Development of a desuperheater rating method for energy code calculations

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DEVELOPMENT OF A DESUPERHEATER RATING METHOD FOR ENERGY CODE CALCULATIONS

by

TIM MERRIGAN
B.A., Rollins College, 1974
B.S., Columbia University, 1975

THESIS

Submitted in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering
College of Engineering
University of Central Florida
Orlando, Florida

Spring Term
1995
ABSTRACT

The purpose of this project was to develop a rating method for desuperheater water heaters -- heat exchangers that remove superheat from the refrigerant in an air conditioner’s compressor discharge line. The project methodology was to first determine the overall heat transfer coefficient-area product (UA) of the heat exchanger for three basic types of desuperheaters. Both heat transfer analysis and the laboratory performance data indicated that the number of transfer units (NTU) for each heat exchanger was relatively independent of the operating conditions for a properly-charged air conditioner with a single-speed compressor and a desuperheater with a single-speed circulating pump. The measured heat exchanger NTU was then used in the Transient System Simulation (TRNSYS) model to account for the transient effect of changing entering water and refrigerant temperatures on daily desuperheater performance. The TRNSYS program also allowed for daily air conditioner/heat pump operation, hot water use, and the energy supplied by a storage tank’s electric resistance elements. Annual desuperheater performance was then simulated using hourly solar radiation and weather data for Jacksonville, Orlando, and Miami. The results of the project provide a regional rating method for desuperheaters that can be used in the residential calculation procedures of the Florida Energy Efficiency Code for Building Construction.
ACKNOWLEDGEMENTS

In addition to the patience of my graduate advisory committee from the Department of Mechanical and Aerospace Engineering at the University of Central Florida (UCF), a number of other people deserve acknowledgement for their assistance throughout this thesis project. Carlos Colon of UCF’s Florida Solar Energy Center (FSEC) is especially thanked for his help in setting up the laboratory data acquisition system and painstakingly collecting the desuperheater test data. FSEC air conditioning technician Doug Matley is also thanked for installing the laboratory desuperheaters and properly charging and checking the refrigerant system. Patrick Robinson, FSEC’s computer-aided drawing expert, deserves thanks too for preparing the desuperheater and laboratory test drawings. Finally, Bertice Cook is particularly acknowledged for her considerable help in preparing this manuscript.

The patience and support of the Association of Refrigerant Desuperheater Manufacturers (ARDM) is also gratefully acknowledged. ARDM’s member manufacturers provided the desuperheater units for testing and ARDM’s former executive director, Ralph Bollinger, was instrumental in helping this project to initially take shape. Ralph is especially thanked for his insightful observation that desuperheaters needed a method of laboratory testing and rating that could be used to determine credits for the Florida Energy Efficiency Code for Building Construction.
Along those same lines, the Energy Code Program staff of the Florida Department of Community Affairs are also thanked for their support of a parallel field performance study of desuperheaters that was conducted by FSEC at approximately the same time as this thesis project. The members of the advisory committee for this field study are acknowledged for any contributions that may have been used in both projects.

Finally, acknowledgement and thanks is also due Thomas I. Wetherington, the Florida Power Corporation engineer who helped pioneer the use of desuperheaters in the United States over 30 years ago. Tom passed away at the end of 1994 and although only a few of his many publications on desuperheaters could be referenced in this thesis project, his contributions to the field were outstanding. It is sincerely hoped that the spirit of technical analysis and professional engineering in which Tom conducted all his work on desuperheaters has been carried over in a small way in this project.
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CHAPTER 1
INTRODUCTION

Desuperheater System Description

Desuperheater water heating systems use a heat exchanger to transfer heat from the refrigerant in an air conditioner or heat pump compressor discharge line to potable water in a hot water storage tank. The hot refrigerant vapor is in a superheated state, i.e., heated above its saturation temperature, as it leaves the compressor. The desuperheater heat exchanger transfers all or part of this superheat to the water and may even begin to condense the refrigerant if the vapor is cooled to its saturation temperature. In space cooling mode, the heat that is removed from the refrigerant would normally have been rejected through the condenser. Consequently, desuperheaters have traditionally been called "heat recovery" units [Healy and Wetherington, 1965]. Furthermore, since the desuperheater assists the condenser in removing heat from the vapor compression cycle, the air conditioner system efficiency actually improves. From performance measurements, D'Valentine & Goldschmidt [1990] estimated that the cooling Energy Efficiency Ratio (EER) increased from 5 to 10 percent, depending on the air conditioner and desuperheater operating conditions.

However, for desuperheaters connected to a heat pump in its heating mode, the removal of superheat reduces the amount of available heat for space conditioning.
Therefore, the heat pump space heating Coefficient of Performance (COP) is decreased from 15 to 20 percent [D’Valentine & Goldschmidt, 1990]. Nevertheless, when the heat pump is operating, the desuperheater heats water at the efficiency of the heat pump, which is typically more than twice the electrical efficiency of an electric resistance water heater.

Desuperheater water heating systems can be categorized as being either active or passive in the way that they circulate water from the hot water storage tank through the desuperheater. As illustrated in Figure 1, active systems use a circulating pump to move the water through the desuperheater heat exchanger in a counterflow direction to the refrigerant flow. Passive systems do not use a pump but rather rely on natural buoyancy or thermosiphon flow through the desuperheater heat exchanger. Therefore, some
passive systems locate the desuperheater heat exchanger underneath or on the side of the storage tank, as shown in Figure 2. A passive insert system uses a refrigerant heat exchanger that is inserted directly into the storage tank and, hence, does not require either a pump or a thermosiphon arrangement to circulate the water (Figure 3).
Active systems, the most common type of desuperheater water heating system, are generally wired so that the circulating pump turns on whenever the space-conditioning unit's compressor is turned on. Most active systems also utilize bimetallic thermostats to better control the water temperature leaving the desuperheater and to prevent the circulating pump from operating before the refrigerant has reached a high enough temperature for heat recovery. A normally-closed "high limit" thermostat on the water piping opens the electrical circuit to the circulating pump when either the entering or leaving water reaches the desired temperature -- typically around 140 °F (60°C) to 150 °F (66°C). A normally-open refrigerant thermostat does not close the electrical circuit to the pump until the refrigerant reaches the temperature of 115 °F to 130 °F (46-54°C). (This "hot gas" thermostat is necessary especially when the desuperheater is installed on a heat pump that will be operating at condensing temperatures of approximately 115 to 140 °F (46-60°C) and relatively low outdoor temperatures.) Some active systems also use a temperature regulating valve along with the water "high limit" thermostat to better control the water temperature leaving the desuperheater. The temperature regulating valve typically does not open until the water temperature at the heat exchanger outlet reaches the set temperature of the valve, effectively "holding back" the water from returning to the storage tank. Both the "hold back" valve and the "hot gas" thermostat also attempt to prevent the refrigerant from condensing within the desuperheater and possibly causing unstable or unbalanced air conditioner or heat pump operation due to a loss of refrigerant flow control through the space-conditioning system expansion device. Finally, a normally-open thermostat may also be used on an active desuperheater's water line to turn on the pump in order to circulate water for freeze protection.
Passive desuperheater systems typically do not use any controls since they do not use a circulating pump. However, the passive insert system does use a low-voltage thermostat to control a solenoid valve that allows refrigerant to bypass the heat exchanger when the water temperature reaches 160 F (71°C). The valve also maintains 20 percent flow through the vertical heat exchanger in order to prevent refrigerant oil entrapment.

**Literature Review**

Over the last 30 years, many researchers have investigated the performance of desuperheater water heaters -- both experimentally and analytically. Healy and Wetherington [1965] published one of the first detailed analyses of the superheat recovery process and described how the water flow rate through an active desuperheater affected not only the heat exchanger performance but also the storage tank stratification. They concluded that having a stable water temperature leaving the desuperheater would be the best compromise between a high flow rate (that increased the heat exchanger performance) and a low flow rate (that increased the leaving water temperature and enhanced tank stratification). Wetherington [1975] further elaborated on adapting a conventional mixing or tempering valve in order to control the temperature of the water leaving the heat exchanger at the desired hot water use temperature. This "hold back" valve design, which also prevents the refrigerant from condensing in the heat exchanger, is still a prominent feature of one of the most popular desuperheater models today.

The economics of desuperheater water heating was also extensively discussed in the early literature on heat recovery. Mason and Bierenbaum [1977] presented an engineering economic analysis showing a simple payback of six years for a desuperheater
installed on a residential air conditioner in Florida. Olszewski [1984] used a more elaborate temperature bin analysis to show that residential desuperheaters were economically competitive (less than a seven year payback) with electric resistance water heaters in 25 of 28 cities in the United States. (Only Minneapolis, with long severe winters, and Portland and Seattle, with extremely low electricity prices, did not show favorable results.) Both these economic analyses assumed that approximately 20\% of the air conditioner or heat pump capacity is available to heat water. This conservative assumption restricts the energy available for water heating to the superheat portion of the vapor compression cycle, meaning that the refrigerant vapor does not condense in the heat recovery unit. As recognized by Wetherington [1964], this assumption is correct only if the heat recovery unit has a limited surface area available for heat transfer or employs a method to control the leaving water temperature and prevent condensation.

Because of the technical and economic potential of desuperheaters, a conference on "Waste Heat Recovery for Energy Conservation -- Residential and Light Commercial Heat Pumps, Air Conditioning, and Refrigeration Systems" was held in 1980 [Hawks et al.(ed.), 1980]. Sessions on economics, equipment, systems, codes and standards, installation, and servicing were conducted. In one of the more informative papers, Tu and Fischler [1980] presented the experimental results of operating two separate desuperheaters that had been retrofitted to laboratory-controlled heat pumps and operated with or without heat recovery. They found that the amount of heat recovered was dependent upon both the outdoor temperature and the design of the particular desuperheater unit that was retrofitted to the heat pump system. Furthermore, they concluded that there were no adverse effects on the heat pump compressor when a
A desuperheater was used, since both desuperheater units decreased the compressor discharge pressure and its compression ratio. Tu and Fischler [1981] also collaborated with Davies to simulate the performance of a desuperheater in four U.S. cities using a typical house, a 3-ton heat pump, and a standard 40 gallon electric water heater. They first used the National Bureau of Standards Load Determination (NBSLD) simulation program [Kusuda, 1976] to determine the hourly heating and cooling loads of the typical house on the standard heat pump. Next they used the performance data from one of the laboratory-tested desuperheaters to extrapolate the fraction of the heat pump capacity that the desuperheater provided to the hot water storage tank at various outdoor temperatures. This water heating factor (WHF) ranged from 33 percent at an outdoor temperature of 107.5°F (42°C) to 3 percent at 62.5°F (17°C) for cooling and from 18 percent at 57.5°F (14.2°C) to 2 percent at 12.5°F (-10.8°C) for heating. The WHF’s, the heat pump capacity at the various outdoor temperatures, and the hourly heat pump loads were then used with an hourly hot water draw profile to estimate the energy savings of the desuperheater operating with the heat pump over a whole year. Tu et al. [1981] also developed an alternative estimation method using temperature bin data for those locations where hourly weather data was not available. However, they still needed the desuperheater capacity as a function of outdoor temperature to compute the heat recovered in each temperature bin.

Gorji [1986] also numerically simulated the performance of a desuperheater in conjunction with a "typical house with a typical usage of hot water and air-conditioning". Using the hourly simulation program TRNSYS -- a Transient System Simulation program -- developed at the University of Wisconsin [Klein et al., 1975-76], he performed a
sensitivity analysis of the following parameters: house insulation value, storage tank volume and heat loss coefficient, hot water use, compressor discharge temperature, desuperheater outlet temperature, and the tank thermostat setting. Unfortunately, he used an 80 gallon water heater as his base system and a second 80 gallon water heater as an alternative preheat system, so his sensitivity analysis did not have a close correlation with the actual residential practice of using one 40 gallon or 52 gallon water heater. He also only ran the simulation for one day using the weather data for June 14, 1982 in Orlando, Florida. Therefore, his objective of determining "optimum system parameters while providing adequate hot water for household consumption" was limited to a typical summer day. Nevertheless, his overall methodology for the simulation was noteworthy in that his models for the house, the air conditioner, and the heat recovery unit were first simplified by engineering evaluation.

In another simulation-related study, D'Valentine and Goldschmidt [1989] modeled a refrigerant desuperheater using conventional heat transfer correlations and the effectiveness - NTU (number of transfer units) method, in order that the model could be incorporated into publicly available heat pump system simulation programs (such as HPSIM from the National Institute of Standards and Technology [Domanski and Didion, 1983]). To calculate the Nusselt number for refrigerant-side heat transfer, D'Valentine and Goldschmidt used an empirically determined adjustment factor to the Dittus-Boelter equation for single-phase convection in circular tubes. However, after comparing their model's prediction with experimental test data for a desuperheater, they found some discrepancies that they attributed to inaccuracies in the calculation of the refrigerant-side heat transfer coefficient. Therefore, they calculated a Nusselt number correction factor
based on the mass flow rate of the refrigerant in order to adjust the desuperheater model to the empirical data.

D'Valentine and Goldschmidt [1990] also reported the representative performance data for a desuperheater installed on a 2.5 ton air-to-air heat pump. As indicated earlier, they showed that a desuperheater increases the cooling Energy Efficiency Ratio (EER) of the heat pump approximately 5 to 10 percent and decreases the heating Coefficient of Performance (COP) by approximately 15 to 20 percent. They also showed that the desuperheater capacity depended on entering water temperature, the water flow rate, and the outdoor temperature. However, the performance data indicated that, at approximately the same water flow rate, the slope of the inverse linear relationship between desuperheater capacity and entering water temperature remained nearly the same for the two outdoor temperatures used in the cooling mode tests -- 82 F (27.8°C) and 95 F (35°C). Although slightly different from the cooling mode, the slopes of the same relationship were also similar for the two outdoor temperatures used in the heating mode tests -- 35 F (1.7°C) and 47 F (8.3°C). Finally, D'Valentine and Goldschmidt found that, for an air conditioner with a thermal expansion valve for refrigerant flow control, full condensation in the desuperheater -- without any corrective design modifications -- leads to a drop in cooling system efficiency.

Holladay [1984] was one of the few researchers to recognize that properly applied desuperheaters (without a leaving water temperature control) usually condense some refrigerant when the inlet water temperature is less than the refrigerant saturation temperature. He therefore termed the device an auxiliary desuperheater - condenser (ADC) and used heat transfer relations for the superheat and condensing regions to
calculate a leaving water temperature for various compressor operating conditions. Holladay also realized that while the temperature of the refrigerant entering the ADC is dependent on the condensing (saturation) temperature of the air conditioner or heat pump, any additional condensing surface in the ADC is able to reduce this temperature. Therefore, he concluded that when condensing occurs, it is necessary to iterate on the condensing temperature and the leaving water temperature in order to calculate the actual heat transfer in the desuperheater.

**Desuperheaters and the Florida Energy Code**

The purpose of an energy code is to provide design requirements to achieve energy efficient buildings. Prior to the issuing of any residential building permit in Florida, certification of Florida energy code compliance must be presented to the local building official. Recognizing that there are many methods and approaches to effectively utilize energy, the Florida Energy Efficiency Code for Building Construction (the Energy Code) provides a number of paths by which a building can comply [Florida Department of Community Affairs (FDCA), 1993]. Residential buildings less than three stories in height are able to comply with Chapter 6 of the Energy Code by either of three methods:

- **Method A** - the Whole Building Performance Method,
- **Method B** - the Component Prescriptive Method,
- **Method C** - Limited Applications Prescriptive Method.

Method A of Chapter 6 -- the most commonly used compliance method -- uses a calculation procedure for water heating that assigns multipliers to electric resistance,
gas-fired, and other fossil fuel water heaters based on their rated efficiencies. The Code also assigns credit multipliers to solar and desuperheater water heaters that are inversely proportional to the percentage of the annual water heating load that they can provide [FDCA, 1986a]. While the credit multipliers for solar water heaters vary with climate location and system efficiency rating, the Code gives all desuperheater water heating systems the same credit multipliers because no standardized efficiency ratings currently exist for desuperheaters.

With the incorporation of the hot water credit multiplier (HWCM) methodology into the 1986 edition of the Energy Code, the credit multipliers for desuperheater water heaters have been 0.62 if the unit is installed on an air conditioner and 0.58 if the unit is installed on a heat pump. The respective water heating load contributions of 38% and 42% have been assumed using data from a 1982-83 field monitoring study performed by the Florida Solar Energy Center (FSEC) for the Florida Public Service Commission (FPSC) [Merrigan, 1983]. Since 1986, the contribution credits for desuperheaters have been repeatedly challenged as being too low by the Association of Refrigerant Desuperheater Manufacturers (ARDM). ARDM asserts that the units being manufactured today are more efficient than the units tested in the field in 1982-83 [FDCA, 1988]. Beginning in 1986 the Energy Code has required that desuperheaters have a minimum "net useful heat exchange effect" of 50 percent as tested according to the Air-Conditioning and Refrigeration Institute (ARI) Standard 470 [1980] with Florida regulatory modifications. Appendix A lists the eight manufacturers found in the 1993 ARDM Directory of Certified Refrigerant Desuperheater Heat Recovery Unit Water Heaters and their rated net useful heat exchange effect. ("Net useful heat exchange
"Useful heat exchanger effect" is defined as the heat \( q \) gained by the water and is \( \text{"the product of the mass flow} \ (m) \ \text{of the water, the specific heat} \ (c_p) \ \text{and the temperature difference} \ (\Delta t) \ \text{of the water.} \) 

\[ q = mc_p \Delta t \] (1)
Table 1 lists the ARI Standard 470 rating conditions as they were published in the first revision to the standard in 1987. (The published table contains a typographical error in the conversion of the 120 F (48.9°C) entering water temperature to degrees Celsius.)

Table 1
ARI 470-87 Standard Rating Conditions

<table>
<thead>
<tr>
<th>Type System</th>
<th>Saturated Temperature of Entering Refrigerant Vapor</th>
<th>Actual Temperature of Entering Refrigerant Vapor</th>
<th>*Temperature of Entering Water</th>
<th>Temperature of Leaving Water</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>F</td>
<td>°C</td>
<td>F</td>
<td>°C</td>
</tr>
<tr>
<td>Air Cooled</td>
<td>125</td>
<td>51.7</td>
<td>220</td>
<td>104.4</td>
</tr>
<tr>
<td>Water Cooled</td>
<td>105</td>
<td>40.6</td>
<td>180</td>
<td>82.2</td>
</tr>
</tbody>
</table>

*Tests shall be run at the two entering water temperatures shown.

However, Standard 470 did not directly specify the size of the refrigeration system with which the desuperheater would be tested. Therefore, desuperheaters could be tested on large systems with a high refrigerant mass flow rate or on small systems with a small amount of available superheat and an increased potential for condensation heat transfer. Standard 470 did state that published ratings may include a nominal refrigerating system capacity that should be based "upon a total heat transfer effect in the desuperheater/water heater of 2000 Btu/hr (588 W) per ton of total system capacity at the 75 F (23.9°C) entering water temperature." (The original 1980 version of the standard specified entering water temperatures of 75 F (23.9°C) and 115 F (46.1°C).) In 1986 the Florida Department of Community Affairs (FDCA) issued regulatory
modifications to ARI Standard 470 in order to express the "net useful heat exchange effect" as the percentage of total system hot gas superheat, after deducting 3.412 Btu/hr per rated watt from the useful heat exchanger effect if a water circulating pump was used [FDCA, 1986b]. Desuperheaters were also required to have a minimum net "useful heat exchange effect" of 50 percent when tested at only one entering water temperature - 80 F (26.7°C). The nominal system capacity was also changed so that it was based "on a refrigerant 22 mass flow rate of 180 pounds per hour (.02268 kg/s) per ton..." Table 2 presents these modified standard rating conditions for both air-cooled and water-cooled units. The underlined entries in the table indicate the Florida modifications that are different from ARI Standard 470-87.

Table 2
Florida Regulatory Modifications to ARI Standard 470-80
Standard Rating Conditions

<table>
<thead>
<tr>
<th>Type System</th>
<th>Saturated Temperature of Entering Refrigerant Vapor</th>
<th>Actual Temperature of Entering Refrigerant Vapor</th>
<th>Temperature of Entering Water</th>
<th>Temperature of Leaving Water</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>F</td>
<td>C</td>
<td>F</td>
<td>C</td>
</tr>
<tr>
<td>Air Cooled</td>
<td>125</td>
<td>51.7</td>
<td>220</td>
<td>104.4</td>
</tr>
<tr>
<td>Water Cooled</td>
<td>105</td>
<td>40.6</td>
<td>180</td>
<td>82.2</td>
</tr>
</tbody>
</table>

For air-cooled units, the refrigerant vapor is superheated 95 F (35°C) above its saturation temperature of 125 F (51.7°C) temperature. Holladay [1984] states that air-cooled condensers are usually sized so that the condensing temperature is about 30 F above that of the surrounding air. Therefore, this condition of available superheat corresponds to
a condensing temperature typical of the ARI Standard 210/240 [1989] outdoor condition of 95 F (35°C) used in the steady-state testing and rating of air-source cooling units. However, the entering water temperature of 80 F is also below the refrigerant saturation temperature. Hence, it is possible that the refrigerant vapor will condense before it exits the heat exchanger of the desuperheater. As seen in Appendix A, six of the desuperheater units listed in the ARDM Directory have a "net superheat recovery" (defined the same as "net useful heat exchange effect") of 100 percent, meaning that all of the available refrigerant superheat was transferred into the water when the unit was tested. Furthermore, all of the listed units with a net superheat recovery less than 100 percent but still greater than 88 percent are rated for a maximum air conditioning capacity of 5 tons (17.6 kW). Proprietary test reports for two of these units indicate that they were tested on 5 ton (17.6 kW) air conditioning systems at two separate independent test laboratories. However, if these units were tested on air conditioners of smaller capacity and less available superheat, it is entirely possible that they too would have had a net superheat recovery of 100 percent and have begun to condense the refrigerant. Hence, even the Florida regulatory modifications to ARI Standard 470 do not remove all the ambiguities of desuperheater testing and rating.

