTS, SL3, TMC, TT, TYA, EQW4, COPA3)

SL3 = SL + SL3
ET = ET1
CT = CT1
ID = ID + 30
IF (SL3.GE.0.) RETURN

ACT3 = ET3 = CT3 = TA03 = 0.

CALLING FOR ONLY NAT. CONV. IN THE TANK DURING THE REMAINING COOLING.

CALL H3J (T1T, T1L, CU, CUL, P, W3, G, ETS, CTS, SL3, TMC, TT, TYA, EQW4, COPA3)

ACT3 = ACT3 + TYA
WE = WE + W3 + TYA
EGW = EGW + EQW4
TT = TT + EQW4/TMC
ET3 = ET3 + ETS*TYA
CT3 = CT3 + CTS*TYA
TA03 = TA03 + TAO3*TYA
SL3 = SL3 + R*TYA
IF ((ACT1 + ACT3).LT.95) GO TO 1
ESL = SL3 - SL
SL = SL = SL
ID = ID + 20
GO TO 2
1 IF (SL3.LT.0.) GO TO 7
2 ET3 = ET3/ACT3
CT3 = CT3/ACT3
TA03 = TA03/ACT3
QL = QUL * (CT3 - TLI)
TL0 = TLI + QL/CPF
ID = ID + 20

TYA = ACT1 + ACT3
ACT1 = ACT1/TYA
ACT3 = ACT3/TYA
CT = CT*ACT1 + CT3*ACT3
ET = ET*ACT1 + ET2*ACT3
TA0 = TA0 + ACT1 + TA03 + ACT3
COPA = -SL/W
RETURN
END
SUBROUTINE HPU (TI, TLI, CU, CUL, R, W, Q, C, CT, SL, TM, TT, TYA, EQW, COPA)

CALCULATES HEAT PUMP UNIT PERFORMANCE FOR THE CALLING MODE.
*7 IS AN OPTIMUM RATIO FOR FORMING THE NEXT GUESS.

COMMON HI, CS, T2, WLP, DLW, TWI, TWO, CTW, ETW, CUM, X
DATA (Z=.7), (ZZ=.3)

MODIFICATION FACTORS TO BRING STOECKER EQUATIONS TO ASHRAE SIZE AND COP.
SR=CS/3.1
SW=CS/2.3
I=0

CAN USE THESE (PREVIOUS SOLUTIONS) FOR THE FIRST TRY.
IF(CT,.T.190.) CT=120.
IF(ET,.T.130.) ET=45.
C=CT
E=ET

ITERATIVE SOLUTION OF THE HEAT PUMP, CONDENSER AND EVAPORATOR TEMP.
1 I=I+1
IF(I,.GT.20) CALL EXIT
C=CT*Z+C*ZZ
E=ET*Z+E*ZZ
CC=C*C
EE=E*E

W=2550*E-375*EE+712.5*C-3.125*CC+10.375*E*C
C=1.1375*EE*C+0.04375*E*CC+0.000625*EE*CC
W=W*SW
R=14000.0+21150.*E-65.*EE+2275.*C-13.75*CC-231.*E*C
R=R*SR
Q=W+R

CAN FIND ENTHALPY AND TEMP. OF R22 CYCLE.
1 IF SUBCOOLING AND 20F SUPERHEATING. AVERAGED R22 VAPOR CP=.2
5 TANK OF SUPERHEATER SECTION HAS UA=170.
H4=H(C-10.,0.)
T1=E+20.
RMC=R/(H1-H4)*.2
T2=W/RMC+T1
X=RMC*(1.-EXP(-X*TYA))/(T2-TT)

IF(SL) 8,7,10

SPACE HEATING. WITH A 2% LOCALER RUN
CAN ACCOUNT FOR NAT. CONV. TO WATER TANK.
10 TYA=SL/2*1.02
EQW=TM*C*(1.-EXP(-X*TYA))*T2-TT
Q=Q+Q*TYA
Q=Q
GO TO 6