Water Heater Testing And Rating Standards

The Florida Energy Efficiency Code for Building Construction (the Energy Code) presently uses three separate methods to determine the hot water multipliers and credit multipliers for water heating systems. Electric resistance, heat pump, gas-fired, and other fossil fuel water heaters utilize the U.S. Department of Energy (DOE) Test
Procedures for Water Heaters [1990] in order to obtain an efficiency rating termed "Energy Factor". Solar water heating systems require either a American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) collector [1991a] or complete system test [1987] in order to be rated under a Florida Solar Energy Center system approval procedure [1985] that calculates a "Florida Energy Factor". As mentioned in the previous section, desuperheater water heaters must be tested according to the Air-Conditioning and Refrigeration Institute (ARI) Standard 470 with Florida regulatory modifications to determine their net superheat recovery. Table 3 summarizes the various testing and rating methods used by the present Energy Code and also indicates the typical rating’s range and whether it varies with climate.

Table 3
1993 Florida Energy Efficiency Code for Building Construction
Water Heating Credit Characteristics

<table>
<thead>
<tr>
<th>System Type</th>
<th>Test Method</th>
<th>Rating Method</th>
<th>Rating</th>
<th>Rating Range</th>
<th>Climate Sensitive</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas, Oil-fired Water Heater</td>
<td>DOE Test Procedures</td>
<td>DOE Test Procedures</td>
<td>Energy Factor</td>
<td>.50 - .75</td>
<td>No</td>
</tr>
<tr>
<td>Electric Resistance</td>
<td>DOE Test Procedures</td>
<td>DOE Test Procedures</td>
<td>Energy Factor</td>
<td>.81 - .77</td>
<td>No</td>
</tr>
<tr>
<td>Heat Pump Water Heater</td>
<td>DOE Test Procedures</td>
<td>DOE Test Procedures</td>
<td>Energy Factor</td>
<td>1.5 - 2.5</td>
<td>No</td>
</tr>
<tr>
<td>Solar Hot Water Heater</td>
<td>ASHRAE Standards 93&amp;95</td>
<td>FSEC System Approval</td>
<td>Florida Energy Factor</td>
<td>1.4 - 35</td>
<td>Yes</td>
</tr>
<tr>
<td>Desuperheater Water Heater</td>
<td>ARI (Florida)</td>
<td>ARI (Florida)</td>
<td>Net Superheat Recovery</td>
<td>.50 - 1.0</td>
<td>No</td>
</tr>
</tbody>
</table>
The DOE Test Procedures for Water Heaters were first published by the Federal Energy Administration in the October 4, 1977 Federal Register and subsequently amended in the October 19, 1978, September 7, 1979, and October 17, 1990 Federal Registers. Title 10, Part 430 of the Code of Federal Regulations now requires that all electric, gas, oil-fired, and heat pump water heaters be tested according to these procedures in order to demonstrate that they meet the minimum energy efficiency requirements of the National Appliance Energy Conservation Act of 1987. (Oil-fired water heaters are currently granted a test procedure waiver and, hence, test according to a slightly different method.)

The two significant components of the DOE test procedures are the determination of "energy factor" and the determination of the "first hour rating". "Energy factor" is DOE's overall measure of water heater energy efficiency and is defined in terms of energy output compared to energy consumption over a 24 hour simulated use test. This use test includes six hot water draws of 10.71 gallons (40.5 liters) each at one hour intervals and then an 18 hour standby period. The total hot water use equals the reported national average daily hot water use of 64.3 gallons (243.4 liters) [Gilbert et al., 1985]. The hot water temperature is set at 135 F (57.2°C) and the cold water temperature is 58 F (14.4°C) in order to give a temperature differential of 77 F (25°C). An adjustment procedure for other temperatures is also provided. "First hour rating" was developed for use in selecting the proper size water heating system and is defined as the amount of hot water that the water heater can supply in one hour of operation. The Gas Appliance Manufacturers Association (GAMA) biannually publishes a directory of energy factors
and first hour ratings for gas-fired water heaters. In addition, the GAMA directory [1994] includes the ratings for electric, oil-fired, and heat pump water heaters.

ASHRAE Standard 93-1986 (Reaffirmed 1991), "Methods of Testing to Determine the Thermal Performance of Solar Collectors"

ASHRAE Standard 93-1986 was first promulgated in 1977 in order to determine the thermal conversion efficiency of flat-plate and concentrating solar energy collectors. ASHRAE Standard 95-1987 was first published in 1981 to determine the energy performance of a complete solar water heating system. Both standards specify the method of testing but do not specify the conditions to be used for obtaining a standard rating. ASHRAE prefers that the rating conditions be specified by a separate entity such as an industry organization [ASHRAE, 1987].

In Florida, the Florida Solar Energy Center (FSEC) is required by law to develop standards for solar energy equipment sold or manufactured in the state. FSEC has developed a solar water heating system performance rating that uses the ASHRAE Standard 93 or ASHRAE Standard 95 test results in conjunction with monthly average weather data for three Florida locations -- Miami, Orlando, and Appalachicola [Block et al., 1993]. This rating procedure uses the f-Chart solar design method developed at the University of Wisconsin Solar Energy Laboratory [Klein et al., 1976] to calculate the electrical energy used by the storage tank under a hot water load of 64.3 gallons (243.4 liters) per day. The rating is presented as a regional "Florida Energy Factor" by dividing the annual hot water energy delivered by the annual electrical energy consumed.
In the United States, the Solar Rating and Certification Corporation (SRCC) is an independent national organization which certifies and rates the performance of solar energy equipment. SRCC was formed in 1980 by the Solar Energy Industries Association -- the national solar trade association -- and the Interstate Solar Coordination Council, which represents governments and publicly-owned utilities. Under its document OG-300-89, "Operating Guidelines and Minimum Standards for Certifying Solar Water Heating Systems", SRCC determines a thermal performance rating for a solar water heating system based on ASHRAE Standard 93 or Standard 95 test results in conjunction with a computer simulation. The simulation program -- TRNSYS from the University of Wisconsin’s Solar Energy Laboratory [Klein et al., 1975-76] -- uses the performance parameters from the ASHRAE tests along with accepted engineering practices to model the system and calculate a performance rating. This performance rating is now presented as both a "solar energy factor" and one of six daily water heating energy quantities that the system is able to save for a standard day defined by SRCC. With the exception of the solar radiation, outdoor temperature, and hot water draw profiles, the conditions for the standard SRCC day are the same as those used in the DOE Test Procedures for Water Heaters. SRCC publishes an annual directory of OG-300 certified solar water heating systems [1994].

**Combination Appliance Testing and Rating Standards**

The Air-Conditioning and Refrigeration Institute (ARI) and the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) -- with support from the National Institute of Standards and Technology (NIST) -- are presently
developing testing and rating methods for residential space-conditioning appliances that also include water heaters. However, these methods are primarily aimed at integrated appliances that combine air conditioning, space heating, and water heating in one unit. While these packaged appliances are gradually increasing in the residential marketplace, they do not have nearly the Florida market penetration as field-installed desuperheater water heaters.


ASHRAE Standard 124 was approved for publication in June 1991. Although this standard primarily addresses gas-fired water heaters used for both space and water heating, it too specifies a 24 hour simulated use test -- similar to the DOE Test Procedures for Water Heaters -- for the water heating functions of the combination appliance. ASHRAE Standard 118.2, "Method of Testing for Rating Residential Water Heaters," was also approved in 1993. It too specifies a method of test for rating water heaters that is similar to the DOE test procedures.

ASHRAE SPC 137P (Proposed), "Methods of Testing for Efficiency of Space-Conditioning/Water-Heating Appliances that include a Desuperheater Water Heater"

ASHRAE Standards Project Committee (SPC) 137P first met in June 1988 to begin the development of a test procedure to determine the efficiency of space-conditioning appliances that include a desuperheater. A draft standard that covers both factory-installed and field-installed desuperheaters was disseminated for public review in April 1994. This standard was originally patterned after the Carrier Corporation's
modification to the DOE Test Procedures for Central Air Conditioners [DOE, 1988] that was approved for the Carrier Hydrotech 2000 combined appliance [Code of Federal Regulations, 1990]. This modification basically extends the air conditioning test procedures to include water heating modes at the various ARI Standard 210/240 [1989] testing conditions. ASHRAE Standard 137P therefore includes two water heating mode tests at the ARI test conditions. One of the water heating mode tests is similar to the ARI Standard 470 test procedure for active desuperheaters and the other is more suitable to a passive desuperheater. In addition, Standard 137P includes a simulated use test similar to the DOE test procedures for water heaters. However, while the hot water to cold water temperature differential is still set at 77 F (25°C) for the water heating tests, the proposed standard only specifies a 32.2 gallons (121.9 liters) hot water draw -- the same as a proposed U.S. test procedure by the National Institute of Standards and Technology [Dougherty, 1989]. Nevertheless, both proposals do use 64.3 gallons (243.4 liters) a day in modifications to the standard air conditioning bin calculation procedures to determine combined appliance performance factors and annual energy usage.

Related to the ASHRAE 137P committee’s work is the concurrent development of ARI Standard 290P, "Air-Conditioning and Heat Pump Equipment Incorporating Potable Water Heating Devices." The standard applies to factory-assembled appliances or matched assemblies incorporating refrigerant to potable water heat exchangers. As of mid-1994, the 290P Subcommittee had not finished the proposed standard.
CHAPTER 2
METHODOLOGY

Objective

This project had the following overall objective:

Develop a rating method for desuperheater water heaters that can be directly compared to the ratings of electric resistance, gas, oil-fired, heat pump, and solar water heaters. This method should accurately account for the varying amounts of heat transfer between the refrigerant in the space conditioning unit and the water in the storage tank due to changing compressor operating conditions and tank water temperatures.

General Procedure

The methodology that has been selected to meet the objective of this project is to use laboratory test results for the desuperheater in conjunction with a simulation program in order to model water heating system performance. Using standard specifications for the other components in the system (e.g., circulating pump(s), storage tank(s), air conditioner or heat pump), this methodology is similar to the procedures used to rate solar water heating systems in both the U.S. and in Florida. The project methodology is also similar to the procedures employed by Tu, Davies, and Fischler [1980] in which they simulated the performance of the desuperheater using a typical
house, a standard heat pump, and a standard electric water heater. However, the project methodology differs from the approach of Tu et al. in that the laboratory performance of the desuperheater heat exchanger is determined separately from the performance of the heat pump and/or air conditioner on which it is installed. The heat exchanger performance characterization is then combined with the performance characteristics of a standard heat pump/air conditioner in the selected simulation model.

Therefore, the first step in the project methodology is to experimentally determine the heat transfer characteristics of the desuperheater heat exchanger in its typical operating modes. Next these measured characteristics are used to describe the desuperheater performance in a simulation model that includes descriptions of a standard electric water heater, a standard air conditioner/heat pump, and a standard building load. The results of an annual simulation are then used to calculate an "energy factor", which is defined as the hot water energy output divided by the electrical energy consumption of the total hot water system. This dimensionless measure is a straightforward and logical method of comparing the performance of desuperheater water heating systems to both conventional and other alternative water heaters.

Energy Balance

It is unfortunate that desuperheater water heaters, like solar water heating systems, do not lend themselves to the laboratory determination of "energy factor" as simply as do electric, gas, oil-fired, or heat pump water heaters. Both desuperheater and solar water heaters do not respond to hot water draws with a fixed recovery efficiency. Rather desuperheater and solar water heaters provide heated water to a hot water storage
tank over a period of time. Both water heaters also respond to the influences of outdoor
temperature and solar radiation, although in the case of the desuperheater, these
influences are complicated by the effects of the building structure and the operation of
the space conditioning appliance.

Hence, for both desuperheater and solar water heating systems, it is necessary
to determine the long-term energy flows at the hot water storage tank in order to
calculate the actual energy efficiency of the total system. Drawing a control volume
around the storage tank, the energy flows in and out of the control volume (over a period
of time) are described by the following energy balance:

\[ Q = E + T - U - L \]  

where

\( Q \) = heat delivered to the household by the hot water system
\( E \) = electric (or gas) energy supplied by the storage tank elements (or
burner)
\( T \) = thermal energy supplied by the desuperheater or solar system
\( U \) = increase in internal energy in the tank over a period of time
\( L \) = heat loss from the tank to its surroundings.

In addition, active desuperheater and solar systems both require auxiliary electrical
energy \( E_a \) to operate their circulating pumps and/or controls.

Fortunately, the basic definition of energy factor is energy output divided by
energy consumption over a stated period of time:

\[ \text{Energy Factor} = \frac{Q}{E + E_a} \]  

For electric, gas, oil-fired, and heat pump water heaters, the period of time defined by
the DOE Test Procedures for Water Heaters is 24 hours. If these systems are installed
indoors or in a conditioned space, the 24 hour energy factor for a simulated use of 64.3 gallons (243.4 liters) would not be expected to differ significantly from a energy factor calculated for the same daily use over a period of a year. (The results of this comparison are presented in Chapter 5.) For desuperheater and solar water heaters, however, it is necessary to define a seasonal or annual energy factor since the efficiency of these systems does vary with the season of the year. Of course, this is the same rationale that is used in the determination of the seasonal energy efficiency ratio (SEER) and the heating seasonal performance factor (HSPF) for rating the efficiencies of air conditioners and heat pumps [Code of Federal Regulations, 1993].

Furthermore, it is also necessary to specify a climate for which the seasonal or annual rating will be calculated. The Florida Energy Efficiency Code for Building Construction divides Florida into nine climate zones with three zones in each of three regions -- North, Central, and South Florida. Since the desuperheater rating is designed to be used for the Energy Code's calculation procedures, the annual energy factor will be calculated using climate data from a representative city in each of the three regions. Jacksonville, Orlando, and Miami are three representative cities for which long-term average weather data is available.

**Desuperheater Heat Exchanger Analysis**

The desuperheater heat exchanger transfers heat between the refrigerant and the water. The actual rate of heat transferred to the water \( q_w \) can be expressed in a form that is similar to equation (1):
\[ q_w = \dot{m}_w c_{p,w} (T_{w,o} - T_{w,i}) \]  

where

\[ \dot{m}_w = \text{mass flow rate of water} \]
\[ c_{p,w} = \text{specific heat of the water} \]
\[ T_w = \text{temperature of the water (at inlet i and at outlet o)} \]

The actual rate of heat transferred from the refrigerant \( q_r \) can be expressed as

\[ q_r = \dot{m}_r (h_{r,i} - h_{r,o}) \]  

where

\[ \dot{m}_r = \text{mass flow rate of the refrigerant} \]
\[ h_r = \text{enthalpy of the refrigerant (at inlet i and at outlet o)} \]

This refrigerant heat \( q_r \) includes both the superheat and the energy made available if the refrigerant vapor begins to condense. If the refrigerant is prevented from condensing within the desuperheater by a temperature regulating valve or thermostat, then the maximum heat available to be transferred is restricted to the superheat \( q_{SH} \) and can be expressed as

\[ q_{SH} = \dot{m}_r c_{p,r} (T_{r,i} - T_{r,g}) \]  

where

\[ \dot{m}_r = \text{mass flow rate of the refrigerant vapor} \]
\[ c_{p,r} = \text{specific heat of the refrigerant} \]
\[ T_r = \text{temperature of the refrigerant (at inlet i and at the saturated vapor state g)} \]

Conventional heat exchanger analysis states that the heat exchanger effectiveness \( \epsilon \) is defined as the ratio of the actual heat transferred \( q \) to the maximum possible amount of heat transfer \( q_{max} \) [Kays and London, 1984].

\[ \epsilon = \frac{q}{q_{max}} \]
For a desuperheater in which the refrigerant remains in a single phase, the heat exchanger effectiveness is

\[ \epsilon = \frac{m_c p_c (T_{r,o} - T_{w,i})}{C_{min}(T_{r,i} - T_{w,i})} \]  

(8)

where \( C_{min} \) is the minimum heat capacity rate of the two heat exchanger fluids evaluated at the fluid bulk temperature. If the refrigerant does not condense within the heat exchanger and the minimum heat capacity rate is that of the refrigerant, the effectiveness can also be expressed as a function of the refrigerant and water temperatures only.

\[ \epsilon = \frac{m_r c_{pr} (T_{r,i} - T_{r,o})}{C_{min} (T_{r,i} - T_{w,i})} = \frac{(T_{r,i} - T_{r,o})}{(T_{r,i} - T_{w,i})} \]  

(9)

From comparison of equation (8) with both equations (4) and (6), it is apparent that the definition of "net superheat recovery" (NSR) found in the Florida regulatory modifications to ARI Standard 470 (NSR = \( q_w / q_{sh} \), neglecting any deduction for pump energy) is similar to the definition of heat exchanger effectiveness. However, it is also apparent that "the net superheat recovery" equals the heat exchanger effectiveness only when the inlet water temperature \( T_{w,i} \) equals the refrigerant saturation temperature \( T_{r,g} \). When the inlet water temperature is less than the refrigerant saturation temperature, as specifically required by the Standard 470 rating conditions in Tables 1 and 2, then the "net superheat recovery" will always be greater than the more conventionally-accepted heat exchanger effectiveness.

For a single-phase counterflow heat exchanger, Kays and London have shown that the effectiveness \( \epsilon \) is equal to
\[
\varepsilon = 1 - e^{-NTU \left(1 - \frac{C_{\text{min}}}{C_{\text{max}}} \right)} \frac{1 - \left(\frac{C_{\text{min}}}{C_{\text{max}}}\right) e^{-NTU \left(1 - \frac{C_{\text{min}}}{C_{\text{max}}} \right)}}{1 - \left(\frac{C_{\text{min}}}{C_{\text{max}}}\right) e^{-NTU \left(1 - \frac{C_{\text{min}}}{C_{\text{max}}} \right)}}
\] (10)

where \(C_{\text{max}}\) is the higher heat capacity rate of the two fluids and NTU signifies the number of transfer units. NTU is considered a nondimensional indication of the "heat transfer size" of the heat exchanger and is defined as

\[
NTU = \frac{UA}{C_{\text{min}}}
\] (11)

where

\[U = \text{overall heat transfer coefficient of the heat exchanger}\]
\[A = \text{heat transfer area}\]

Equation (10) can also be manipulated to express NTU as a function of the effectiveness \(\varepsilon\) and the \(\frac{C_{\text{min}}}{C_{\text{max}}}\) ratio for a counterflow heat exchanger.

\[
NTU = \frac{1}{(\frac{C_{\text{min}}}{C_{\text{max}}} - 1) \ln \left(\frac{\varepsilon - 1}{\varepsilon (\frac{C_{\text{min}}}{C_{\text{max}}} - 1)}\right)}
\] (12)

For refrigerant R-22 and water, the minimum heat capacity rate \(C_{\text{min}}\) is almost always that of R-22 except when condensation occurs. Then the refrigerant remains at a constant temperature and its specific heat and its heat capacity rate are by definition equal to infinity. Therefore, \(C_{\text{min}}\) becomes the heat capacity rate of water and

\[
\frac{C_{\text{min}}}{C_{\text{max}}} = 0
\] (13)

For condensation, the effectiveness then reduces to the relationship

\[
\varepsilon = 1 - e^{-NTU}
\] (14)

and the number of transfer units can be expressed as

\[
NTU = -\ln (1-\varepsilon)
\] (15)
Heat exchanger analysis also states that the rate of heat transferred by the desuperheater heat exchanger can be expressed by the relationship

\[ q = UA \Delta T_m \]  

(16)

where \( \Delta T_m \) is the mean temperature difference between the refrigerant and the water. For a counterflow heat exchanger or a heat exchanger in which one fluid temperature is substantially constant, the mean temperature difference can be expressed through the use of the logarithmic mean temperature difference (LMTD).

\[
LMTD = \frac{(T_{r,i} - T_{w,o}) - (T_{r,o} - T_{w,i})}{\ln\left(\frac{T_{r,i} - T_{w,o}}{T_{r,o} - T_{w,i}}\right)}
\]  

(17)

The log mean temperature difference for a pure condensing process (LMTD\(_c\)) uses the saturation temperature \( T_{r,g} \) for both the refrigerant inlet and outlet temperatures, since the refrigerant temperature remains the same.

\[
LMTD_c = \frac{(T_{r,g} - T_{w,o}) - (T_{r,g} - T_{w,i})}{\ln\left(\frac{T_{r,g} - T_{w,o}}{T_{r,g} - T_{w,i}}\right)}
\]  

(18)

For a process in which both desuperheating and condensing occur, however, Bell [1972] points out that the simple use of the actual entering and leaving refrigerant temperatures to calculate the LMTD is invalid for two reasons:

1. the specific heats of the refrigerant vapor and liquid phases are different rather than constant as assumed in the derivation of LMTD.
2. the overall heat transfer coefficients for desuperheating ($U_d$) and condensing ($U_c$) are different rather than constant as assumed in the LMTD derivation. Therefore, it is necessary to separate the analysis of the desuperheating process from a combined desuperheating and condensing process in order to use the LMTD to characterize heat exchanger performance. Both Bell and Holladay [1984] recognized that the temperature of the water at the superheat-condensation boundary needs to be determined in order to calculate the separate LMTD’s for desuperheating and condensing. Both authors also recognized that in a counterflow desuperheater-condenser, the refrigerant vapor transfers its superheat before it begins to condense, while the flow of water will accept the heat of condensation first and then the superheat before it exits the heat exchanger.

For a combined desuperheating and condensing process, equations (7) and (8) can be manipulated to determine the water temperature entering the desuperheating portion of the heat exchanger. Since the maximum possible amount of heat transfer $q_{MAX}$ in this portion is limited to the superheat $q_{SH}$, its entering water temperature is the "boundary temperature" $T_{w,b}$ such that

$$T_{w,b} = T_{w,o} - \frac{\epsilon q_{SH}}{m_w c_{p,w}} \tag{19}$$

The boundary temperature can then be used in place of the leaving water temperature $T_{w,o}$ in equation (18) to calculate the condensing LMTD$_c$.

Once the LMTD’s have been determined for the desuperheating and possibly condensing processes, the overall heat transfer coefficient-area product $UA$ for each heat exchange can be determined from equation (16). By definition, the overall heat transfer
The coefficient-area product $UA$ for the desuperheater heat exchanger is dependent on the refrigerant and water convection coefficients $h_r$ and $h_w$, the refrigerant and water heat transfer areas $A_r$ and $A_w$, the thermal conductivity $k$ of the heat exchanger wall(s) of thickness $t_x$ and cross-sectional area $A_x$, and the surface fouling factors $r_{fr}$ and $r_{fw}$.

$$\frac{1}{UA} = \frac{1}{h_rA_r} + \frac{r_{fr}}{A_r} + \frac{t_x}{kA_x} + \frac{r_{fw}}{A_w} + \frac{1}{h_wA_w} \quad (20)$$

The convection heat transfer coefficient $h$ for fluids flowing in heated tubes is related to the Nusselt number $Nu$ by the definition

$$Nu = \frac{hd}{k_f} \quad (21)$$

where

- $d$ = the hydraulic diameter of the tube
- $k_f$ = thermal conductivity of the fluid

For single-phase turbulent flow in circular tubes, the Dittus-Boelter equation empirically correlates the Reynolds number $Re$ and the Prandtl number $Pr$ to the Nusselt number.

$$Nu = 0.023 \, Re^0.6 \, Pr^n \quad (22)$$

where $n = 0.3$ for the cooling of the refrigerant by the water and $n = 0.4$ for the heating of the water by the refrigerant.

The Reynolds number is defined as the ratio of the inertial forces to the viscous forces in a fluid and the Prandtl number is defined as the ratio of the momentum diffusivity to the thermal diffusivity of a fluid. Therefore, these ratios will remain relatively constant when the flow rates, viscosity, specific heat, and thermal conductivity for the water and the refrigerant do not change substantially. If the Reynolds number and the Prandtl number do not change, then the Nusselt number and the fluid convection
coefficient in equation (21) will also remain constant as long as the geometry of the heat exchanger remains fixed. Therefore, equation (20) implies that the \( UA \) for the desuperheater will be relatively constant when the Reynolds number and the Prandtl number for the refrigerant and water do not change substantially.

For a single-speed desuperheater circulating pump, the mass flow rate of the water is constant. (If the desuperheater has a leaving water temperature control valve, then the outlet temperature of the water is held constant and the water flow rate will change. Fortunately, it was found from the laboratory testing discussed in Chapter 3 that the flow rate remains relatively constant over the specific settings of the valve.) The specific heat of water is also relatively constant over the desuperheater inlet and outlet temperature range of 68 to 185 F (20-85°C). Since the specific heat \( c_p = c_p(P,T) \) for a compressible fluid, \( c_p \) for the refrigerant vapor entering the desuperheater is a function of the compressor discharge temperature \( T \) and discharge pressure \( P \). Figure 4 depicts the increase of specific heat for R-22 as compressor discharge pressure increases over the range of operating temperatures. Fortunately, as indicated in equation (22), the Nusselt number for the refrigerant only depends on the Prandtl number raised to the 0.3 power. \( (Pr = \text{function} (c_p)) \). Therefore, a 10 percent change in the refrigerant specific heat from 0.21 to 0.23 Btu/lbm-F (0.88-0.96 kJ/kg-°C) results in a less than 3 percent change in the Nusselt number. Therefore, as long as the vapor does not condense, the Nusselt number of the refrigerant can be considered to remain constant. The refrigerant specific heat can also be determined at a bulk temperature and pressure for a given compressor operating condition.
The compressor operating conditions also determine the refrigerant mass flow rate. However, Farzad and O'Neal [1992] have shown that the refrigerant flow rate for a properly charged air conditioning system does not change more than 10% when the outdoor temperature varies from 27°C (82°F) to 37.8°C (100°F). For a system using a constant area expansion device such as a capillary tube for refrigerant flow control, they determined that the flow rate increases slightly as the condensing temperature increases. Conversely, for a system using a thermal expansion valve, the refrigerant flow rate decreases slightly as the outdoor temperature increases.
Therefore, for a properly charged air conditioner with a single-speed compressor and an active desuperheater with a single-speed circulating pump, the Reynolds, Prandtl, and Nusselt numbers and, hence, the overall heat transfer coefficient-area product \((UA)_d\) for a desuperheating only process would not be expected to change significantly as outdoor conditions change. Furthermore, since the volumetric heat capacity rates for both the refrigerant and water do not change substantially, the number of transfer units \((NTU = UA/C_{min})\) indicating the heat transfer size of the heat exchanger should also remain relatively constant.

However, for a combined desuperheating and condensing process, the condensation heat transfer coefficient \(U_c\) will be different than \(U_d\) since the refrigerant convection coefficient \(h_{r,c}\) is different for condensation. Furthermore, the condensation heat transfer area \(A_c\) will vary based on the desuperheater operating conditions. Nevertheless, as long as the refrigerant does not fully condense and make a transition to slug flow, the condensation heat transfer area \(A_c\) remains relatively small when compared to the total area of the heat exchanger. Furthermore, four reported correlations for the condensation heat transfer coefficient \(h_{r,c}\) for a refrigerant in annular flow indicate that \(h_{r,c}\) remains relatively constant for the initial distance along the flow passage [Carey, 1992]. Therefore, even though the condensation heat transfer coefficient \(h_{r,c}\) itself may be larger than the desuperheating \(h_r\), the overall heat transfer coefficient-area product \(U_cA_c\) for partial condensation in the heat exchanger is smaller than \(U_dA_d\) and remains relatively constant for the same operating conditions. It will be seen in Chapter 3 that it is possible to calculate this typical \(U_cA_c\) from the combined desuperheating and condensing process using equation (19), once the desuperheating heat exchanger
effectiveness $\epsilon_d$ is known. It will also be seen that the magnitude of $U_cA_c$ relative to $U_dA_d$ is dependent on the presence of thermostats or control valves that attempt to limit condensation. Finally, these two $U_dA_d$ and $U_cA_c$ products, or their corresponding number of transfer units (NTU), characterize the performance of the desuperheater heat exchanger.
CHAPTER 3
DESUPERHEATER CHARACTERIZATION

Laboratory Testing

In 1992-93, four representative desuperheaters were tested at a range of entering water and refrigerant temperatures using the procedures specified in the 1987 version of ARI Standard 470 [Colon and Merrigan, 1993]. Appendix B contains a description of the equipment, the instrumentation, and the procedures that were used in this testing. (The actual test data can be found in the referenced test report.) This section presents a summary of the test results as well as an analysis of the data according to the $q$-$\text{LMTD}$ and $\epsilon$-$\text{NTU}$ heat exchanger performance equations discussed in Chapter 2. This data analysis is necessary for the estimate of the overall heat transfer coefficient-area products (UA) that are used in the performance simulation of the desuperheaters. Any special conditions or control features that must be considered for the simulation of each desuperheater type are also discussed.

Active Desuperheaters

Two commercially available models of an active desuperheater with thermostat controls were tested. One model (A1) utilized a tube-on-tube refrigerant-to-water heat exchanger while the other model (A2) used a tube-in-tube heat exchanger. Both models
were listed in the 1993 ARDM directory as rated for use on air conditioners up to a maximum capacity of 5 tons (17.6 kW). Model A1 had a listed "net superheat recovery" of 100 percent, while model A2 had a "net superheat recovery" slightly greater than 88 percent. Model A1 utilized a high limit thermostat on the entering water line that was set at 140 F (60°C), while model A2 used a 150 F (66°C) thermostat on the leaving water line. Both models also used a refrigerant thermostat that kept the circulating pump from operating unless the entering refrigerant temperature was greater than 115 F (46°C) for model A1 and 130 F (54°C) for model A2.

In order to illustrate some of the various operating regimes encountered in the testing, Figure 5 displays the useful heat exchanged $q$, as defined by equation (1), versus a range of entering water temperatures for active desuperheater A1. The higher sets of data points in Figure 5 represent a water flow rate through the desuperheater heat exchanger of approximately 1.5 gallons per minute (0.1 liters per second). This flow rate is the normal output of desuperheater A1’s circulating pump and, therefore, it is the regular operating condition that is encountered in a typical residential installation. However, ARI Standard 470 specifies that the temperature of the leaving water should be regulated at 140 F (60°C), so the lower set of data points in Figure 5 also indicates the useful heat exchanged when the water flow rate was externally throttled to maintain this temperature.

A least-squares linear regression analysis was applied both to the full water flow data and to the regulated outlet temperature data. The separate regression lines are plotted in Figure 5 to illustrate not only the linear relationship between the amount of heat exchanged and the inlet water temperature, but also the change in slope that occurs
Figure 5. Tested Capacity of Active Desuperheater A1 in Cooling Mode

for the full water flow condition at the lower entering water temperatures. As described in Appendix B, the desuperheaters were installed on a water-source heat pump with a nominal two ton (7 kW) cooling capacity in the laboratory testing. For water-cooled air conditioning equipment, the refrigerant saturation temperature specified in ARI Standard 470 was 105 F (40.6°C). Figure 5 indicates that when the entering water is below this saturation temperature, the slope of the line fitted to the full flow data points is considerably greater than the slope of the line for the full flow points above the saturation temperature. Therefore, as confirmed by the exiting refrigerant temperatures in the test data, condensation of the refrigerant vapor occurred at the full flow condition when the
entering water temperature was less than the saturation temperature (minus approximately 5 F (3°C)). When the water flow was throttled to obtain the Standard 470-specified outlet temperature, the slope of the line fitted to the data points remained the same for entering water temperatures both above and below the refrigerant saturation temperature. In other words, when the outlet temperature was regulated to 140 F (60°C), condensation of the refrigerant vapor did not occur. However, desuperheater A1 had a listed "net superheat recovery" of 100%; hence, condensation must have occurred when it was tested to the Florida modifications of Standard 470. (Table 2 shows that these modifications require the outlet temperature to be regulated to 130 F (54.4°C) for air-cooled systems and to 120 F (48.9°C) for water-cooled systems.)

In any case, control of the leaving water temperature is not the typical operating mode of this desuperheater. When the circulating pump is operating, the water flow through the desuperheater is unregulated. The test data reveals that, under these conditions, condensation of the refrigerant vapor may occur when the desuperheater is installed on air conditioners with a cooling capacity less than 5 tons (17.6 kW). (It should be noted here that the manufacturer of desuperheater A1 has recently introduced a similar desuperheater model that incorporates a temperature regulating valve to "circulate water through the heat exchanger until it heats to 120 F" (49°C) before it is released [Today's Air Conditioning, 1994]. As discussed in Chapter 1, this valve attempts to prevent condensation of the refrigerant within the desuperheater.)

Having identified the desuperheating and condensing regimes in the test data for active desuperheater A1, it is possible to next determine the log mean temperature difference (LMTD) according to equations (17) and (18) for these processes. Figure 6
Figure 6. Capacity Regimes of Active Desuperheater A1 in Cooling Mode

presents the useful heat exchanged $q$ versus the calculated LMTD for model A1 in its desuperheating (both full and throttled water flow) and condensing regimes. For the condensing regime, the desuperheating contribution to the useful heat exchanged has been deducted in order to calculate a "condensing only" LMTD$_C$. Therefore, the condensing regime data appears as the lowest set of points in Figure 6. Again a least-squares linear regression analysis is applied to the data and the separate regression lines for each operating condition are also shown in the figure. Correlation coefficients for the regression lines range from 94 to 99 percent. (The three highest points in Figure 6 were omitted from the desuperheating regression analysis, since it appears that some
condensation of the refrigerant vapor must have been occurring, even though the exiting refrigerant temperatures were slightly above the saturation temperature. As described in Appendix B, the temperature of the refrigerant was measured on the outside surface of the vertical refrigerant piping entering and exiting the desuperheater. Therefore, the exiting refrigerant temperature sensor would not have indicated saturation conditions unless the condensed refrigerant was in full annular flow over that section of the piping. For example, the omission of the three highest data points in Figure 6 improves the correlation coefficient of the calculated regression line from 94 to 99 percent.)

As indicated by equation (16), the overall heat transfer coefficient-area product UA for the heat exchanger equals the heat exchanged q divided by the LMTD. Therefore, the high correlation coefficients of the regression lines in Figure 6 indicates that both the desuperheating and condensing UA's are constant for the test conditions encountered by desuperheater A1. Furthermore, equation (16) is useful when plotting the measured effectiveness ε of the desuperheater heat exchanger, as defined by equation (9), versus the measured number of transfer units (NTU), since NTU equals the UA divided by \( C_{min} \), the minimum heat capacity rate of the two fluids. Figure 7 illustrates that desuperheater A1 closely follows the ε-NTU relationship for counterflow heat exchangers as defined in equations (10) and (14). The lowest set of points in Figure 7 is for the full water flow-condensing only regime where the measured NTU was less than 0.25, so that the effectiveness is effectively independent of the \( C_{min}/C_{max} \) ratio, as indicated by equation (14). The middle set of points is for the regulated 140°F (60°C) outlet temperature where the water flow rate varied and, hence, the \( C_{min}/C_{max} \) ratio ranged from 0.3 to 0.9. Finally, the upper set of data points is for the full water flow-
desuperheating only regime where $C_{\text{min}}/C_{\text{max}}$ was approximately equal to 0.10. The measured NTU for this regime ranged from approximately 2.6 to 3.6 and the effectiveness ranged from 0.90 to 0.96, respectively.

\[
\begin{align*}
\text{Effectiveness} & \quad \text{NTU} \\
\text{Condensing} & \quad (q/LMTD/C_{\text{min}}) \\
\end{align*}
\]

Figure 7. Effectiveness vs. the Number of Transfer Units (NTU) for Active Desuperheater A1

The number of transfer units can also be calculated from the measured effectiveness $\varepsilon$ and the ratio of the minimum to the maximum heat capacity rates ($C_{\text{min}}/C_{\text{max}}$), as indicated in equation (12). This calculated NTU is plotted in Figure 8 for the three LMTD regimes of desuperheater A1. For the full water flow-desuperheating only case, the calculated mean NTU of 3.6 with a standard error of 0.1 is relatively constant over the LMTD’s encountered in the testing. Again the three highest data points
are omitted from the least-squares regression analysis for this case. For the full water flow-condensing only case, a mean NTU of 0.2 with a standard error of 0.01 is also relatively constant over the condensing LMTDc’s. It is only for the atypical regulated water flow case that the calculated NTU varied from 1.5 to 2.4 over the LMTD’s.

![Diagram showing Number of Transfer Units for Active Desuperheater A1](image)

Figure 8. Number of Transfer Units for Active Desuperheater A1

Hence, Figure 8 indicates that it is possible to determine the number of transfer units or "heat transfer size" of desuperheater A1 from its measured effectiveness and $C_{min}/C_{max}$ ratio under a full water flow condition. From the definition of NTU in equation (11), it is also possible to estimate the overall heat transfer coefficient-area product UA for desuperheater A1 when it is installed on air conditioning equipment of other sizes and
refrigerant heat capacity rate. For example, using the typical refrigerant 22 mass flow rate (specified in the Florida regulatory modifications to ARI Standard 470) of 180 pounds per hour (81.6 kg/hr) per ton, the desuperheating \((UA)_d\) for model A1 installed on a three ton (10.5 kW) air conditioner would be estimated as

\[
(UA)_d = (NTU)_d (C_{min}) = (NTU)_d (m_c p_r) = 3.6(180 \text{ lbm/hr/ton})(3 \text{ tons})(0.210 \text{ Btu/lbm-F}) = 408 \text{ Btu/hr-F} = 215 \text{ W/°C}
\]

This method of estimating \((UA)_d\) is based on equation (22) which indicates that the Nusselt number for the refrigerant will increase by the change in the Reynolds number raised to the 0.8 power. Hence, if the refrigerant flow rate is increased by 1.5 when the air conditioning capacity is changed from two tons (7 kW) to three tons (10.5 kW), then the Nusselt number will increase by \((1.5)^{0.8} = 1.38\) or 92 percent of the capacity rate increase. Equation (20) also indicates that the refrigerant convection coefficient \(h_r\) is the only part of the \((UA)_d\) that will change when the refrigerant flow is changed. Therefore, the other terms in equation (20) remain constant and are not affected by the change in \(h_rA_r\).

For the "condensing only" case, the minimum heat capacity rate is that of the water. Therefore, the estimated condensing \((UA)_c\) remains constant whenever the water flow rate stays the same. This method of estimating \((UA)_c\) is consistent with the correlations that report a constant condensation heat transfer coefficient for a refrigerant in the initial distance along the flow passage [Carey, 1992].

\[
(UA)_c = (NTU)_c (C_{min}) = (NTU)_c (m_w c_{p,w}) = 0.2(1.5 \text{ gal/min})(60 \text{ min/hr})(8.33 \text{ lbm/gal})(0.997 \text{ Btu/lbm-F})
\]
\[ = 150 \text{ Btu/hr-F} \]
\[ = 79 \text{ W/\degree C} \]

Figure 9. Tested Capacity of Active Desuperheater A2 in Cooling Mode

A similar \( q-LMTD \) and \( \epsilon-NTU \) analysis was also conducted for active desuperheater A2. In its two lower sets of data points, Figure 9 presents the test results for this desuperheater for the same test conditions as desuperheater A1. Even though desuperheater A2 had a slightly higher water flow rate at the full flow condition than desuperheater A1, the actual heat transferred to the water by model A2's heat exchanger is less than model A1's. Therefore, at the full flow condition depicted by the middle set of points in Figure 9, condensation did not occur for entering water temperatures below the refrigerant saturation temperature of 105 F (40.6\degree C). When the water flow was
throttled to obtain the 140 °F (60°C) outlet temperature, the actual heat exchanged was even less, although the lower set of data points in Figure 9 reveals that the amount of heat exchanged decreased very little as the entering water temperature was increased.

To test the supposition that the overall heat transfer coefficient-area product (UA) for full water flow does not change considerably at other compressor operating conditions, the cooling mode test was also conducted at a refrigerant discharge temperature of 220 °F (104.4°C) and a discharge pressure corresponding to a refrigerant saturation temperature of 125 °F (51.7°C). (These temperatures are the air-cooled rating conditions specified in ARI Standard 470-87, as indicated in Table 1.) Figure 9 also depicts the useful heat exchanged for this condition in the separated upper set of data points. At the higher entering water temperatures, the slope of the regression line fitted to the rightmost set of points is nearly the same as the line fitted to the middle set of data points for the full water flow condition at the saturation temperature of 105 °F (40.6°C). At the colder entering water temperatures, however, it appears that condensation may also have occurred at the 125 °F (51.7°C) saturation temperature. Even though the exiting refrigerant temperatures in the test data were above the saturation temperature, the slight change of slope in the regression line indicates some condensation heat transfer was added to the desuperheating effect.

It must also be noted that the results in Figure 9 are only for cooling mode. When the desuperheater is installed on a heat pump, then condensation of the refrigerant may also occur in heating mode if the amount of available superheat is reduced. Accordingly, Figure 10 presents the useful heat exchanged versus the entering water temperature for active desuperheater A2 when the laboratory heat pump was switched to
the heating mode. As described in Appendix B, the constant area expansion device for the heat pump in heating mode had a smaller diameter than the separate expansion device used in cooling mode; therefore, the condensing pressure at the ARI Standard 470 compressor discharge temperature of 180 F (82.2°C) was higher than in cooling mode. Consequently, the refrigerant saturation temperature increased to 115.7 F (46.5°C) in heating mode and the amount of available superheat decreased. The exiting refrigerant temperatures in the test data confirm that condensation did occur for desuperheater A2 in heating mode when the entering water temperature was less than 95 F (35°C). Figure 10 indicates that condensation probably began to occur when the entering water
temperature was 105°F (40.6°C) or approximately 10°F (5.5°C) less than the refrigerant saturation temperature.

Figure 11. Capacity Regimes of Active Desuperheater A2 in Heating and Cooling Modes

Figure 11 next presents a $q$-$\Delta$MTD plot for desuperheater A2 that is similar to Figure 6 except that data are also shown for the space conditioning system in heating mode as well as in the second cooling mode. Furthermore, separate regression lines are plotted for both the "desuperheating only" and the "condensing only" data points in heating mode. A separate regression line is also shown in the upper righthand corner of Figure 11 for the combined desuperheating and condensing case for the second cooling mode. It is evident from the figure that the coefficients of the regression line for the
desuperheating only case in heating mode are equivalent to the coefficients of the
desuperheating regression line for the first cooling mode. Both lines are also nearly
equivalent to the desuperheating line for the second cooling mode. This supports the
premise that the overall heat transfer coefficient-area product (UA) for full water flow
does not change appreciably at typical compressor operating conditions, as long as
condensation does not occur.