WATER HEATING (93.5% IS IN FORCED CONV. SECTION.)
7 QU=Q*.335

6 CT=TI+Q/Q
ET=TLI-2/CUL
GO TO 9
COOLING MONS.
8 TYA=SL/R
IF(COPA.GE.3.) GO TO 11

COOLING WITH TANK FILLING AND NC LOOP FLOW.
CT=TH+Q/CW
GO TO 13

COOLING WITH NAT. CONV. IN TANK AND LOOP IS THE MAIN SINK.
11 IF(TYA.GT.C.1) TYA=0.1
   CT=TM+Q-EW/TYA/CUL
   GO TO 13

13 ET=TI-R/CU

ITERATE UNTIL CORRECT COND. AND EVAP. TEMP. - ALL CASES -
9 IF(ABS(TC-C).GT.0.5 .OR. ABS(ET-E).GT.0.5) GO TO 1

COP CALCULATIONS.
1 IF(ISL=3,4,2
2 COPA=CfW
   RETURN
3 COPA=P/W
4 RETURN
END

FUNCTION P(T)
CAN FIND PRESSURE OF SATURATED R22 WHEN ITS TEMP. IS GIVEN.

P=8.3058*10.52971*(0.00472+2.0226E-05*T)*T-30.0394*EXP(T/100.)
RETURN
END

FUNCTION H(T,AP)
CAN FIND ENTHALPY (BTU/JP) OF R22.

DATA ((( A(I,J),J=1,4),I=1,3)=0.563246E02,-0.3046307,0.2340083E-03
+0.1905331E-08,0.7154574E-04,-0.4870323E-06,0.1911951E-11)

CALCULATES H(E) AT T)SAT. (F) IF AP IS GIVEN AS 0 OR - .
1 H=28.974+(-0.15812-(8.7924E-04+1.4444E-05*T)*T)*T+39.523*EXP(T/100
+0.)
RETURN
CALCULATES SJHIT. R22 H WHEN T (RANKINE) AND AP (PSIA) IS GIVEN.
2 DO 3 T=1,3
3 DO 1 *(A(I,2)+(A(I,3)+A(I,4)*AP)*AP)*AP
   TR=T+40.*
   H=D(1)+(D(2)+D(3)*TR)*TR
RETURN
END
Input Variables

I. DATA Statements

TCW (K), K=1,12  Monthly (average) city-water temperatures. January-December.
TLW (K), K=1,12  Monthly (average) arriving loop-water temperatures.
HF (K), K=1,24  Fraction of daily hot water use for each hour.

II. Within Program

WLL (=24 inches)  Height of usable hot water space in the main tank.
TT (=130°F)  Water temperature in the main tank (initial value).
TAI (=72°F)  Inside (return) air temperature.

III. Read from file

BTH (=65°F)  Base temperature for heating degree hours.
BTC (=65°F)  Base temperature for cooling degree hours.
GPD (=840 GPD)  Daily hot water use in gallons.
SW (=4 inches)  Dead storage height in the main tank.
TWM (=120°F)  Minimum temperature specification for the main tank.
CS (D.less)  Size of the compressor compared to the ASHRAE compressor.
IC (D.less)  Switch variable that is set to zero for non-consecutive input.
IW (D.less)  Switch for hot water performance evaluation.
TOA(K), K=1,24  Hourly outside temperatures (°F) of a day.
M  Month number (1=January to 12=December).
MD  Day of the month (1 to 28-31).

IV. Constants used within program

0.3 (Dimensionless) Fraction of daily water used that is heated to 160°F for washers.
20.7 Btu/°F-inch  Heat capacity of tank water for a 4 ft²-base.
0.4 inch/gallon  Filling (emptying) conversion for a 4 ft²-base tank.
9.45 minutes  Empty tank fill-up duration (design parameter).
160°F  Temperature of the water supplied to the washers.
150.3 in/hr  Tank filling rate.
0.7 (Dimensionless) Fraction used to form the next guess in iteration.
3.1 and 2.3 (D-less) Adjustment factors for compressor C and W.
0.5°F  Accuracy range for the T_{con} and T_{ev} iterations.
10°F  Subcooling in R-22 cycle.
20°F  Superheating in R-22 cycle.
1.0 Btu/lb°F  C_p of water for all temperatures.
0.2 Btu/lb°F  C_p of superheated R-22 vapor.
62.1 lb/ft³  Water density.
2%  Compressor operation-time increase during space heating to balance convection to the tank.
0.1 hr.  Time steps for updating the tank temperature during space cooling with full tank.
### Output Variables