![Diagram showing effectiveness vs. NTU for Active Desuperheater A2](image)

Figure 12. Effectiveness vs. the Number of Transfer Units (NTU) for Active Desuperheater A2

Figure 12 next displays the measured effectiveness of desuperheater A2's heat
exchanger versus the NTU that was determined by dividing the useful heat exchanged
$q$ by the product of the LMTD and $C_{\text{min}}$. Like Figure 7 for desuperheater A1, this figure
demonstrates that desuperheater A2 also follows the $\epsilon$-$NTU$ relations for counterflow heat exchangers. The full water flow-condensing only points had measured NTU’s less than 0.05 while the regulated outlet temperature data had the $C_{\text{min}}/C_{\text{max}}$ ratio vary from 0.3 to 0.8. The full flow cooling mode points are congregated at measured NTU’s that ranged from 1.0 to 1.15, while the full flow heating NTU’s ranged from 1.1 to 1.2.

Figure 13 then depicts the NTU that was calculated from the measured effectiveness and the $C_{\text{min}}/C_{\text{max}}$ ratio. For the full water flow condition, a mean NTU of 1.18 is approximately constant (standard error = 0.025) over the desuperheating LMTD’s in heating mode. A mean NTU of 1.11 is also relatively constant (standard
error = 0.015) over the wide range of desuperheating LMTD’s encountered during the cooling mode testing. This less than 7 percent difference in the mean NTU between heating and cooling modes for these "desuperheating only" cases is not considered significant when tested at the 95 percent confidence level. Even at the highest LMTD’s when condensation begins to occur, the difference in NTU is insignificant. Finally, for the full water flow-condensing only regime in heating mode, a relatively low mean (NTU)_c of 0.02 with a standard error of 0.005 is indicated. This low value illustrates that desuperheater A2 limits condensation of the refrigerant vapor, even on heat pumps of relatively small capacity.

The overall heat transfer coefficient-area products (UA) for active desuperheater A2 on a 3 ton (10.5 kW) heat pump at full water flow conditions were estimated in the same manner as desuperheater A1. Since the heat capacity rate for the refrigerant in heating mode was 99 percent of the heat capacity rate for cooling mode, the desuperheating (UA)_d for heating and for cooling was assumed to be equivalent and was estimated at 125 Btu/hr-F (66 W/°C). For the condensing only condition in heating mode, the (UA)_c was estimated to be 17 Btu/hr-F (9 W/°C) at desuperheater A2’s water flow rate of 1.7 gallons per minute (0.1 L/sec).

**Active Desuperheater with Outlet Temperature Control**

Active desuperheater A2 was also available in a model with a temperature regulating valve in order to control the temperature of the water leaving the desuperheater. As discussed in Chapter 1, this valve attempts to prevent condensation of the refrigerant vapor by throttling the water flow through the desuperheater heat
exchanger. This model was listed in the 1993 ARDM directory as having a "net superheat recovery" slightly higher than its sister model (but still less than 100 percent) and was also rated for use on air conditioners up to 5 tons (17.6 kW).

![Graph](image)

**Figure 14.** Tested Capacity of Active Desuperheater A2 (with an Outlet Temperature Control Valve) in Cooling Mode

Figure 14 displays the useful heat exchanged versus the entering water temperature for this active desuperheater with the outlet temperature control valve. This adjustable valve is normally set by the manufacturer in order to prevent the water in the heat exchanger from flowing until it reaches an exit temperature of 140 F (60°C). After the valve opens, water continues to flow until the temperature reaches 135 F (57.2°C) and then the valve closes. Hence, this opening and closing of the control valve was responsible for the three separate regimes of data points apparent in Figure 14. (As
described in Appendix B, the test period was for a fixed interval of 16 minutes. At the lower entering water temperatures, the valve stayed closed for a longer period of time than at higher entering water temperatures. At the highest entering water temperatures, the valve stayed open for all of the test interval.) Nevertheless, in the three regimes, approximately equal amounts of heat were transferred to the water. As expected, Figure 14 and the exiting refrigerant temperatures in the test data indicate that condensation did not occur in any of the three regimes.

Figure 15 presents the useful heat exchanged \( q \) versus the log mean temperature difference (LMTD) for desuperheater A2 with the outlet temperature control valve. At the lowest range of LMTD’s (highest entering water temperatures), the actual water flow
averaged 0.20 gpm (0.013 L/sec). At the higher LMTD’s, the overall water flow rate averaged 0.08 to 0.11 gpm (0.005-0.007 L/sec). The relatively similar slope of the regression lines plotted in Figure 15 for the three separate regimes of the data indicates that the rate of heat transfer for desuperheater A2 was relatively constant. However, because of the spread in the data, the regression correlation coefficients ranged from 71 percent to 86 percent.

Figure 16 presents the measured effectiveness of the desuperheater heat exchanger versus the measured NTU. For all but a few of the lower data points, the minimum heat capacity rate was that of the water. The $C_{min}/C_{max}$ ratio varied from 0.45
Figure 17. Number of Transfer Units for Active Desuperheater A2 (with an Outlet Temperature Control Valve)

to 1.0. Figure 17 displays the calculated NTU from the $e$-$NTU$ relationship versus the LMTD’s encountered during the testing. At the lowest LMTD’s, a mean NTU of 0.74 is relatively constant with a standard error of 0.01, since the water flow rate was relatively constant. Conversely, at the higher LMTD’s where the flow rate varied considerably, the NTU also varied. The overall heat transfer coefficient-area product $UA$ for a 3 ton (10.5 kW) heat pump was estimated to be 84 Btu/hr-F (44 W/°C) for the constant water flow condition. This UA value was approximately two-thirds of the desuperheating UA of 125 Btu/hr-F (66 W/°C) determined for active desuperheater A2
without an outlet temperature control valve. This difference is attributed to the smaller water flow rate through the temperature regulating valve.

Insert Desuperheater

The fourth commercially available desuperheater that was tested was a passive type in which the desuperheater heat exchanger was inserted inside the hot water storage tank. This insert desuperheater was the only passive type listed in the ARDM directory. It had a listed "net superheat recovery" of 64.5 percent and was rated for use on air conditioners with a maximum capacity of 3.5 tons (12.3 kW). Since there was no circulating pump included with this passive desuperheater, it was necessary to slightly modify the ARI Standard 470 testing procedure in order to determine the useful heat exchanged by the insert system.

Appendix B describes the modifications to the laboratory test equipment and procedures that were made for the insert system. The main difference between the modified test procedure and the procedure for an active desuperheater was that the water in the storage tank containing the insert desuperheater was drawn out immediately after the test in order to calculate the useful heat exchanged. However, it was found that the calculation of the heat exchanged by the refrigerant according to equation (5) was a more precise method of determining the efficiency of this passive desuperheater. For, unlike an active desuperheater, any heat lost between the measurement of the refrigerant entering and exit conditions had to be transferred to the water in the storage tank which surrounded the insert heat exchanger.
Figure 18 shows the tested capacity of the insert desuperheater in heating and cooling modes. The figure displays the useful heat exchanged for both the water (crosses) and the refrigerant (squares) versus the initial tank temperature for the insert desuperheater in both cooling and heating modes. As in the active desuperheater testing, the refrigerant saturation temperature was 105°F (40.6°C) in cooling mode and 115.7°F (46.5°C) in heating mode. Because of the modified test procedure, the correlation coefficient for the water regression lines (the solid lines shown in Figure 18) ranged from 23 percent for the data in cooling mode to 93 percent for the data in heating mode. The correlation coefficients for the dashed refrigerant regression lines, however, ranged from 89 to 95 percent, respectively. Although the exiting refrigerant temperature data did not indicate that condensation occurred in either mode, the change in slope of the upper
regression lines for cooling mode in Figure 18 indicates some condensation of the refrigerant must have occurred at the lowest initial tank temperatures. It is also interesting to note that condensation did not appear to occur in heating mode even though the amount of available superheat was less than in cooling mode. This may be explained by the slightly lower refrigerant flow rate in heating mode, especially since there was no forced water flow in the storage tank.

![Graph showing heat exchanged vs. log mean temperature difference (F)](image)

Figure 19. Capacity Regimes of Insert Desuperheater in Heating and Cooling Modes

Figure 19 presents the useful heat exchanged for the insert desuperheater for a "modified" log mean temperature difference (LMTD$_m$). Since the heat exchanger is inserted into the hot water storage tank, Farrington and Bingham [1986] suggest that the inlet and outlet water temperatures used in the definition of LMTD be set equal to the
initially constant storage tank temperature $T_i$. Therefore, equation (17) for the LMTD is modified to become

$$\text{LMTD}_m = \frac{(T_{r,0}) - (T_{r,i})}{\ln \left( \frac{(T_{r,0} - T_i)}{(T_{r,i} - T_i)} \right)}$$

where only the refrigerant temperatures and the initial tank temperature $T_i$ are used. (The upper data points in Figure 19 where condensation is suspected to be occurring have not had their desuperheating contribution deducted as in Figures 6 and 11, since the useful heat exchanged is still less than the measured available superheat.)

![Figure 20. Effectiveness vs. the Number of Transfer Units (NTU) for Insert Desuperheater](image)

Figure 20 presents the measured effectiveness of the insert desuperheater versus the measured UA divided by the minimum heat capacity rate. Since the water in the
storage tank has a much larger heat capacity relative to the refrigerant flow, the $C_{min}$ was always that of the refrigerant. However, the insert desuperheater cannot be considered a true counterflow heat exchanger like the active desuperheaters, since there is no forced flow on the water side. Nevertheless, the insert desuperheater still follows the $e$-NTU relationship expressed in equation (14) for all heat exchangers when $C_{min}/C_{max} = 0$. This relationship is shown as the dotted line in Figure 20 over the range of measured NTU’s.

Figure 21 displays the NTU calculated from equation (15) for both the water and refrigerant heat exchanged in the insert desuperheater. Both the cooling and heating NTU’s are relatively constant and the water and refrigerant means for desuperheating only are within a value of 0.02 of each other. However, the correlation coefficient for the refrigerant heat transfer was always higher than the correlation for the heat exchanged on the water side. This comparison demonstrates the usefulness of the refrigerant measurements to determine the number of transfer units for this type of passive desuperheater. Similar to desuperheater A2, the small difference between the heating and cooling NTU is also considered insignificant at the 95 percent confidence level. A mean NTU of 0.43 with a standard error of 0.01 is indicated by the results in Figure 21.

For desuperheating only, the estimated UA for an insert desuperheater on a 3 ton (10.5 kW) heat pump was 51 Btu/hr-F (27 W/°C). In the combined desuperheating and condensing mode, the estimated UA was 53 Btu/hr-F (28 W/°C). This less than 4 percent difference in UA between desuperheating only and desuperheating plus condensing modes for the insert desuperheater is considered insignificant when compared with the condensing only $(UA)_c$ calculated for desuperheaters A1 and A2.
Figure 21. Number of Transfer Units for the Insert Desuperheater

Summary

In summary, Table 4 presents the estimated UA for the four tested desuperheater models when they are installed on a 3 ton (10.5 kW) heat pump.
Table 4

Estimated Overall Heat Transfer Coefficient - Area Product UA for Installation on a 3 Ton (10.5 kW) Heat Pump

<table>
<thead>
<tr>
<th>Desuperheater Model</th>
<th>Desuperheating Btu/hr-F (W/°C)</th>
<th>Condensing Btu/hr-F (W/°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1 - full water flow</td>
<td>408 (215)</td>
<td>150 (79)</td>
</tr>
<tr>
<td>A2 - full water flow</td>
<td>125 (66)</td>
<td>17 (9)</td>
</tr>
<tr>
<td>A2 with outlet temp. control</td>
<td>84 (44)</td>
<td>Not Applicable</td>
</tr>
<tr>
<td>Passive Insert</td>
<td>51 (27)</td>
<td>Not Significant</td>
</tr>
</tbody>
</table>

Table 5 further summarizes the circulating pump characteristics and the thermostat control capabilities of the four desuperheater models.

Table 5

Desuperheater Pump and Control Characteristics

<table>
<thead>
<tr>
<th></th>
<th>A1</th>
<th>A2</th>
<th>A2 w/Valve</th>
<th>Insert</th>
</tr>
</thead>
<tbody>
<tr>
<td>Circulating Pump</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Flow Rate (gpm)</td>
<td>1.5 (340 kg/hr)</td>
<td>1.7 (385 kg/hr)</td>
<td>0.2 (45 kg/hr)</td>
<td>N/A</td>
</tr>
<tr>
<td>Power (W)</td>
<td>92 (331 kJ/hr)</td>
<td>87 (313 kJ/hr)</td>
<td>87 (313 kJ/hr)</td>
<td>N/A</td>
</tr>
<tr>
<td>High Limit Thermostat</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Setting Location</td>
<td>140 F (60°C) Water In</td>
<td>150 F (66°C) Water Out</td>
<td>150 F (66°C) Water Out</td>
<td>160 F (71°C) Tank Top</td>
</tr>
<tr>
<td>Refrigerant Thermostat</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Setting Location</td>
<td>115 F (46°C) R-22 In</td>
<td>130 F (54°C) R-22 In</td>
<td></td>
<td>N/A</td>
</tr>
</tbody>
</table>
CHAPTER 4
SIMULATION MODEL

The results of the desuperheater heat exchanger performance characterizations are then used in a simulation model in order to account for the transient effect of changing entering water temperatures, outdoor air temperatures, and compressor discharge temperatures on daily desuperheater performance. The selected simulation model, TRNSYS -- a Transient System Simulation program [Klein et al., 1975-76] -- allows for daily hot water use, heat energy supplied to the hot water tank by backup electric resistance elements, and heat loss through the tank shell. TRNSYS also permits the water heating model to be combined with a simulation of a building that is conditioned by an air conditioner and/or heat pump and whose operation is controlled by a thermostat.

TRNSYS itself is a modular program in which mathematical models of individual system components are connected together to form a complete system for simulation. The performance of each individual component depends on its characteristic parameters, the performance of other components, and/or time-dependent forcing functions. Written in American National Standards Institute (ANSI) compliant FORTRAN-77, the source code for each TRNSYS component was compiled using the Microsoft FORTRAN PowerStation product to create a 32-bit application that will run on a 386 or higher
personal computer without the 640 kilobyte limitation of the Microsoft Disk Operating System (DOS). The latest version of the TRNSYS program -- TRNSYS 14.1 introduced in October 1994 -- also includes a number of new utility programs that make it easier to use than its predecessors.

Figure 22. TRNSYS Component Model of an Active Desuperheater Water Heating System

Figure 22 graphically represents how an active desuperheater water heating system is modeled in TRNSYS by connecting the inputs and outputs of the following components: a heat exchanger, a storage tank with heating elements, a circulating pump, connecting piping, and a pump controller. In addition, a residential desuperheater system simulation must also include a building load or house component that incorporates solar
heat gain, an air conditioner/heat pump, and a thermostat controller, as represented in Figure 23. Figure 24 then combines the components in the previous two figures along with Typical Meteorological Year (TMY) weather data, a solar radiation processor (SRP), time-dependent forcing functions (TDFFs), and some TRNSYS output components to depict the full TRNSYS desuperheater simulation that was developed for this project.

Appendix C contains the actual text of the TRNSYS input file that is shown graphically in Figure 24. The major components of the input file will be discussed in
Figure 24. Complete TRNSYS Simulation Model of a Residential Desuperheater Water Heating System
this section; however, in general, each component is described by its characteristic parameters and its required inputs along with each input’s initial values. Lines starting with an asterisk are interpreted by the TRNSYS program as comments. Therefore, the outputs of each component are only listed in Appendix C for completeness and ease of programming.

As indicated in Figure 24, the weather and the hot water load are two time-dependent forcing functions that are necessary to determine transient system performance. The TRNSYS program then solves the set of algebraic and differential equations that describe the system at a user-selectable timestep. The simulation timestep selected for this project was 6 minutes (0.1 hour) in order to keep the average deviation of the daily energy balance on the hot water storage tank to approximately one percent. However, such a small timestep caused the annual simulation of 8760 hours to require over 2.4 million iterative calls to the component models and take approximately 15 hours to run on a 20 Megahertz 386-based personal computer! Increasing the timestep increased the percentage deviation of the energy balance and the simulation still took overnight to achieve an accuracy of approximately five percent. Fortunately, on a 100 Megahertz Pentium personal computer, the annual simulation with the 6 minute timestep took less than an hour to run.

**Building Load**

TRNSYS permits a building’s heat loss or gain to be modeled by using a single conductance or overall heat transfer coefficient (UA), for the structure. The building load is then calculated using a simple energy balance that includes thermal capacitance
effects and sensible heat gains. It is recognized that the instantaneous energy loads calculated in this manner may be occasionally in error; however, over a period of time, this method provides a reasonable estimate of the heating and cooling loads with a minimum of computational effort. Furthermore, this method avoids the need for a detailed description of the physical characteristics of a standard building. And, as will be seen in the following discussion, the use of a single heat transfer coefficient for the structure considerably simplifies the proper sizing of the space conditioning system.

### Table 6

**Florida Energy Efficiency Code for Building Construction**

1993 Residential Baseline Requirements

<table>
<thead>
<tr>
<th>Component</th>
<th>North Florida</th>
<th>Central Florida</th>
<th>South Florida</th>
</tr>
</thead>
<tbody>
<tr>
<td>Walls</td>
<td>R-19 frame</td>
<td>R-19 frame</td>
<td>R-19 frame</td>
</tr>
<tr>
<td>Glazing Type</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Glazing Orientation</td>
<td>Double clear</td>
<td>Double clear</td>
<td>Double clear</td>
</tr>
<tr>
<td>Overhang</td>
<td>None</td>
<td>Same</td>
<td>Same</td>
</tr>
<tr>
<td>Ceiling</td>
<td>R-30</td>
<td>R-30</td>
<td>R-30</td>
</tr>
<tr>
<td>Floor</td>
<td>Slab R-3.5</td>
<td>Slab R-3.5</td>
<td>Slab R-O</td>
</tr>
<tr>
<td>Cooling system</td>
<td>10.0 SEER</td>
<td>10.0 SEER</td>
<td>10.0 SEER</td>
</tr>
<tr>
<td>Heating system</td>
<td>6.8 HSPF</td>
<td>Electric</td>
<td>Electric</td>
</tr>
<tr>
<td>Ducts</td>
<td>R-6</td>
<td>R-6</td>
<td>R-6</td>
</tr>
<tr>
<td>Hot water</td>
<td>.88 EF electric</td>
<td>.88 EF</td>
<td>.88 EF</td>
</tr>
</tbody>
</table>

A nominal (UA)ₙ of 1000 Btu/hr-F (1900 kJ/hr-C) was selected for the standard residential building conductance in north, central, and south Florida. The choice of a constant building (UA)ₙ throughout Florida is consistent with the Florida Energy
Efficiency Code for Building Construction’s baseline requirements for residential structures that are listed in Table 6 [FDCA, 1994].

Using the heat transfer relationship

\[ q = (UA)\Delta T \]  

with the 97.5 percent winter design dry bulb temperature [ASHRAE, 1993b] and an indoor dry bulb temperature \( (T_{DB}) \) of 68 F \( (20^\circ C) \), Table 7 indicates the calculated sensible heating load for the standard residential building located in the cities of Jacksonville, Orlando, and Miami.

### Table 7

Winter Design Temperatures and Residential Heating Load

<table>
<thead>
<tr>
<th>City</th>
<th>97.5% Design ( T_{DB} )</th>
<th>Heating Load</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>F ( ^{\circ} C )</td>
<td>Btu/hr</td>
</tr>
<tr>
<td>Jacksonville</td>
<td>32 ( 0 )</td>
<td>36,000</td>
</tr>
<tr>
<td>Orlando</td>
<td>38 ( 3 )</td>
<td>30,000</td>
</tr>
<tr>
<td>Miami</td>
<td>47 ( 8 )</td>
<td>21,000</td>
</tr>
</tbody>
</table>

As seen in Table 7, the typical heating load requires a minimum three ton (10.5 kW) heat pump in north Florida. In central Florida, a two and one-half ton (8.8 kW) heat pump or a nominal 10 kW electric resistance heater would be required and in south Florida, an approximate two ton (7.0 kW) heating load is indicated. Since the capacity of cooling equipment at the 95 F \( (35^\circ C) \) rating condition typically equals the heat pump capacity at the 47 F \( (8.3^\circ C) \) standard heating rating condition [ARI, 1994] and, since the summer design temperatures in north, central, and south Florida are approximately the
same [ASHRAE, 1993b], the heating loads in Table 7 are also assumed to be typical of a house with a corresponding three ton (10.5 kW) cooling load in all three regions of Florida.

In addition to the overall heat transfer coefficient \((UA)_h\), TRNSYS also utilizes a lumped capacitance to account for the thermal mass of the building. Based on short-term test data, Balcomb et al. [1993] suggest that an effective heat capacity of 6.4 Btu/°F-ft\(^2\) (131 kJ/m\(^2\)-C) is a reasonable choice for a furnished frame house built on a concrete slab-on-grade with perhaps two-thirds of the floor carpeted. They further suggest that a heat capacity of 4.0 Btu/°F-ft\(^2\) (82 kJ/m\(^2\)-C) is reasonable for a furnished frame house built over a crawlspace or basement. As indicated by Table 6, the slab-on-grade value was chosen for the simulation of a standard Florida residence.

Since the effective heat capacity was suggested on an area basis, a 1,500 square foot (139 m\(^2\)) house was also selected as the standard residence size to be used in the simulation. While the mean single family residential floor area in Florida between 1987 and 1993 was reported as 1,915 ft\(^2\) (178 m\(^2\)) in a 1993 University of Florida study of 4,758 randomly-selected Energy Code submittal forms, the median floor area was actually between 1,400 and 1,699 ft\(^2\) (130-158 m\(^2\)). An earlier Florida Power and Light study of 165 customers in 1978-79 found that 83 percent of all sample houses were less than 2,000 ft\(^2\) (186 m\(^2\)) with the average being 1,466 ft\(^2\) (136 m\(^2\)) [Paxson et al., 1980]. Furthermore, the average central air conditioner size in the FPL study -- which also measured actual air conditioner operating hours and thermostat settings -- was exactly 3.0 tons (10.5 kW), consistent with the building loads calculated in Table 7. Therefore, in
order to use compressor operating hours as a calibration tool for the building load simulation, a nominal 1,500 ft\(^2\) (139 m\(^2\)) floor area was selected for this project.

Finally, TRNSYS also permits the building load to be modified with time-varying sensible heat gains. Therefore, solar gains are incorporated into the calculation of the building load through the use of east and west facing windows as well as a pitched roof and attic component with R-30 insulation. The longitudinal axis of the house was oriented east-west with a 5:3 aspect ratio and the roof was pitched at 5 in 12 (22.6 degrees from the horizontal). The surface of the roof was assumed to be covered with generic light-colored shingles with a solar absorptance of 0.85 and a far infrared emittance of 0.91 [Parker et al., 1993]. As indicated in Table 6, the clear double-pane glazing was 15 percent of the floor area with a U-factor of 0.7 Btu/hr-ft\(^2\)-F (4 W/m\(^2\)-°C) and a solar transmittance of 60 percent [ASHRAE, 1993b].

Weather Data

Typical Meteorological Year (TMY) weather data for the three cities of Jacksonville, Orlando, and Miami was used in the TRNSYS annual simulation. 8760 hours of solar radiation, dry-bulb temperature, and wind speed data were accessed by TRNSYS’s 1975-named "Card Reader" component. Next TRNSYS’s "Solar Radiation Processor" converted the direct normal and the global horizontal solar radiation data to the total amount of radiation falling on the east and west-facing vertical windows and the north and south-facing sloped roof. The Perez et al. [1988] diffuse radiation model was used to calculate the diffuse portion of this total from the direct beam and the reflected
solar radiation. Finally, for the timesteps between the hourly TMY data, the weather
data was interpolated.

**Air Conditioner/Heat Pump**

Of course, developing a rating for a desuperheater water heater also involves the
actual performance of the air conditioner/heat pump that it may be packaged with or be
installed on in the field. Higher efficiency space-conditioning units typically operate at
lower refrigerant discharge temperatures such that the amount of available superheat is
less than it is for less efficient units. Reedy [1991] presented the compressor discharge
temperature versus the Seasonal Energy Efficiency Ratio (SEER) for two levels of
compressor efficiencies representing a good "standard" compressor and the best available
in 1991. He showed that, although the use of a higher efficiency compressor alone does
not significantly affect discharge temperature, the discharge temperature will decrease
from over 200 F (93°C) for an 8 SEER system to almost 160 F (71°C) for a 12 SEER
system. A typical 10 SEER system -- the minimum efficiency level currently permitted
by the National Appliance Energy Conservation Act of 1987 for air-cooled units less than
65,000 Btu/hr (19 kW) -- had a 180 F (82°C) discharge temperature at the 95 F (35°C)
ARI test condition.

Reedy also showed the dependance of the compressor discharge temperature for
a 10 SEER air-source system on outdoor temperature for both a capillary tube and a
thermal expansion valve. With a thermal expansion valve, the discharge temperature
increases with increasing outdoor temperature. However, with a capillary tube, the
discharge temperature remains relatively constant or even decreases slightly with
increasing outdoor temperature. Therefore, in order to rate the water heating efficiency of all desuperheater systems equally, a nominal three ton (10.5 kW) capillary tube air-source space-conditioning unit that met the 1994 Florida Energy Code’s minimum requirements of a 10.0 SEER and/or a HSPF of 6.8 was first selected for this project. Based on Reedy’s 1991 work, the compressor discharge temperatures for this nominal unit were selected to be 180 F (82°C) for the 95 F (35°C) cooling test condition and 182 F (83°C) at 85 F (29.4°C). In addition, a 12.0 SEER capillary tube air conditioning unit with a 162 F (72.2°C) discharge temperature at the 95 F (35°C) test condition was also selected for simulation in this project. Again based on Reedy’s data, the discharge temperature was set at 164 F (73.3°C) at 85 F (29.4°C). For ambient air temperatures between or outside 85 F (29.4°C) and 95 F (35°C), the compressor discharge temperature was linearly interpolated or extrapolated, respectively. For both the 10.0 and 12.0 SEER heat pumps in heating mode, the compressor discharge temperature was set at 100 F (56°C) above the outside ambient temperature. This temperature difference was based on the common method of checking the proper refrigerant charge in a heat pump in heating mode [Air-Conditioning, Heating, and Refrigeration News, 1989].

The three ton (10.5 kW) air conditioner/heat pump heating and cooling capacities are taken specifically from Carrier Corporation Model 38YK036. Figures 25 and 26 display the 38YK036 heating and cooling capacities respectively at the rated outside air temperatures. This information is entered into TRNSYS as a performance map so that building energy balance can include the heating and cooling equipment inputs at any ambient temperature. For north and central Florida, a 5 kW supplemental electric resistance heat strip was also required to be used with the heat pump at temperatures
lower than the house balance point. As shown in Figure 25, this balance point (neglecting solar gains) is approximately 38 F (3°C). Therefore, the 5 kW supplemental resistance heater is only used when the outside temperature is less than 38 F (3°C). For simulation of the straight 38YK036 air conditioner, a 10 kW electric resistance heater is used as the only source of heating.

The thermostat settings for the air conditioner/heat pump were selected to be a nominal 77 F (25°C) for cooling and 68 F (20°C) for heating. Florida Power and Light (FPL) found in its 1978-79 air conditioning study that the predominant cooling setting was 78 F (25.6°C), although the average setting was 75.7 F (24.3°C).[Paxson et al.,
Vieira and Parker [1991] found in a questionnaire study of over 370 Florida homes that the average cooling setting was 77.7 F (25.4°C) and the average heating setting was 71.4 F (21.9°C). The majority of the homeowners also set back the heating thermostat, so a nighttime heating setting of 55 F (12.8°C) was used in the simulation between 11:00 p.m. and 6:00 a.m. Finally, a standard thermostat deadband of 2.0 F (1.1°C) was used [Henderson, 1992]. These standard selections further emphasize that the energy factor rating method is meant as a relative comparison of water heating efficiencies under the same loads and the same operating conditions.
Auxiliary Electric Water Heater

An auxiliary water heater is necessary in a desuperheater system to provide water heating when space conditioning is not necessary or when the hot water demand exceeds the desuperheater’s capacity. Again this project chose an electric water heater that met the 1994 Florida Energy Code’s minimum requirement of an energy factor of 0.88 for a 40 gallon (151 liter) water heater. The electric water heater had interlocking 4500 Watt elements and polyurethane foam R-12 insulation. The temperature setting of the upper element thermostat was nominally chosen as 131°F (55°C) and the lower thermostat was set down to 113°F (45°C), typically the minimum possible setting as recommended in the installation instructions of most desuperheater manufacturers. From laboratory testing, Huggins [1994] has determined that the deadband on the lower element thermostat is typically 5°F (2.8°C), while the upper thermostat has a much larger deadband of 22°F (12.2°C). The tank environment temperature was set at 75°F (23.8°C) and the cold water temperature was specified as 72°F (22.2°C), the same temperatures used for calculating the Florida Energy Factor for solar water heaters, as described in Chapter 1 [Block et al., 1993]. The water heater’s physical dimensions were taken from Mor-Flo Industries Model MEFR 40D, which has a listed energy factor of 0.88 [GAMA, 1994].

The TRNSYS component model of a stratified hot water storage tank uses all this characteristic information to perform an energy balance on each of three separate segments of the tank -- the top, middle, and bottom. Duffie and Beckman (1980) suggest that three tank segments represent a reasonable compromise between no segmentation at all and a model with a large number of segments representing a high level of
stratification that may not be achievable in actual practice. The temperatures of each segment are then determined by the TRNSYS program through the integration of their time derivatives using the modified Euler analytical technique. Finally, the energy flows into and out of each segment as well as any changes in internal energy are calculated from the temperatures at each timestep.

Hot Water Use

The reported national average hot water use of 64.3 gallons (243.4 liters) a day [Gilbert et al., 1985] was selected as the hot water load to be simulated in TRNSYS. As discussed in Chapter 1, this is the same hot water load used in the DOE water heating test procedures for the determination of the energy factor for electric and gas water heaters. Fanney and Dougherty [1992] have also shown experimentally that the draw schedule for the DOE water heating load does not affect the calculation of energy factor for electric resistance water heaters. Therefore, the time-of-day schedule for the simulation was taken from the measured average pattern of weekday and weekend hot water energy use for 24 solar hot water systems during 1985 [Merrigan, 1988]. Shown in Figure 27, these measured hourly profiles were compared by Becker and Stogsdill [1990] to other measured profiles from the United States and Canada and were found to be graphically similar as long as the other profiles excluded senior citizens and rental units from the comparison. Furthermore, the distinction between weekday and weekend profiles was found to be significant. As indicated in Figure 27, the weekday pattern had the maximum hot water use during the hour ending at 7:00 a.m. and a secondary use peak between 7:00 and 8:00 p.m. The weekend pattern displays maximum hot water use
Figure 27. Average Hot Water Use Profile of 24 Residential Hot Water Heaters

between 8:00 and 10:00 a.m. and a secondary peak from 7:00 to 10:00 p.m. This later morning peak on the weekend understandably corresponds to the difference in schedules between the normal work week and Saturday, Sunday, and holidays. Therefore, the weekend hot water use pattern was also used on nine federal holidays (New Year’s Day, President’s Day, Memorial Day, the Fourth of July, Labor Day, Columbus Day, Veteran’s Day, Thanksgiving, and Christmas) for the annual water heating simulation.
As mentioned earlier, an active desuperheating system was simulated in TRNSYS by connecting mathematical models of a heat exchanger and a circulating pump to the hot water storage tank. TRNSYS models a counterflow heat exchanger according to the effectiveness relationship expressed in equation (10) and then determines the temperature of the water leaving the heat exchanger from equation (9). In order to determine the number of transfer units (NTU) of the heat exchanger, the overall heat transfer coefficient was estimated from the laboratory testing, as described in Chapter 3 and summarized in Table 4. However, since the desuperheaters were tested on a two ton (7 kW) heat pump in the laboratory, but were being simulated on a three ton (10.5 kW) unit in TRNSYS, it was decided to use only the desuperheating contribution to the overall UA of the heat exchanger. As indicated in Table 4, even on the two ton (7 kW) heat pump, the condensing heat transfer coefficient was relatively insignificant for all but one of the four tested desuperheaters. This decision therefore rewards those desuperheater designs that attempt to control condensation of the refrigerant, since condensation could lead to unstable compressor operation and a loss of space conditioning system efficiency.

The refrigerant flow rate through the heat exchanger was set at 540 lbm/hr (245 kg/hr) for the standard three ton (10.5 kW) heat pump, based on the typical R-22 mass flow of 180 lbm/hr (0.02268 kg/s) per ton. The water flow rate of the circulating pump was determined from the laboratory testing. The electrical power for the circulating pump was taken from the actual pump specifications, as indicated in Table 5. If the desuperheater utilized high limit thermostats or a temperature control valve, a controller component was also used in TRNSYS to control the circulating pump operation or the
water flow. Furthermore, the circulating pump would not operate unless the house thermostat called for compressor operation.

The size of the connecting piping between the hot water storage tank and the heat exchanger was specified as 5/8 inch (1.6 cm) outside diameter and 25 feet (7.6 m) in length each way, in general accordance with common practice and the ARDM Residential Heat Recovery Installation Guide.[1988] The piping was covered with nominal 1/2 inch (1.25 cm) closed cell foam pipe insulation with an actual thickness of 0.4 inch (1.0 cm) and a thermal conductivity of 0.26 Btu-in/hr-ft²-F (0.037 W/m-°C).

The insert desuperheating system was simulated by modifying the TRNSYS component model of a stratified hot water storage tank, since there was no available component in TRNSYS for a passive heat exchanger. Fortunately, the TRNSYS tank component did model heat loss to an optional internal gas flue, so this FORTRAN subroutine was modified to simulate the insert heat exchanger. Physically, the insert heat exchanger was 39.25 inches (1 meter) in length and was installed vertically so that it ran almost the full height of the storage tank. This geometry was very comparable to the conditions assumed for a gas flue in that its heat was transferred to the water in all segments of the storage tank. The heat transfer within the tank was programmed using equation (8) in the instantaneous energy balance for each segment. The heat exchanger effectiveness was calculated from equation (14) and, as discussed in Chapter 3, the minimum heat capacity rate $C_{\text{min}}$ was always that of the refrigerant for the insert desuperheater. The overall heat transfer coefficient of the heat exchanger was estimated from the laboratory testing and again only the desuperheating contribution was used.
Appendix D contains the FORTRAN source code for this modified TRNSYS component and the insert heat exchanger additions have been highlighted.
CHAPTER 5
SIMULATION RESULTS

Annual Operating Hours

In order to check the reasonableness of the building load parameters described in the previous chapter, the annual operation of the air conditioner/heat pump was first simulated in TRNSYS without a desuperheater. Table 8 indicates the simulated annual compressor operating hours of the standard heat pump using the Typical Meteorological Year (TMY) hourly solar radiation and weather data for the three cities of Jacksonville, Orlando, and Miami. Similarly, Table 9 indicates the simulated annual operating hours of a "straight-cool" air conditioner as well as the operating hours of a 10 kW electric resistance heat strip.

Table 8
Simulated Annual Heat Pump Compressor Operating Hours

<table>
<thead>
<tr>
<th>City</th>
<th>Cooling Load Hours</th>
<th>Heating Load Hours</th>
<th>Total Load Hours</th>
</tr>
</thead>
<tbody>
<tr>
<td>Jacksonville</td>
<td>1390</td>
<td>813</td>
<td>2203</td>
</tr>
<tr>
<td>Orlando</td>
<td>1608</td>
<td>395</td>
<td>2003</td>
</tr>
<tr>
<td>Miami</td>
<td>1869</td>
<td>136</td>
<td>2005</td>
</tr>
</tbody>
</table>
Table 9

Simulated Annual Air Conditioner (with Electric Resistance Heat) Operating Hours

<table>
<thead>
<tr>
<th>City</th>
<th>Cooling Load Hours</th>
<th>Heating Load Hours</th>
<th>Total Load Hours</th>
</tr>
</thead>
<tbody>
<tr>
<td>Jacksonville</td>
<td>1387</td>
<td>903</td>
<td>2290</td>
</tr>
<tr>
<td>Orlando</td>
<td>1606</td>
<td>462</td>
<td>2068</td>
</tr>
<tr>
<td>Miami</td>
<td>1870</td>
<td>163</td>
<td>2033</td>
</tr>
</tbody>
</table>

While the cooling load hours for both the heat pump in Table 8 and the straight air conditioner in Table 9 are nearly equivalent, the heating load hours for the 10 kW electric strip heater exceed the heat pump hours because the heat pump also relies on a 5 kW supplemental heater when the outdoor temperature falls below 38 F (3.3°C). All three cities indicate slightly more than 2,000 total heating and cooling load hours.

For comparison, Florida Power & Light (FPL) reported 1850 average annual compressor operating hours for cooling only in its 1978-79 study of 165 customers. [Paxson et al., 1980] This result was for central air conditioners that operated under full thermostatic control, as in this project’s TRNSYS simulation. No mention was made in the FPL report of where the customers were located; however, the largest part of FPL’s service territory lies in south Florida.

For comparison to the heating load hours, the U.S. Department of Energy publishes in its "Uniform Test Method for Measuring the Energy Consumption of Central Air Conditioners" a map of the heating load hours for the United States. A line of 500 hours crosses central Florida and a line of 1000 hours runs through the Florida panhandle and exits above Jacksonville. The map also indicates that 750 regional heating load
hours can be used for most of north and central Florida [Code of Federal Regulations, 1993].

**Annual Water Heating Energy Factor**

The annual operation of the electric water heater was also simulated in TRNSYS without any desuperheater interaction. Removing a daily load of 64.3 gallons (243 L) at the hot water draw profile and thermostat settings specified in the previous chapter resulted in annual energy factor (EF) of 0.90. This was two percent greater than the rated EF of 0.88 determined in a 24 hour laboratory test for the simulated water heater. However, it is almost within the 1.5 percent estimated experimental uncertainty associated with determining the laboratory energy factor [Fanney and Dougherty, 1992]. The remaining deviation is attributed to the difference between the measured and the rated insulation value of the storage tank. Furthermore, the 24 hour laboratory test procedure requires that the storage tank thermostats be adjusted to provide a mean tank temperature of $135 \pm 5^\circ F (57.2 \pm 2.8^\circ C)$, but it does not specify the exact settings. However, for heat pump water heaters, the laboratory test procedure does specify that the thermostat should be set in accordance with the manufacturer’s installation instructions. This was the thermostat procedure followed for the desuperheater simulation.

**Annual Desuperheater Energy Factor**

The annual operation of the desuperheater was then simulated in TRNSYS by combining it with the building load and the auxiliary electric water heater. Figures 28
and 29 present the results of one typical summer week of the annual simulation for desuperheater A2 installed on a 10.0 SEER heat pump in Orlando. Figure 28 displays the daily change in the outside ambient temperature (TAMB), the simulated room temperature (TROOM), the desuperheater outlet temperature (TUNIT), and the temperature of the hot water supplied to the load (TLOAD). Figure 29 displays the "on-time's" of the air conditioner/heat pump (KWAC or KWHP), the 4500 watt electric resistance water heater (KWAUX), and the 87 watt desuperheater circulating pump (KWPUMP) by plotting the electrical demand (in kilowatts) for these appliances. (The horizontal time axis in both figures is labelled "Zeit" because the plots were generated by a TRNSYS component developed by the German TRNSYS distributor.)

A complete annual output from the same TRNSYS simulation presented in Figures 28 and 29 is contained in Appendix E for active desuperheater A2. This text output first echoes the control statements of the TRNSYS input file and then prints the requested output in the form of histograms and summary tables. Between the echoed input and the histograms are some warning messages as well as the performance data of the nominal three ton (10.5 kW) air conditioner and heat pump/heat strip (logical units 11 and 12). Following this section are annual average histogram plots (in daily kilowatt-hours) for the following calculations: the solar load on the building's roof (KWROOF), the sensible cooling load of the building (QSENSL), the total building cooling load (QCOOL), the air conditioner's electric use (KWHAC), the building heating load (QHEAT), the heating system's electric use (KWHHP), the desuperheater contribution to the storage tank (KWUNIT), the desuperheater's circulating pump energy (KWPUMP), the auxiliary electric element energy use (KWAUX), and the total electric
Figure 28. Simulated Temperatures for 7 July Days in Orlando

Figure 29. Simulated Appliance Electrical Demand for 7 July Days in Orlando
energy use (KWTOT) for the pump and the auxiliary electric element. These average histograms indicate the annual hourly profile of each quantity over a 24-hour period. Three additional histograms indicate the total cooling load hours (CLH), the total heating load hours (HLH), and the total desuperheater operating hours (RUNHRS), i.e., the length of time the desuperheater circulating pump turned on during the year. (This time may be less than the total of the cooling load hours and the heating load hours because of the desuperheater controls.) After the histograms in Appendix E are monthly tables summarizing the energy flows and temperatures of the annual simulation. These quantities are also used to calculate an energy balance on the storage tank as a means of determining the simulation accuracy. As can be seen at the end of Appendix E, the storage tank model in TRNSYS was able to calculate the annual energy flows into and out of the tank with a relative inaccuracy of less than one-quarter of one percent for active desuperheater A2 installed on a heat pump.

Active Desuperheaters

Table 10 next presents the simulated annual energy factor for active desuperheater A1 installed on both a 10.0 SEER and 12.0 SEER heat pump as well as on both a 10.0 and 12.0 SEER straight air conditioner for the three cities of Jacksonville, Orlando, and Miami. Table 11 presents the simulated annual desuperheater operating hours for the same conditions. The simulated operating hours and energy factors for desuperheater A1 installed on a heat pump are understandably higher than for the desuperheater installed on an air conditioner. Furthermore, the simulated energy factors for desuperheater A1 installed on a 10.0 SEER heat pump or straight air conditioner are
expectedly higher than for the desuperheater installed on a 12.0 SEER unit. However, the simulated operating hours for the desuperheater on a 10.0 SEER heat pump or air conditioner are slightly less than the operating hours on a 12.0 SEER unit, because the higher discharge temperature of the 10.0 SEER unit activates the high limit thermostat on the desuperheater water piping more often than the lower discharge temperature of the 12.0 SEER unit.