<table>
<thead>
<tr>
<th>Program Variable</th>
<th>Print Variable</th>
<th>Heading</th>
</tr>
</thead>
<tbody>
<tr>
<td>M, MD, TLI</td>
<td></td>
<td>Month, Day, Loop temperature.</td>
</tr>
<tr>
<td>J, HR</td>
<td></td>
<td>Hour (1-24).</td>
</tr>
<tr>
<td>OT, TOA</td>
<td></td>
<td>Outside air temperature (°F).</td>
</tr>
<tr>
<td>QW, HW-HT</td>
<td></td>
<td>Water heating load (Btu).</td>
</tr>
<tr>
<td>WKW, WORK</td>
<td></td>
<td>Water heating work (KWH).</td>
</tr>
<tr>
<td>COPW, COP</td>
<td></td>
<td>Water heating COP.</td>
</tr>
<tr>
<td>CTW, CON</td>
<td></td>
<td>Water heating $T_{con}$ (°F).</td>
</tr>
<tr>
<td>ETW, EV</td>
<td></td>
<td>Water heating $T_{ev}$ (°F).</td>
</tr>
<tr>
<td>TWI, T/I</td>
<td></td>
<td>City water inlet temperature (°F).</td>
</tr>
<tr>
<td>TWO, T/0</td>
<td></td>
<td>Temperature to which city water is heated during tank filling. (Increased later by natural convection.)</td>
</tr>
<tr>
<td>DW, DT</td>
<td></td>
<td>Increase in hot water temperature during filling (°F).</td>
</tr>
<tr>
<td>DLW, LP</td>
<td></td>
<td>Decrease in loop water temperature (°F).</td>
</tr>
<tr>
<td>WLP, HP</td>
<td></td>
<td>Height of usable water at the end of the hour (inch.).</td>
</tr>
<tr>
<td>SL, SP-HT</td>
<td></td>
<td>Space H/C load (Btu).</td>
</tr>
<tr>
<td>WKA, WORK</td>
<td></td>
<td>Space H/C work (KWH).</td>
</tr>
<tr>
<td>COPA, COP</td>
<td></td>
<td>Space H/C COP</td>
</tr>
<tr>
<td>CTA, CON</td>
<td></td>
<td>Space H/C $T_{con}$ (°F).</td>
</tr>
<tr>
<td>ETA, EV</td>
<td></td>
<td>Space H/C $T_{ev}$ (°F).</td>
</tr>
<tr>
<td>TAI, T/I</td>
<td></td>
<td>Return air temperature (°F).</td>
</tr>
<tr>
<td>TAO, T/0</td>
<td></td>
<td>Supply air temperature (°F).</td>
</tr>
<tr>
<td>DA, DT</td>
<td></td>
<td>Change in air temperature (°F).</td>
</tr>
<tr>
<td>DLA, LP</td>
<td></td>
<td>Change in loop water temperature (°F).</td>
</tr>
<tr>
<td>TYA, OP</td>
<td></td>
<td>Compressor running duration (minutes).</td>
</tr>
<tr>
<td>EQW, +HWHT</td>
<td></td>
<td>Energy input to the tank by natural convection (Btu).</td>
</tr>
<tr>
<td>EDT, +T</td>
<td></td>
<td>Temperature increase due to natural convection input during space heating (°F) (Set to zero for cooling).</td>
</tr>
<tr>
<td>TT, WTT</td>
<td></td>
<td>Tank temperature at the end of the hour (°F).</td>
</tr>
<tr>
<td>QL, TH-POL</td>
<td></td>
<td>Energy added (plus) to or received (negative) from the loop water (Btu).</td>
</tr>
<tr>
<td>TWK, TWK</td>
<td></td>
<td>Total work demand (KWH).</td>
</tr>
<tr>
<td>TYM, OP</td>
<td></td>
<td>Total running time (minute).</td>
</tr>
<tr>
<td>T2, T2</td>
<td></td>
<td>The last value of the R-22 temperature as it leaves the compressor (°F).</td>
</tr>
<tr>
<td>ID, ID</td>
<td></td>
<td>A two-digit number indicating operation type.</td>
</tr>
</tbody>
</table>
AD(K),K=11,20
Hourly average values for certain variables.
(QW, WKW, WKA, TYA, EQW, EDT, TT, QL, TWK, TYM)

AD(K),K=1,10
Daily totals for the items listed above (except for EDT and TT which are printed as zero).