Table 10
Simulated Annual Energy Factor for Active Desuperheater A1

<table>
<thead>
<tr>
<th>City</th>
<th>Heat Pump</th>
<th>Air Conditioner</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10.0 SEER</td>
<td>12.0 SEER</td>
</tr>
<tr>
<td>Jacksonville</td>
<td>2.4</td>
<td>2.1</td>
</tr>
<tr>
<td>Orlando</td>
<td>2.3</td>
<td>2.0</td>
</tr>
<tr>
<td>Miami</td>
<td>2.4</td>
<td>2.0</td>
</tr>
</tbody>
</table>

The simulated heat pump energy factors for desuperheater A1 are also slightly higher in north Florida than in central Florida since the heat pump operating hours are higher. However, the simulated heat pump energy factors are also as high or higher in south Florida than in central Florida because the cooling load hours are considerably higher, as indicated in Table 8. Correspondingly, the operating hours of desuperheater A1 installed on a straight air conditioner are highest in south Florida, leading to the highest energy factors for desuperheater A1 installed on an air conditioner.
Table 11

Simulated Annual Operating Hours for Active Desuperheater A1

<table>
<thead>
<tr>
<th>City</th>
<th>Heat Pump</th>
<th>Air Conditioner</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10.0 SEER 12.0 SEER</td>
<td>10.0 SEER 12.0 SEER</td>
</tr>
<tr>
<td>Jacksonville</td>
<td>2152 2169</td>
<td>1344 1360</td>
</tr>
<tr>
<td>Orlando</td>
<td>1937 1972</td>
<td>1556 1578</td>
</tr>
<tr>
<td>Miami</td>
<td>1926 1973</td>
<td>1810 1837</td>
</tr>
</tbody>
</table>

Table 12 next presents the simulated annual energy factor for active desuperheater A2 installed on both a heat pump and on a straight air conditioner for the two SEER levels in the three Florida cities. Table 13 presents the simulated desuperheater operating hours for the same conditions. Although desuperheater A2’s annual operating hours are slightly higher than desuperheater A1, the simulated energy factors for desuperheater A2 -- with a 70 percent smaller heat exchanger UA -- are 7 to 21 percent lower than desuperheater A1. The simulated energy factors for desuperheater A2 installed on a heat pump are again as high or slightly higher in north and south Florida than in central Florida. The operating hours of desuperheater A2 on a straight air conditioner are also highest in south Florida, again leading to the highest energy factor for this category.
Table 12

Simulated Annual Energy Factor for Active Desuperheater A2

<table>
<thead>
<tr>
<th>City</th>
<th>Heat Pump</th>
<th>Air Conditioner</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10.0 SEER</td>
<td>12.0 SEER</td>
</tr>
<tr>
<td>Jacksonville</td>
<td>1.9</td>
<td>1.7</td>
</tr>
<tr>
<td>Orlando</td>
<td>1.9</td>
<td>1.6</td>
</tr>
<tr>
<td>Miami</td>
<td>2.0</td>
<td>1.7</td>
</tr>
</tbody>
</table>

Table 13

Simulated Annual Operating Hours for Active Desuperheater A2

<table>
<thead>
<tr>
<th>City</th>
<th>Heat Pump</th>
<th>Air Conditioner</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10.0 SEER</td>
<td>12.0 SEER</td>
</tr>
<tr>
<td>Jacksonville</td>
<td>2178</td>
<td>2179</td>
</tr>
<tr>
<td>Orlando</td>
<td>2001</td>
<td>2001</td>
</tr>
<tr>
<td>Miami</td>
<td>2005</td>
<td>2005</td>
</tr>
</tbody>
</table>

Active Desuperheater with Outlet Temperature Control

Table 14 next presents the simulated annual energy factor for active desuperheater A2 with the outlet temperature control valve installed on both a heat pump and on a straight air conditioner for the two SEER levels in the three cities of Jacksonville, Orlando, and Miami. Table 15 presents the simulated desuperheater operating hours for the same conditions. Although the operating hours of the circulating pump are almost the same as desuperheater A2 in Table 13, the simulated energy factors for desuperheater A2 with the control valve are 8 to 18 percent lower than its sister
model without the valve in Table 12. Again, the operating hours of this desuperheating unit on a straight air conditioner are highest in south Florida, leading to the highest energy factor for this category.

Table 14

Simulated Annual Energy Factor for Active Desuperheater A2 with an Outlet Temperature Control Valve

<table>
<thead>
<tr>
<th>City</th>
<th>Heat Pump</th>
<th></th>
<th></th>
<th>Air Conditioner</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10.0 SEER</td>
<td>12.0 SEER</td>
<td>10.0 SEER</td>
<td>12.0 SEER</td>
<td></td>
</tr>
<tr>
<td>Jacksonville</td>
<td>1.6</td>
<td>1.4</td>
<td>1.3</td>
<td>1.2</td>
<td></td>
</tr>
<tr>
<td>Orlando</td>
<td>1.6</td>
<td>1.4</td>
<td>1.4</td>
<td>1.2</td>
<td></td>
</tr>
<tr>
<td>Miami</td>
<td>1.7</td>
<td>1.4</td>
<td>1.6</td>
<td>1.4</td>
<td></td>
</tr>
</tbody>
</table>

Table 15

Simulated Annual Operating Hours for Active Desuperheater A2 with an Outlet Temperature Control Valve

<table>
<thead>
<tr>
<th>City</th>
<th>Heat Pump</th>
<th></th>
<th></th>
<th>Air Conditioner</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10.0 SEER</td>
<td>12.0 SEER</td>
<td>10.0 SEER</td>
<td>12.0 SEER</td>
<td></td>
</tr>
<tr>
<td>Jacksonville</td>
<td>2178</td>
<td>2176</td>
<td>1386</td>
<td>1385</td>
<td></td>
</tr>
<tr>
<td>Orlando</td>
<td>2001</td>
<td>1999</td>
<td>1606</td>
<td>1605</td>
<td></td>
</tr>
<tr>
<td>Miami</td>
<td>2005</td>
<td>2002</td>
<td>1870</td>
<td>1867</td>
<td></td>
</tr>
</tbody>
</table>

Insert Desuperheater

Table 16 next presents the simulated annual energy factor for the passive insert desuperheater installed on both a heat pump and on a straight air conditioner for the two SEER levels in the three Florida cities. Table 17 presents the simulated desuperheater
operating hours for the same conditions. The operating hours for the 10.0 SEER unit are approximately 400 hours less than the heating and cooling load compressor operating hours presented in Tables 8 and 9 because the insert desuperheater control valve allows refrigerant to bypass the heat exchanger when the water temperature at the top of the storage tank reaches 160°F (71°C). This solenoid valve has a measured electrical consumption of 17 watts when it is energized or approximately 7 kilowatt-hours over the whole year. (This relatively small electrical energy consumption has still been included in the calculation of the simulated energy factor.) In contrast, the operating hours for the 12.0 SEER unit are nearly equivalent to the compressor operating hours in Tables 8 and 9, because the compressor discharge temperature itself is 162°F (72.2°C) at an outdoor temperature of 95°F (35°C); therefore, the solenoid valve is almost never activated.

Table 16

Simulated Annual Energy Factor for the Insert Desuperheater

<table>
<thead>
<tr>
<th>City</th>
<th>Heat Pump</th>
<th>Air Conditioner</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10.0 SEER</td>
<td>12.0 SEER</td>
</tr>
<tr>
<td>Jacksonville</td>
<td>2.7</td>
<td>2.5</td>
</tr>
<tr>
<td>Orlando</td>
<td>2.5</td>
<td>2.3</td>
</tr>
<tr>
<td>Miami</td>
<td>2.6</td>
<td>2.4</td>
</tr>
</tbody>
</table>
### Table 17

Simulated Annual Operating Hours for the Insert Desuperheater

<table>
<thead>
<tr>
<th>City</th>
<th>Heat Pump</th>
<th>Air Conditioner</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10.0 SEER</td>
<td>12.0 SEER</td>
</tr>
<tr>
<td>Jacksonville</td>
<td>1814</td>
<td>2176</td>
</tr>
<tr>
<td>Orlando</td>
<td>1583</td>
<td>2002</td>
</tr>
<tr>
<td>Miami</td>
<td>1580</td>
<td>2005</td>
</tr>
</tbody>
</table>

### Summary

In summary, Tables 18 through 20 present the simulated annual energy factors for the four tested desuperheaters installed on both a heat pump and a straight air conditioner for the two SEER levels in the cities of Jacksonville, Orlando, and Miami, respectively.

### Table 18

Simulated Desuperheater Energy Factors for Jacksonville (North Florida)

<table>
<thead>
<tr>
<th>Desuperheater Model</th>
<th>Heat Pump</th>
<th>Air Conditioner</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10.0 SEER</td>
<td>12.0 SEER</td>
</tr>
<tr>
<td>A1 - full water flow</td>
<td>2.4</td>
<td>2.1</td>
</tr>
<tr>
<td>A2 - full water flow</td>
<td>1.9</td>
<td>1.7</td>
</tr>
<tr>
<td>A2 with outlet valve</td>
<td>1.6</td>
<td>1.4</td>
</tr>
<tr>
<td>Insert</td>
<td>2.7</td>
<td>2.5</td>
</tr>
</tbody>
</table>
Table 19

Simulated Desuperheater Energy Factors for Orlando (Central Florida)

<table>
<thead>
<tr>
<th>Desuperheater Model</th>
<th>Heat Pump</th>
<th>Air Conditioner</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10.0 SEER</td>
<td>12.0 SEER</td>
</tr>
<tr>
<td>A1 - full water flow</td>
<td>2.3</td>
<td>2.0</td>
</tr>
<tr>
<td>A2 - full water flow</td>
<td>1.9</td>
<td>1.6</td>
</tr>
<tr>
<td>A2 with outlet valve</td>
<td>1.6</td>
<td>1.4</td>
</tr>
<tr>
<td>Insert</td>
<td>2.5</td>
<td>2.3</td>
</tr>
</tbody>
</table>

Table 20

Simulated Desuperheater Energy Factors for Miami (South Florida)

<table>
<thead>
<tr>
<th>Desuperheater Model</th>
<th>Heat Pump</th>
<th>Air Conditioner</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10.0 SEER</td>
<td>12.0 SEER</td>
</tr>
<tr>
<td>A1 - full water flow</td>
<td>2.4</td>
<td>2.0</td>
</tr>
<tr>
<td>A2 - full water flow</td>
<td>2.0</td>
<td>1.7</td>
</tr>
<tr>
<td>A2 with outlet valve</td>
<td>1.7</td>
<td>1.4</td>
</tr>
<tr>
<td>Insert</td>
<td>2.6</td>
<td>2.4</td>
</tr>
</tbody>
</table>

The simulated results in these regional tables indicate that the annual desuperheater energy factor depends on the design of the desuperheater, the compressor discharge temperature of the space conditioning unit (as reflected in its Seasonal Energy Efficiency Ratio), the operating hours of the space conditioning unit, and if the desuperheater is installed on an air conditioner or a heat pump. The highest energy factors resulted from the simulation of the passive insert desuperheater and active
desuperheater A1. Both these desuperheaters have double-wall heat exchanger designs that place the tubes separating or containing the refrigerant and the water in metal-to-metal contact with each other. Energy factors were also 8 to 20 percent higher when a desuperheater was installed on a space conditioning unit with a 10.0 SEER rather than a 12.0 SEER cooling efficiency. In addition, higher energy factors resulted when the desuperheater was installed on a heat pump rather than on a straight air conditioner. However, this space conditioning system difference was more significant in north and central Florida than it was in south Florida. Finally, the simulated annual energy factor for each desuperheater model installed on a heat pump with the same SEER level did not differ significantly by geographical location in Florida. This was primarily a consequence of the result shown in Table 8 that the total heating and cooling load hours were approximately equal in all three regions of the state.
CHAPTER 6
DISCUSSION AND RECOMMENDATIONS

Field Comparison

In order to check the reasonableness of the desuperheater simulation results presented in the previous chapter, an approximate comparison can be made to selected field monitored results that are reported in the literature. However, it must be emphasized that monitored results are influenced by many uncontrolled field variables, such as the actual weather and the homeowner's interaction with his space conditioning and water heating systems, in addition to any specific deviations from the average parameter values assumed in the simulation. Nevertheless, it is still informative to make such a comparison, if only to determine that the simulation results are of the right order of magnitude.

Since some field monitored results in the desuperheater literature are presented as percentage energy savings, it is also necessary to convert the simulated energy factor (EF) to this expression. The conversion appears relatively straightforward at first, since a desuperheater EF of 2.0 should result in a 50 percent energy savings. However, it must be remembered that energy savings are relative to some other type of water heater. If that baseline water heater is an electric resistance unit with an EF of 0.90, then a 50 percent energy savings actually results from a desuperheater EF of 1.8. Table 21
therefore presents the equivalent percentage energy savings relative to an electric resistance water heater (EF of 0.90) for desuperheater EF’s ranging from 1.0 to 2.9.

Table 21

<table>
<thead>
<tr>
<th>EF</th>
<th>%</th>
<th>EF</th>
<th>%</th>
<th>EF</th>
<th>%</th>
<th>EF</th>
<th>%</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>10</td>
<td>1.5</td>
<td>40</td>
<td>2.0</td>
<td>55</td>
<td>2.5</td>
<td>64</td>
</tr>
<tr>
<td>1.1</td>
<td>18</td>
<td>1.6</td>
<td>44</td>
<td>2.1</td>
<td>57</td>
<td>2.6</td>
<td>65</td>
</tr>
<tr>
<td>1.2</td>
<td>25</td>
<td>1.7</td>
<td>47</td>
<td>2.2</td>
<td>59</td>
<td>2.7</td>
<td>67</td>
</tr>
<tr>
<td>1.3</td>
<td>31</td>
<td>1.8</td>
<td>50</td>
<td>2.3</td>
<td>61</td>
<td>2.8</td>
<td>68</td>
</tr>
<tr>
<td>1.4</td>
<td>36</td>
<td>1.9</td>
<td>52</td>
<td>2.4</td>
<td>63</td>
<td>2.9</td>
<td>69</td>
</tr>
</tbody>
</table>

In 1982, Eno reported annual desuperheater energy savings in the range of 17 to 35 percent for six residential desuperheater units in the central Florida area. These field monitored results are similar to those reported by Merrigan [1983] for approximately the same time period; however, the latter field results were presented as an average annual Coefficient of Performance (COP) -- roughly equivalent to the definition of annual Energy Factor -- of 1.2 for five desuperheaters in east central Florida. For these five plus 15 other desuperheaters in north, west central, and south Florida, the average annual COP was reported as 1.1. However, the best eight performing desuperheaters had an average annual COP of 1.5. For comparison, the average annual COP for 19 electric resistance water heaters was reported as 0.8 for the same time period.

Just as the energy factor for electric resistance water heaters has improved to the 0.9 range since 1982, the performance of desuperheaters can also be expected to have improved over that same time. As mentioned in Chapter 1, this improvement has been
asserted by the Association of Refrigerant Desuperheater Manufactures (ARDM) since 1986. Although there are no completed field monitored studies in the recent literature, a current on-going study by the Florida Solar Energy Center has reported interim performance results for 10 Florida desuperheaters [Merrigan and Colon, 1992]. For example, active desuperheater A2 was installed on a 11.5 SEER heat pump in central Florida and operated for 2,376 hours with an annual COP of 1.9. The passive insert desuperheater was installed on a 9.85 SEER air conditioner in south Florida and had an annual COP of 2.6 with a compressor operating time of 2,483 hours. Both these monitored COP’s are approximately similar to the simulated energy factors presented in Tables 19 and 20 in Chapter 5, although the SEER levels are not exactly the same and the monitored compressor operating hours are 19 and 33 percent higher, respectively.

For a comparison to a monitored result where the operating hours were nearly equivalent to the simulation, desuperheater A2 with the outlet temperature control valve was installed on a 12.8 SEER heat pump in north Florida and operated for 2,184 hours with an annual COP of 1.2. This monitored COP is 14 percent less than the simulated energy factor presented in Table 18; however, the SEER level of the monitored heat pump is also somewhat higher. Furthermore, as the individual owners of the monitored systems reduced their compressor operating hours, the monitored annual desuperheater COP correspondingly decreased. A low annual COP of 1.1 was reported for active desuperheater A1 installed on a 12.0 SEER air conditioner that only operated for 883 yearly hours in central Florida. These results emphasize again that the energy factor rating method developed in this project is meant as a relative comparison of water
heating efficiencies and is not meant as a prediction of actual field performance or even long-term reliability.

**Project Accomplishments**

As indicated in Chapter 2, it was the objective of this project to develop an energy factor rating method for desuperheater water heaters that accurately accounts for changing compressor operating conditions and water temperatures. In order to meet that objective, it was necessary to first modify the existing procedure for testing desuperheater water heaters -- ARI Standard 470 [1987] -- in order to obtain characteristic performance data on both active and passive desuperheaters. Specifically, active desuperheaters were tested at their normal operational flow rate rather than at a controlled outlet temperature as required in ARI Standard 470. Passive insert desuperheaters were tested using a modified log mean temperature difference ($LMTD_m$) based on the refrigerant inlet and outlet temperatures and the initial tank temperature.

The desuperheater testing also revealed that it was necessary to test desuperheaters at more than just one entering water temperature in order to describe their performance. Specifically, it was necessary to test desuperheaters at entering water temperatures both above and below the refrigerant saturation temperature in order to account for both desuperheating and possible condensation of the refrigerant. In addition, condensation was shown to be possible when desuperheaters are tested according to the Florida regulatory modifications to ARI Standard 470, resulting in a "net superheat recovery" of 100 percent. Use of the conventional heat exchanger concepts of effectiveness and number of transfer units (already presented in ARI Standard 470)
was shown to be the preferred method of reporting desuperheater test results. Finally, even though condensation was not a significant mode of heat transfer for three out of the four tested desuperheaters, it was important to identify when it was occurring, since uncontrolled condensation could possibly cause a loss in space conditioning system efficiency.

Once the performance of the desuperheater heat exchanger was determined, it was combined with a simplified residential building load and the performance characteristics of both a standard air conditioner/heat pump and an electric resistance hot water tank in the TRNSYS simulation program. Compressor discharge temperatures corresponding to Seasonal Energy Efficiency Ratios (SEER) of 10.0 and 12.0 were used for a space conditioning unit with a capillary tube expansion device. In addition, it was necessary to modify the TRNSYS component model of a stratified hot water storage tank in order to simulate the passive insert heat exchanger. Annual simulations were then performed using average hourly weather data for the three Florida cities of Jacksonville, Orlando, and Miami.

The results of the TRNSYS simulations without a desuperheater indicated that the assumptions concerning the building load, the air conditioner/heat pump, the electric water heater, and the hot water use were realistic. The results of the TRNSYS simulations with a desuperheater showed that the annual energy factor was dependent on the type of desuperheater, the climate location, the operating hours and SEER level of the space conditioning unit, and whether the desuperheater was installed on an air conditioner or a heat pump. However, for units installed on a heat pump, the simulated
annual energy factor did not differ significantly by location, since the total annual heating and cooling load hours were approximately equivalent in all three Florida cities.

**Recommendations**

The energy factor rating method developed in this project should be used to replace the existing desuperheater water heater rating method in the Florida Energy Efficiency Code for Building Construction. Active desuperheater units need to be tested to a slightly modified Air-Conditioning and Refrigeration Institute (ARI) Standard 470 procedure in order to determine their overall heat transfer coefficient for desuperheating. It is recommended that this testing be conducted on a heat pump/air conditioner of 3 ton (10.5 kW) capacity and that the desuperheater outlet temperature not be externally regulated. Passive insert desuperheaters would also need to be tested according to procedures similar to those presented in Appendix B.

The simulation methodology developed for this project should then be used to determine the energy factor for each tested desuperheater in the three climate regions of Florida. Hot water credit multipliers for the Energy Code residential compliance procedure could be calculated in the same manner as it is presently done for heat pump water heaters and solar water heating systems, in which one credit multiplier is used for a range of energy factors (e.g., 2.0-2.49). If this same methodology is adopted for desuperheaters, it is also recommended to use one SEER level (e.g., 11.0 SEER) in the simulation procedure in order to simplify the rating calculation and still be within ± 10 percent of the minimum and maximum expected performance. This is the same percentage variation that was used in determining the one energy factor rating for solar
water heating systems installed at various collector tilt angles and orientations in both north Florida and in central and south Florida combined [Block et al., 1993]. The Energy Code would then finally have one overall, consistent method of comparing the efficiencies of all water heating system types to each other.
APPENDIX A

ASSOCIATION OF REFRIGERANT DESUPERHEATER MANUFACTURERS

Directory of Certified Refrigerant Desuperheater Heat Recovery Unit Water Heaters
DIRECTORY OF CERTIFIED REFRIGERANT DESUPERHEATER
HEAT RECOVERY UNIT
WATER HEATERS

EFFECTIVE: February 15, 1992 - February 14, 1993

THE HRU'S LISTED ARE CERTIFIED IN TESTS CONDUCTED IN ACCORDANCE WITH FEEC STD 100-89 (ARI STD 470-87, WITH FLORIDA MODIFICATION)

Administered By

ASSOCIATION of REFRIGERANT DESUPERHEATER MANUFACTURERS Inc.

9B Northlake, Orange City, Florida, 32763-6197
(904) 774-6923

SUPERSEDES ALL PREVIOUS DIRECTORIES

PRICE: $1.50
INTRODUCTION & DEFINITIONS

To qualify for listing in the Directory of Certified Refrigerant Desuperheater Heat Recovery Units, each unit listed must have been tested and certified by an independent testing laboratory to the Florida Regulatory Modifications to ARI Standard 470-87, effective January 1, 1989.

The significant definitions and procedures in the standard can be summarized as follows:

1. DESUPERHEATER WATER HEATER: A factory-made assembly of elements by which the flows of refrigerant vapor and water are maintained in such heat transfer relationship that the refrigerant vapor is desuperheated and the water is heated. A water circulating pump may be included as part of the assembly.

2. TOTAL SYSTEM HOT GAS SUPERHEAT: The total heat removal required to completely desuperheat the refrigerant discharge vapor. This value is the product of the mass flow of refrigerant and the difference in enthalpy between the refrigerant vapor entering the desuperheater and the vapor at saturation leaving the desuperheater.

3. TOTAL USEFUL HEAT EXCHANGE EFFECT: The total heat transferred to the water in the heat recovery heat exchanger, corrected for the effect of the water circulating pump, if used.

4. NET SUPERHEAT RECOVERY: The ratio of the total useful heat exchange effect to the total system hot gas superheat expressed as a percentage.

5. TESTING, will be performed at one or both of the following standard rating conditions:

<table>
<thead>
<tr>
<th>Type System</th>
<th>Saturated Temp. of Entering Refrig. Vapor</th>
<th>Actual Temp. of Entering Refrig. Vapor</th>
<th>Temp. of Entering Water</th>
<th>Temp. of Leaving Water</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>F C</td>
<td>F C</td>
<td>F C</td>
<td>F C</td>
</tr>
<tr>
<td>Air Cooled</td>
<td>125 51.7</td>
<td>220 104.4</td>
<td>80 26.7</td>
<td>130 54.4</td>
</tr>
<tr>
<td>Water Cooled</td>
<td>105 40.6</td>
<td>180 82.2</td>
<td>80 26.7</td>
<td>120 48.9</td>
</tr>
</tbody>
</table>

6. PUBLISHED RATINGS: Ratings published in the Directory will include:
   A. Unit Model Number
   B. Maximum A/C tons which is the maximum capacity of the air conditioning unit on which that model should be used, expressed in tons.
   C. Net superheat recovery.

INDEX TO MANUFACTURERS OF CERTIFIED HRU'S

*AMERICAN ENERGY PROD.
John Borzoni
11307 43rd Street North
Clearwater, FL 34622
(813) 573-1490

*AMERICAN EQUIPMENT SYS., CORP.
Dick Williamson
190 Scarlet Blvd.
Oldsmar, FL 34677
(813) 854-5411

CRISPAIRE CORP.
David Shuford
3570 American Dr.
Atlanta, GA 30341
(404) 458-6643

*ENERGY CONSERVATION UNLIMITED
Rod Weaver
311 E. Georgia Avenue
PO Box 585
Longwood, FL 32752
(407) 834-0400

*ENRO MFG., INC.
Dick McKinnon
6461 Garden Road, Suite 102
Riviera Beach, FL 33404
(407) 845-0465

*FHP MFG.
DIV. OF HARROW PROD.
Brad Grahl
601 NW 65th Ct.
Ft. Lauderdale, FL 33309
1-800-352-4328

NATIONAL ENERGY SYSTEMS
Brian Jones
651 Day Hill Road
Windsor, Ct. 06095
(203) 683-2005

*ADDISON PRODUCTS COMPANY
WEATHERKING DIVISION
Brad Harris
PO Box 607715
Orlando, FL 32860-7715
(407) 292-4400

*Member of ARDM
Basic Order of Classification of HRU's:

<table>
<thead>
<tr>
<th>Type</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>FP-HR</td>
<td>Active, or forced pump heat recovery unit.</td>
</tr>
<tr>
<td>IT-HR</td>
<td>Insert type heat recovery unit.</td>
</tr>
<tr>
<td>TS-HR</td>
<td>Passive, or thermo-syphon heat recovery unit.</td>
</tr>
</tbody>
</table>

**AMERICAN EQUIPMENT SYSTEMS**

**TRADE NAME: DYNAMAX, LECTRA-SAVER, ULTRA-SAVER, ENCON**

<table>
<thead>
<tr>
<th>TYPE: FP-HR</th>
<th>MAXIMUM A/C TONS</th>
<th>NET SUPERHEAT RECOVERY</th>
</tr>
</thead>
<tbody>
<tr>
<td>MODEL NUMBER</td>
<td>50RA, V</td>
<td>100.0%</td>
</tr>
<tr>
<td></td>
<td>L6000DW, V</td>
<td>100.0%</td>
</tr>
<tr>
<td></td>
<td>E6000DW, V</td>
<td>100.0%</td>
</tr>
</tbody>
</table>

**AMERICAN ENERGY PRODUCTS**

**TRADE NAME: AQUEFIER**

<table>
<thead>
<tr>
<th>TYPE: FP-HR</th>
<th>MAXIMUM A/C TONS</th>
<th>NET SUPERHEAT RECOVERY</th>
</tr>
</thead>
<tbody>
<tr>
<td>MODEL NUMBER</td>
<td>R4K</td>
<td>73.8%</td>
</tr>
<tr>
<td></td>
<td>R6K</td>
<td>91.3%</td>
</tr>
<tr>
<td></td>
<td>R9K</td>
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**CRISPAIRE CORP.**

**TRADE NAME: E-TECH**

<table>
<thead>
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<th>TYPE: FP-HR</th>
<th>MAXIMUM A/C TONS</th>
<th>NET SUPERHEAT RECOVERY</th>
</tr>
</thead>
<tbody>
<tr>
<td>MODEL NUMBER</td>
<td>H-60</td>
<td>88.6%</td>
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**ENERGY CONSERVATION UNLIMITED**

**TRADE NAME: ECU**

<table>
<thead>
<tr>
<th>TYPE: FP-HR</th>
<th>MAXIMUM A/C TONS</th>
<th>NET SUPERHEAT RECOVERY</th>
</tr>
</thead>
<tbody>
<tr>
<td>MODEL NUMBER</td>
<td>HR06</td>
<td>88.4%</td>
</tr>
<tr>
<td></td>
<td>DSO6</td>
<td>92.2%</td>
</tr>
</tbody>
</table>

**ENRO MANUFACTURING, INC.**

**TRADE NAME: ENRO**

<table>
<thead>
<tr>
<th>TYPE: IT-HR</th>
<th>MAXIMUM A/C TONS</th>
<th>NET SUPERHEAT RECOVERY</th>
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</thead>
<tbody>
<tr>
<td>MODEL NUMBER</td>
<td>SHS-1</td>
<td>50.8%</td>
</tr>
<tr>
<td></td>
<td>SHS-2</td>
<td>64.5%</td>
</tr>
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</table>

**FHP MANUFACTURING DIV., HARROW PRODUCTS**

**TRADE NAME: FLORIDA HEAT PUMP**

<table>
<thead>
<tr>
<th>TYPE: FP-HR</th>
<th>MAXIMUM A/C TONS</th>
<th>NET SUPERHEAT RECOVERY</th>
</tr>
</thead>
<tbody>
<tr>
<td>MODEL NUMBER</td>
<td>HR-P1</td>
<td>100.0%</td>
</tr>
<tr>
<td></td>
<td>HR-P2</td>
<td>100.0%</td>
</tr>
<tr>
<td></td>
<td>HR-P3</td>
<td>100.0%</td>
</tr>
</tbody>
</table>
**NATIONAL ENERGY SYSTEMS**  
**TRADE NAME: ESP**

<table>
<thead>
<tr>
<th>TYPE: FP-HR</th>
<th>MODEL NUMBER</th>
<th>MAXIMUM A/C TONS</th>
<th>NET SUPERHEAT RECOVERY</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>341-S</td>
<td>5</td>
<td>52.2%</td>
</tr>
<tr>
<td></td>
<td>341-SE</td>
<td>5</td>
<td>69.4%</td>
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**ADDISON PRODUCTS COMPANY, WEATHERKING DIVISION**  
**TRADE NAME: WEATHERKING**

<table>
<thead>
<tr>
<th>TYPE: FP-HR</th>
<th>MODEL NUMBER</th>
<th>MAXIMUM A/C TONS</th>
<th>NET SUPERHEAT RECOVERY</th>
</tr>
</thead>
<tbody>
<tr>
<td>HRX5</td>
<td>5</td>
<td>92.2%</td>
<td></td>
</tr>
<tr>
<td>HRV-5</td>
<td>5</td>
<td>92.2%</td>
<td></td>
</tr>
<tr>
<td>HRV-5F</td>
<td>5</td>
<td>92.2%</td>
<td></td>
</tr>
<tr>
<td>HRO-5</td>
<td>5</td>
<td>88.4%</td>
<td></td>
</tr>
<tr>
<td>HRO-5F</td>
<td>5</td>
<td>88.4%</td>
<td></td>
</tr>
</tbody>
</table>

* INDICATES A CHANGE SINCE LAST LISTING.
APPENDIX B

LABORATORY TESTING METHODOLOGY

1. Refrigerant System Description
2. Water System Description
3. Test Procedures
4. Instrumentation & Data Reduction
LABORATORY TESTING METHODOLOGY

Refrigerant System Description

An existing water-source heat pump was used in the laboratory testing of the desuperheaters. The heat pump’s cooling capacity was rated at 24,200 Btu/hr (7.1 kW) at a condensing water flow rate of 6 gallons per minute (0.4 L/s) and an entering water temperature of 85°F (29.4°C). The Energy Efficiency Ratio (EER) of the unit was 10.85 when rated according to the procedures in ARI Standard 320 [1986]. The rated heating capacity was 30,500 Btu/hr (8.9 kW) with a Coefficient of Performance (COP) of 3.5 at an entering water temperature of 70°F (21°C). For all of the heating and cooling mode tests on the desuperheaters, the condenser entering water temperatures were between 70°F and 85°F (21-29°C).

The refrigerant used in the heat pump and consequently, in the refrigerant side of the desuperheaters, was R-22. Most of the testing performed under cooling mode had a R-22 flow rate of approximately 6 pounds/minute (2.7 kg/min). During heating mode testing, the R-22 flow rate was slightly lower, averaging 5.3 pounds/minute (2.4 kg/min). After steady state operating conditions were established, the measured refrigerant flow rates in the desuperheater tests were always within 6 percent of these mean values.
Figure B1 displays the refrigerant and water piping that connected the heat pump, the active desuperheater(s), and the water storage tanks in the laboratory testing. The refrigerant piping from the indoor compressor unit to the active desuperheater unit(s) was limited to approximately 5 feet (1.5 m) in length, using 0.5 inch (1.27 cm) outside diameter (O.D.) copper tubing. Following the desuperheater’s exit refrigerant line and the heat pump reversing valve, 0.75 inch (1.91 cm) O.D. tubing was extended to the water-source heat exchanger. A sight glass was located after the heat exchanger in order to permit a visual inspection of the refrigerant charge during cooling mode testing. The proper refrigerant charge was also verified by the suction line superheat temperature and the liquid line subcooling temperature.

A direct expansion coil and blower fan unit were installed in a separate indoor chamber. This unit was equipped with a constant area expansion device for cooling (device #67 in Figure B1) that incorporated a bored-through piston of .067 inches (0.17
cm) inside diameter (I.D.) at the inlet to the expansion coil. The indoor chamber itself was furnished with four electrical heaters rated at 1500 Watts apiece. The heaters were individually controlled to provide the sensible heat load necessary to reach the desired cooling conditions.

The heat pump was also equipped with a reversing valve to allow operation in heating mode. While the direction of refrigerant flow through the compressor and the desuperheater was the same as in cooling mode, the refrigerant moved in the opposite direction through the system components after the reversing valve during heating operation. The constant area expansion device for heating (device #59 in Figure B1) was located at the inlet to the water-source heat exchanger. This expansion device had a slightly smaller diameter (.059 inches (0.15 cm) I.D.) than the cooling mode expansion device at the indoor coil and fan unit.

**Water System Description**

A nominal 120 gallon (454 liters) hot water tank designed for solar water heating systems was used to maintain a constant water temperature source for the active desuperheaters in the laboratory testing. (The actual capacity of this tank was measured at 106.5 gallons (403 L).) A 4500 Watt heating element was installed in the bottom of the tank and was used to keep the entire tank at the desired temperature.

In order to test the passive insert desuperheater, a second nominal 120 gallon solar storage tank was located downstream of the constant water temperature source tank, as shown in Figure B2. The insert desuperheater was installed vertically in the middle of this second tank. To measure the water temperature inside the tank, a "thermocouple
tree" was also inserted vertically into one of the extra solar ports. The thermocouple tree spacing divided the tank into 6 equal volumes with a thermocouple located at the middle of each volume.

**Testing Procedure**

The active desuperheater units were tested according to the procedures specified in the Air-Conditioning and Refrigeration Institute (ARI) Standard 470 for desuperheater water heaters.[1987] The ARI 470-87 standard rating conditions are listed in Table 1 in Chapter 1. However, entering water temperatures between 90 F and 120 F (32.2-48.9°C), as well as the Florida modification of 80 F (26.7°C), were also used in the laboratory testing.

For some of the tests, the water flow rate was externally controlled so that the temperature of the water leaving the desuperheater was 140 F (60°C) as specified in ARI
Standard 470. However, for most of the tests, the leaving water temperature was not controlled in order that the desuperheater could operate under its regular water flow condition.

ARI Standard 470-87 specifies a minimum test period of 15 minutes after establishment of steady refrigerant flow conditions. Therefore, prior to the beginning of each test, the heat pump was started and allowed to reach a steady-state operating condition. For the active desuperheaters, the electrical circuit to the water circulating pump was then enabled and the test was run for an interval of 16 minutes.

The testing of the passive insert desuperheater, however, required a slightly different testing methodology. Prior to the beginning of the test, the heat pump was operated and allowed to reach a steady refrigerant flow condition. At the same time, the water in the tank containing the insert desuperheater was heated with the bottom electric element to the desired testing temperature. However, because the insert desuperheater was also heating the water in the tank during this period, the heat pump operation was stopped after 10 minutes. The heating of the tank was then continued with the electric element only until the desired initial tank temperature was reached.

As soon as the tank containing the insert desuperheater reached the desired temperature, the heat pump was started and, after a 1 minute interval, the test was begun. Due to the prior heat pump operation, the refrigerant pressures and superheat temperatures required by ARI Standard 470 were approached within 6 percent after the first minute of the test and were within 3 percent after the second minute.

After a total of 17 minutes of heat pump operation, the insert desuperheater test was stopped. (In order to be consistent with the active desuperheater test procedure, only
the last 16 minutes of data were included for the determination of the insert refrigerant temperatures and flow.) Then the water in the insert tank was drawn off at a rate of 3 gallons per minute (0.2 L/s) with a supply water temperature equal to the initial tank temperature. Inlet and outlet water temperatures were measured in order to determine the energy content of the tank after the test was concluded.

For testing of both the active and passive desuperheaters in heating mode, the heat pump was allowed to operate just as it would if switching from cooling mode in normal residential operation. At the ARI Standard 470 refrigerant superheat temperature of 180 F (82.2°C), a steady refrigerant pressure of 245 psig (1690 kPa) was measured on the discharge side of the compressor. This condition raised the refrigerant saturation temperature to 115.7 F (46.5°C) over the 105 F (40.6°C) specified in Standard 470. (The compressor discharge temperature was consistently 100 F (56°C) above the entering water temperature to the water-source heat exchanger, indicating proper refrigerant charge for the heating mode.) The indoor chamber condensing air temperature ranged between 67 F and 74 F (19.4-23.3°C) which was typical of residential conditions where a heat pump would cycle on and off to keep the living space comfortable.

Instrumentation and Data Reduction

In order to measure the water temperatures, insertion-type temperature sensors were installed in the entering and leaving water lines to the active desuperheater, as indicated in Figure B1. A positive displacement flowmeter measured the volume of water flow through the desuperheater with a resolution of 0.005 gallon (0.019 L).
In order to measure the refrigerant temperatures, insulated temperature sensors were attached to the surface of the desuperheater inlet and outlet refrigerant lines. Refrigerant gauges were also used to monitor the pressure of the suction and discharge sides of the compressor. A refrigerant flow meter with a resolution of 0.0064 lbm (0.003 kg) was installed in the liquid line between the water-source heat exchanger and the indoor fan coil unit. Additional temperature sensors were also installed to measure the liquid line subcooling and suction line superheat temperatures.

The temperature sensors and flow meters were connected to the laboratory data acquisition system. A data sample rate of 10 seconds was used in the testing for all but one desuperheater unit. A slightly faster sample rate of 8 seconds was used to measure the performance of the desuperheater with the outlet temperature control valve. At the end of each minute for all the tests, the temperature data was averaged and the fluid mass flow was totalized by the data acquisition system. Following the 16 minute test, the one minute data was transferred to a personal computer and processed using a BASIC program. To determine the available superheat, the enthalpies for R-22 were determined from the refrigerant pressure and temperature conditions. To calculate the refrigerant heat exchanged, the enthalpy difference between the inlet and outlet lines to the desuperheater was multiplied by the total amount of refrigerant mass flow during each minute. The heat exchanged on the water side was determined by multiplying the water temperature differential by the mass of water flow recorded during each minute. Finally, totals for the 16 minute test were calculated and average temperature and flow conditions were determined. A number of tests were repeated at the same conditions before a test at a new entering water temperature was conducted.
APPENDIX C

TRNSYS INPUT FILE
Active Desuperheater A2
ASSIGN DHP207YR.LST 6
ASSIGN WEATHER\ORLANDO\ORLANDO.TMY 10
ASSIGN INPUT\38YK036.ACS 11
ASSIGN INPUT\38YK036.HPC 12
ASSIGN DHP207YR.PLT 13

SIMULATION 1. 8759. .1
LIMITS 100 50
WIDTH 72
NOLIST

EQUATIONS 71
* Radiation Parameters
JULDAY = 1
LAT = 28.5
LONG = 81.4
STDMER = 75.0
SHFT = STDMER - LONG
GRDRHO = 0.2
AZIMUT = 0.0
* House Parameters
UAHOME = 1899
LUMCAP = 18227
CPAIR = 1.012
ECMIN = 2795
LHR = 0.286
* Roof Parameters
ALPHA = 0.85
EMIT = 0.91
AROOSFS = 75.5
AROOFE = 0.0
AROOFN = 75.5
AROOFW = 0.0
ACEIL = 139.4
INFIL = 0.0
TILTS = 22.6
TILTE = 90.0
TILTN = 22.6
TILTW = 90.0
* Window Parameters
AWINDW = 10.5
UWINDW = 14.4
TAU = 0.6
* Thermostat Parameters
TCOOL = 25.0
THEAT1 = 20.0
THEAT2 = 20.0
TSETBK = 7.2
TDEADB = 1.1

* Storage Tank Parameters
VOLUME = 0.151
HEIGHT = 1.27
PI = 3.14159
DIAMTR = (4 * VOLUME/(PI * HEIGHT))^0.5
TKAREA = PI * DIAMTR * (HEIGHT + DIAMTR/2)
UTANK = 2.56
Q AUX1 = 16200
TSET1 = 55.0
TDEAD1 = 12.2
Q AUX2 = 16200
TSET2 = 45.0
TDEAD2 = 2.8
H2O RHO = 1000
CPTANK = 4.19
NNODES = 3
HN = HEIGHT/NNODES

* Desuperheater A2 Parameters
UAHX = 238
CPR22 = 0.88
MDOT22 = 245

* Pump Parameters
MDOTMX = 385
POWMAX = 313

* Controller Parameters
THIGHL = 65.6
TURNON = 54.4
TURNOF = 50.0
TZERO = 0.0
TRESET = 65.0

* Pipe Parameters
PIPEID = 0.0127
PIPEOD = 0.0159
LENGTH = 7.62
TINSUL = 0.0102
KINSUL = 0.135
UPipe = KINSUL/LN((PIPEOD + (2*TINSUL))/PIPEOD)

* Temperature Parameters
TAMB = 25.0
TENV = 23.8
TCOLD = 22.2
THOT = 50.0
TAIRH = 35.0
TAIRC = 13.0
TBOIL = 100.0

UNIT 9 TYPE 9 CARD READER
PARAMETERS 16
*m n hour direct-normal global-horiz dry-bulb T
*wind-speed logical unit
2 7 1.0 -3 1.0 0.0 -4 1.0 0.0 5 0.1 0.0 7 1.0 0.0
10
*OUTPUTS 7 + 2
*month hour direct-normal global-horiz Tdry-bulb Wratio
*wind-speed tstart tend

UNIT 16 TYPE 16 SOLAR RADIATION PROCESSOR
PARAMETERS 8
*m fix Perez Julian lat SolCon SHFT SMOOTH
7 1 4 JULDAY LAT 4871. SHFT 2
INPUTS 13
*Ih Idn tlast tnext ground tilt-s azimuth-s tilt-e azi-e
*tilt-n azi-n tilt-w azi-w
9,4 9,3 9,19 9,20 0,0 0,0 0,0 0,0 0,0 0,0
180.0 TILT 90.0
*OUTPUTS 19
*Io zenith azi-sun Ihorz Ihdiff Itot1 Ibeam1 Idif1 thetal
tilt1
*Itot2 Ibeam2 theta2 Itot3 Tbeam3 theta3 Itot4 Ibeam4 theta4

UNIT 18 TYPE 18 PITCHED ROOF AND ATTIC
PARAMETERS 13
*Nroof Nins alpha emit Asouth Aeast Anorth Awest Aceil
*Infil Ucol tilt-s tilt-n
-1 3 ALPHA EMIT AROOF S AROOF E AROOF N AROOF W ACEIL INFIL
0 TILTS TILTN
INPUTS 8
*Tamb Isouth Ieast Inorth Iwest winds Tcol Troom
9,5 16,6 16,11 16,14 16,17 9,7 0,0 12,4
TAMB 0.0 0.0 0.0 0.0 0.0 0.0 0.0 TENV
*OUTPUTS 2
*Qceil Tsol-air

UNIT 35 TYPE 35 EAST WINDOW
PARAMETERS 2
*m area
1 AWINDW
INPUTS 5
*roomT Tamb Ufactor Ieast tau
12,4 9,5 0,0 16,11 0,0
TENV TAMB UWINDW 0.0 TAU
*OUTPUTS 3
*Qtotal Qsolar Qthermal