THDH, TCHD
Daily totals of heating and cooling degree-hours.

FF
Free water-heating fraction. Defined as (Water heating during cooling)/(Total water heating).

RHH
Energy for resistance heating to 160°F (Btu).

TWM
The minimum allowed tank temperature (°F).

RHLI
Energy for resistance heating to prevent temperatures below the minimum (Btu).

ARWH, ARWHW
Average of net resistance heating in Btu/hr and KW.

GE
Average gross electrical use (KW).

NOD
Number of consecutive days for summary.

AD(K),K=21,31
Summary variables for the consecutive run.
(Total heating and cooling degree days, and averages of FF, TYM, TT, TWK, QL (in Btu and in SW), ARWH, ARWHW, GE).
Major program variables in addition to the Input/Output variables.

ACT1;ACT3;ACTS.  Air conditioning timing (filling; nat. conv.; and nat. conv. steps) (hr.).

CLDH  Space cooling load per degree hour (Btu/°F)

COPWF  COPW fixed for the month.

CPG  Heat capacity of water per gallon (Btu/gallon °F).

CT, CT1, C  Condenser temperatures (°F).

CTWF  CTW fixed for the month.

DH  Degree hour for the hour (Δ Base temp. - outside air temp.) (°F).

DLWF;DWF  DLW;DW fixed for the month.

EQW1,EQW2,EQW4  Energy (Btu) added to the water tank by natural convection during space operations.

ESL  Space cooling load left over to next hour (Btu).

ET,ET1,E  Evaporator temperatures (°F).

ETWF  ETW fixed for the month.

FW  "Fee" water heating during space cooling. Cumulative for a day. (Btu)

G;GF  Hourly hor water demand in gallons at 150°F; correction factor for the present tank temperature.

GPDH  Daily hot water demand at higher temp. (160°F) for washers.

H;H1;H4  Refrigerant enthalpy (Btu/lb) (general; inlet of compressor; after throttling valve.

HI  Water-side heat transfer coefficient for the loop unit.

HLDH  Space heating load per degree day.

MDP,MP  MD+1 and M+1 (next day and next month)

P  Pressure (psia)

Q  Heat transfer, total or rate (Btu or Btu/hr)

QA  Heat transfer to air for space heating.

QQ  Heat transfer with forced convection

QWF  QW fixed for the month

R  Capacity (Btu/hr)

SL3  Remaining space cooling load with nat. conv. in water tank.

T;TR;T1  Temperature (°F;°R; refrigerant at the compressor inlet)

TAOS  Temperature of conditioned air in cooling steps.

TF  TYW converted to hours.

TLO; TLOF  Temperature of leaving loop (general, fixed for the month).

TMC;TMCA;TMEM  Thermal capacity of the tank (Btu/°F) (general; after filling; average during filing)

TTM  Average tank temperature during filling with cooling.

TWOF  TWO fixed for the month.

TYW  Time for separate mode tank filling (minutes)
<table>
<thead>
<tr>
<th>Code</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>W;WA;WW;WF</td>
<td>Work (Btu) (general; for space; for water; and for water fixed for the month).</td>
</tr>
<tr>
<td>WL;WL1</td>
<td>Tank level decrease due to load; tank level increase due to filling during space cooling)</td>
</tr>
<tr>
<td>WLL</td>
<td>Tank level limit</td>
</tr>
<tr>
<td>WLRA</td>
<td>Tank level remaining just after the load.</td>
</tr>
<tr>
<td>WLPA</td>
<td>Tank level after the filling during space cooling.</td>
</tr>
<tr>
<td>WTA;WTO</td>
<td>Tank temperature during filling with space cooling (including the natural convection, forced convection only)</td>
</tr>
</tbody>
</table>