UNIT 36 TYPE 35 WEST WINDOW
PARAMETERS 2
*m area
1 AWINDW
INPUTS 5
*roomT Tamb Ufactor Iwest tau
12,4 9,5 0,0 16,17 0,0
TENV TAMB UWINDW 0.0 TAU
*OUTPUTS 3
*Qtotal Qsolar Qthermal

EQUATIONS 2
*Window and Roof Solar Gains
WGAIN = [35,2] + [36,2]
HGAIN = WGAIN + [18,1]

UNIT 12 TYPE 12 DEGREE-HOUR HOUSE LOAD
PARAMETERS 7
*m UAhous Cap roomTi fluidCp eCmin LHR
4 UAHOME LUMCAP TENV CPAIR EDMIN LHR
INPUTS 6
*Tsup mdotsup Tamb Qgain Qheat Qcool
0,0 0,0 9,5 HGAIN 43,1 42,1
TAIRH 0,0 TAMB 0,0 0,0 0,0
*OUTPUTS 8
*Treturn mdotret Qhload roomTavg Qhx Qheat Qsensible Qlatent

UNIT 8 TYPE 8 HOUSE THERMOSTAT
PARAMETERS 8
*nstk 1stg Tmin Tcool Theat1 Theat2 Tsetback Tdeadband
3 0 99 TCOOL THEAT1 THEAT2 TSETBK TDEADB
INPUTS 3
*roomT 1stageT fsetback
12,4 0,0 15,1
TENV -99 1,0
*OUTPUTS 3
*fcontrol-stage1heat fcntrl-stg2heat fcntrl-cool

UNIT 15 TYPE 14 HEATING SETBACK PROFILE
PARAMETERS 12
*time0 value0 t1 v1 ... ti vi ... timelast vlast
0 1.0 6 1.0 6 0.0 23 0.0 23 1.0 24 1.0
*OUTPUTS 1
*averagevalue

UNIT 42 TYPE 42 CONDITIONING EQUIPMENT (AIR CONDITIONER)
PARAMETERS 4
*lu Nx Ny Nxl
11 1 4 4
INPUTS 2
*fcontrol Tamb
8,3 9,5
0 TAMB
*OUTPUTS 4
*Qcool Qsensible Power Tdischarge

UNIT 43 TYPE 42 CONDITIONING EQUIPMENT (HEAT PUMP)
PARAMETERS 4
*lu Nx Ny Nx1
12 1 4 8
INPUTS 2
*fcontrol Tamb
8 2 9 5
0 TAMB
*OUTPUTS 4
*Qheat Power COP Tdischarge

UNIT 4 TYPE 4 TANK
PARAMETERS 22
*m volume Cpf fluidrho Utnk h1 h2 h3 auxm nq1 nt1 tset1
*tdead1 qauxmax1 nq2 nt2 tset2 tdead2 qauxmax2 flueUA
*flueTavg Tboil
1 VOLUME CPTANK H2ORHO UTANK HN HN HN 1 1 2 TSET1
TDEAD1 QAUXX 3 3 TSET2 TDEAD2 QAUXX 0 0
TENV TBOIL
INPUTS 5
*Thot mdoth Tcold mdotload Tenv
31 1 31 2 0 0 41 1 0 0
TENV 0 0 TCOLD 0 0 TENV
DERIVATIVES 3
*Tnode1 Tnode2 Tnode3
TSET1 TSET1 TCOLD
*OUTPUTS 12
*Tn mdot-c Tload mdot-l qdotenv qdot-l deltaU qaux qaux1
*aux2 qdotin Tavg

EQUATIONS 1
*Compressor Parameters
TSUPHT = ([42 4] * [8 3]) + ([8 2] * [43 4])

UNIT 2 TYPE 2 PUMP CONTROLLER
PARAMETERS 4
*nstk deltha1 deltalo Tmax Treset
3 TURNON TURNOFF THIGHL TRESSET
INPUTS 4
*Thi Tlow Thilimit fcontrol-in
TSUPHT 0 0 5 3 2 1
TAMB TZERO TAMB 0 0
*OUTPUTS 1
*fcontrol-out

EQUATIONS 3
*Pump Control
COOLCTRL = AND([2 1],[8 3])
HEATCTRL = AND([2 1],[8 2])
CONTRL = OR (COOLCTRL,HEATCTRL)

UNIT 3 TYPE 3 PUMP
PARAMETERS 4
*mdotmax cpfuid powermax fconvert
MDOTMX CPTANK POWMAX 0
INPUTS 3
*Tin mdotin fcontrol
4,1 4,2 CONTRL
TENV 0.0 0.0
*OUTPUTS 3
*Tout mdotout Power-use

UNIT 30 TYPE 31 UNIT SUPPLY PIPING
PARAMETERS 6
*pipeid run Upipe fluidrho Cpf Tstart
PIPEID LENGTH UPIPE H2ORHO CPTANK TENV
INPUTS 3
*Tin mdot Tenv
3,1 3,2 0,0
TENV 0.0 TENV
*OUTPUTS 6
*Tout mdot qdotloss deltai deltaU Tavg

EQUATIONS 1
*Compressor Parameters
FLOW22 = CONTRL * MDOT22

UNIT 5 TYPE 5 HEAT EXCHANGER
PARAMETERS 4
*m UA Cphot Cpcold
2 UAHX CPR22 CPTANK
INPUTS 4
*hotTin mdothot coldTin mdotcold
TSUPHT FLOW22 30,1 30,2
TAMB MDOT22 0.0 0.0
*OUTPUTS 6
*hotTout mdothot coldTout mdotcold Qdothx ehx

UNIT 31 TYPE 31 UNIT RETURN PIPING
PARAMETERS 6
*pipeid run Upipe fluidrho Cpf Tstart
PIPEID LENGTH UPIPE H2ORHO CPTANK TAMB
INPUTS 3
*Tin mdot Tenv
5,3 5,4 0,0
TAMB 0.0 TENV
*OUTPUTS 6
*Tout mdot qdotloss deltai deltaU Tavg

UNIT 13 TYPE 14 WEEKDAY HOT WATER LOAD
PARAMETERS 80
*time0 value0 t1 v1 ... ti vi ... timelast vlast
0 0.0 5 0.0 5 4.9 6 4.9 6 25.8 7 25.8 7 19.6 8 19.6
8 13.1 9 13.1 9 9.8 10 9.8 10 12.3 11 12.3 11 9.8 12 9.8
12 6.1 13 6.1 13 6.1 14 6.1 14 7.4 15 7.4 15 8.6 16 8.6 16
16 11.0 17 11.0 17 9.8 18 9.8 18 19.6 19 19.6 19 20.0 20 20.0
20 18.4 21 18.4 21 16.0 22 16.0 22 18.8 23 18.8 23 6.1 24
6.1
*OUTPUTS 1
*average value

UNIT 14 TYPE 14 WEEKEND HOT WATER LOAD
PARAMETERS 76
*time0 value0 t1 v1 ... ti vi ... timelast vlast
0 0.0 6 0.0 6 1.1 7 1.1 7 9.6
8 9.6 8 24.5 9 24.5 9 24.1 10 24.1 10 20.7 11 20.7 11 15.7
12 15.7 12 10.3 13 10.3 13 10.3 14 10.3 14 13.8 15 13.8 15
8.0 16 8.0 16 10.3 17 10.3 17 10.3 18 10.3 18 13.4 19 13.4
19 17.2 20 17.2 20 17.6 21 17.6 21 18.2 22 18.2 22 10.7 23
10.7 23 7.3 24 7.3
*OUTPUTS 1
*average value

UNIT 41 TYPE 41 LOAD PROFILE SEQUENCER
PARAMETERS 19
*Nv day1 dy2 dy3 dy4 dy5 dy6 dy7 Nhday hd hd1 hd2 hd3 hd4 *hd5
hd6 hd7 hd8 hd9
1 2 1 1 1 1 2 9 2 1 46 151 185 249
284 315 329 359
INPUTS 2
*profile1 profile2
13.1 14.1
0.0 0.0
*OUTPUTS Nv
*profile inputs

EQUATIONS 11
*kJ/hour to kW
KWUNIT = [4,11]/3600
DELTAU = [4,7]/3600
KWLOSS = ([4,5] + [30,3] + [31,3])/3600
KWLOAD = [4,6]/3600
KWAUX = [4,8]/3600
KWPUMP = [3,3]/3600
KWTOT = KWAUX + KWPUMP
*daily averages for histograms
DKWUNIT = KWUNIT/365
DKWPMP = KWPUMP/365
DKWAUX = KWAUX/365
DKWTOT = KWTOT/365

EQUATIONS 10
*kJ/hour to kW
QSENSL = [12,7]/3600
QLATL = [12,8]/3600
QCOOL = [42,1]/3600
QSENSA = [42,2]/3600
KWAC = [42,3]/3600
KWROOF = [18,1]/3600
*daily averages for histograms
DQSENS = QSENSL/365
\[ DQCOOL = \frac{QCOOL}{365} \]
\[ DKWAC = \frac{KWAC}{365} \]
\[ DQROOF = \frac{KWROOF}{365} \]

**EQUATIONS**

*kJ/hour to kW*

\[ QHEAT = \frac{[43,1]}{3600} \]
\[ KWHP = \frac{[43,2]}{3600} \]

*daily averages for histograms*

\[ DQHEAT = \frac{QHEAT}{365} \]
\[ DKWHP = \frac{KWHP}{365} \]

**UNIT 28 TYPE 28 SIMULATION SUMMARY (TANK)**

**LABELS**

**KWUNIT TLOAD TTANK DELTAU KWLOSS KWLOAD KWHAUX KWHPUMP KWHTOT EF**

**INPUTS**

*delU kWunit Tload Ttank kWloss kWload kWaux kWpump kWtot*

**PARAMETERS**

*hr ton toff lu mode*

*24 1 8760 0 2*

*Stack operations*

*1 0*
*0 -4*
*0 -2 2 -4*
*0 -2 2 -4*
*4 -4 0 -4 0 -4*
*0 -4 0 -4*
*4 -16 -19 2 -4*

*Energy balance*

*CHECK .2 1 7 -6 -5 -4*

**UNIT 29 TYPE 28 SIMULATION SUMMARY (HOUSE)**

**LABELS**

**TAMB TROOM QCOOL QSENSL QLATL KWHAC QHEAT KWHP**

**INPUTS**

*Tamb Troom Qcool Qsens Qlat kWhac Qheat kWhp*

**PARAMETERS**

*hr ton toff lu mode*

*24 1 8760 0 2*

*Stack operations*

*0 -2 2 -4*
*0 -2 2 -4*
*0 -4 0 -4*
*0 -4 0 -4*
*0 -4 0 -4*

UNIT 22 TYPE 28 MONTHLY SIMULATION SUMMARY (TANK)

LABELS 10
KWUNIT TLOAD TTANK DELTAU KWLOSS KWLOAD KWAUX KWHPUMP KWHTOT EF
INPUTS 9
*delU kWunit Tload Ttank kWloss kWload kWaux kWpump kWtot
DELTAU KWUNIT 4,3 4,12 KWLOSS KWLOAD KWAUX KWHPUMP KWTOT
PARAMETERS 32
*hr ton toff lu mode
-1 1 8760 0 2
*Stack operations
1 0
0 -4
0 -2 2 -4
0 -2 2 -4
-4
0 -4 0 -4 0 -4
0 -4 0 -4
-16 -19 2 -4
*Energy balance
CHECK .2 1 7 -6 -5 -4

UNIT 23 TYPE 28 MONTHLY SIMULATION SUMMARY (HOUSE)
LABELS 8
TAMB TROOM QCOOL QSENSL QLATL KWHAC QHEAT KWHP
INPUTS 8
*Tamb Troom Qcool Qsens Qlat kWac Qheat kWhp
9,5 12,4 QCOOL QSENSL QLATL KWAC QHEAT KWHP
PARAMETERS 25
*hr ton toff lu mode
-1 1 8760 0 2
*Stack operations
0 -2 2 -4
0 -2 2 -4
0 -4 0 -4
0 -4 0 -4
0 -4 0 -4

UNIT 26 TYPE 27 HISTOGRAM PLOTTER
PARAMETERS 8
*m intprn ireset ton toff startrange endrange interval
2 8760 8760 1 8760 0 24 24
INPUTS 10
DQROOF DQSENS DQCOOL DKWAC DQHEAT DKWHP DKWUNT DKWPMP DKWAUX DKWTOT
KWROOF QSENSL QCOOL KWHAC QHEAT KWHP KWUNIT KWUNIT KWUNIT KWUNIT

UNIT 27 TYPE 27 HISTOGRAM PLOTTER
PARAMETERS 8
*m intprn ireset ton toff startrange endrange intervals
1 8760 8760 1 8760 0.01 10.01 10
INPUTS 3
KWAC KWHP KWUNIT
CLH HLH RUNHRS
UNIT 25 TYPE 25 HOURLY PRINTER
PARAMETERS 5
*intprn ton toff lu munits
1 1 8760 13 2
INPUTS 9
12,4 QCOOL QSENSL QLATL KWAC KWUNIT KWPUMP KWAUX KWTOT
TROOM QCOOL QSENSL QLATL KWAC KWUNIT KWPUMP KWAUX KWTOT

UNIT 65 TYPE 65 ONLINE PLOTTER
PARAMETERS 14
*ntop nbot ymin1 ymax1 ymin2 ymax2 iref iupd units npic *xgrid stop msym mout
4 4 0 75 0 5 1 1 3 52
7 0 1 1
INPUTS 8
4,3 31,1 12,4 9,5 KWAUX KWAC KWHP KWPUMP
TLOAD TUNIT TROOM TAMBC KWAUX KWAC KWHP KWPUMP
LABELS 4
C kW Temperatures Electrical Demand
END
APPENDIX D

MODIFIED TRNSYS SOURCE CODE
Stratified Hot Water Storage Tank
with Insert Heat Exchanger
SUBROUTINE TYPE95(TIME,XIN,OUT,T,DRTT,PAR,INFO,ICNTRL,*)

C******************************************************************************C
THIS ROUTINE SIMULATES A STRATIFIED STORAGE TANK
C CONSISTING OF NEQ FULLY MIXED SECTIONS, THE VALUE OF NEQ
C BEING CHOSEN BY THE USER.
C REFERENCE 'EXPERIMENTAL AND SIMULATED PERFORMANCE OF A
C CLOSED LOOP SOLAR WATER HEATING SYSTEM', BY P.I.COOPER,
C S.A.KLEIN, AND C.W.S.DIXON PRESENTED AT THE ISES MEETING
C IN AUGUST,1975
C
C TEMPERATURE INVERSION FIX ADDED IN AUGUST 1992 -- JWT
C SPECIFIED INLET MODE ADDED (MODE 3) - 10/92 -- JWT
C THERMAL SHORT MODE ADDED - 10/92 -- JWT
C THERMAL SHORT FIX (END CAP) - 2/93 -- JWT
C 1-NODE QLOSS FIX - 2/93 -- JWT
C TANK RELIEF VALVE ADDED - 3/93 -- JWT
C 2ND ELEMENT AND T-STAT ADDED -- JWT
C
C TO MODEL A VERTICAL INSERT HX, FLUE TEMP CHANGED TO AN
C INPUT, CMIN ADDED AS A PARAMETER, AND UAFT VARIABLE
C CHANGED TO THE PRODUCT OF CMIN AND THE HX EFFECTIVENESS.
C HX EFFECTIVENESS = (1 - EXP(-UAFT/CMIN)) - 2/95 -- TIM
C******************************************************************************C

DOUBLE PRECISION XIN,OUT
INTEGER FINAL,DIRECT,AVG,OLD,LOC(2),LOCT(2),AUXMOD
REAL MSCP,MCPN,TSET(2),TDB(2),QBTOT(2),QBOOST(2),
QHE(2)
LOGICAL TMODE,HMODE,UMODE,HTRON(2),LOOPED
CHARACTER*1 TRNEDT,PERCOM,HEADER,PRTLAB,LRNCHK,PRUNIT,
IOCHECK,PRWARN
CHARACTER*3 YCHECK(8),OCHECK(25)

INTEGER*4 INFO
DIMENSION T(15),DRTT(15),XIN(10),OUT(35),PAR(51),
INFO(15)
DIMENSION U(16),V(16),QB(16,2),H(15),USHORT(15)
COMMON/SIM/ TIMEO,TFINAL,DELT,IWARN
COMMON/STORE/ NSTORE,IAV,S(5000)
COMMON/LUNIT/ LUR,LUW,IFORM,LUK
COMMON/CONFIG/ TRNEDT,PERCOM,HEADER,PRTLAB,LRNCHK,
PRUNIT,IOCHECK,PRWARN
DATA IUNIT/0/,NSTK/15/,QB/32*0.0/,PI/3.1415193/,
LOCT/2*0/

C
LOOPED=.FALSE.
DO 21 JJ=1,NEQ
QB(J,1)=0.
QB(J,2)=0.
21 CONTINUE
IF (INFO(7).GE.0) GO TO 10
FIRST CALL OF SIMULATION
NI = 6
IF (INFO(3) .EQ. 7) NI = 7
IF (INFO(3) .EQ. 8) NI = 8
NEQ=INFO(5)
INFO(6)=11+MAX(1,(NEQ-2))

SET UP THE INPUT AND OUTPUT VARIABLE TYPES
DATA YCHECK/'TE1','MF1','TE1','MF1','TE1','TE1','CF1',
1 'CF1'/
DATA OCHECK/'TE1','MF1','TE1','MF1','PW1','PW1','EN1',
1 'PW1','PW1','PW1','TE1','TE1',
1 'TE1','TE1','TE1','TE1','TE1','TE1','TE1',
CALL RCHECK(INFO,YCHECK,OCHECK)

IF (NEQ.LT.1 .OR. NEQ.GT.15) CALL TYPECK(5,INFO,0,0,0)
INFO(10) = NEQ*4+4
TMODE=.FALSE.
UMODE=.FALSE.
HMODE=.FALSE.

IF(INT(PAR(1)+0.1).EQ.3) TMODE=.TRUE.
IF(PAR(5).LT.0) UMODE=.TRUE.
IF(PAR(6).GT.0) HMODE=.TRUE.

ITANK=0
IU=0
IND=1
IF(HMODE) IND=NEQ
IF(TMODE) ITANK=2
IF(UMODE) IU=NEQ

NP=19+IND+ITANK+IU
CALL TYPECK(1,INFO,NI,NP,NEQ)

FIND AVERAGE TEMPERATURE AND SET INITIAL TEMPERATURES
IS = INFO(10) + NEQ - 1
IAVG = INFO(10) + NEQ*2 - 1
IOLD = INFO(10) + NEQ*3 - 1
TI = 0.
HIGH = 0.
DO 5 J = 1,NEQ
IS = IS + 1
IAVG = IAVG + 1
IOLD = IOLD + 1
S(IS) = T(J)
S(IAVG) = T(J)
S(IOLD) = T(J)
HN=ABS(PAR(6))/FLOAT(NEQ)
IF(PAR(6).LT.0.) GO TO 4
HN = PAR(5+J)
4
HIGH = HIGH + HN
5
TI = TI + T(J)*HN
TI = TI/HIGH
IS = INFO(10) + NEQ*4-1
S(IS+1) = TI

C IF UNIT HAS CHANGED, SET THE PARAMETER LIST AGAIN
10 IF (INFO(1).EQ.IUNIT) GO TO 30

IUNIT = INFO(1)
NEQ = INFO(5)
XNEQ = FLOAT(NEQ)

C SET THE PARAMETER MODES
TMODE = .FALSE.
UMODE = .FALSE.
HMODE = .FALSE.

IF (INT(PAR(1)+0.1).EQ.3) TMODE = .TRUE.
IF (PAR(5).LT.0) UMODE = .TRUE.
IF (PAR(6).GT.0) HMODE = .TRUE.

C SET THE PARAMETER IDENTIFIERS
ITANK = 0
IU = 0
IND = 1

IF (HMODE) IND = NEQ
IF (TMODE) ITANK = 2
IF (UMODE) IU = NEQ

C DETERMINE THE TOTAL # OF PARAMETERS
LAST = 19 + IND + ITANK + IU

C GET THE PARAMETERS
C TANK PARAMETERS
MODE = INT(PAR(1)+0.1)
VOL = PAR(2)
CPF = PAR(3)
RHOF = PAR(4)
ULOSS = ABS(PAR(5))
AUXMOD = INT(PAR(6+IND)+0.1)

C 1ST ELEMENT PARAMETERS
LOC(1) = INT(PAR(7+IND)+0.1)
LOCT(1) = INT(PAR(8+IND)+0.1)
TSET(1) = PAR(9+IND)
TDB(1) = PAR(10+IND)
QHE(1) = PAR(11+IND)

C 2ND ELEMENT PARAMETERS
LOC(2) = INT(PAR(12+IND)+0.1)
LOCT(2) = INT(PAR(13+IND)+0.1)
TSET(2) = PAR(14+IND)
TDB(2) = PAR(15+IND)
QHE(2) = PAR(16+IND)

UAF = PAR(17+IND)
CMIN = PAR(18+IND)
TBOIL = PAR(19+IND)

C CHECK ON THE PARAMETERS
IF((AUXMOD.LT.1).OR.(AUXMOD.GT.2))
   CALL TYPECK(4,INFO,O,NP,O)
IF((LOC(1).LT.1).OR.(LOC(1).GT.NEQ))
   CALL TYPECK(4,INFO,O,NP,O)
IF((LOC(2).LT.1).OR.(LOC(2).GT.NEQ))
   CALL TYPECK(4,INFO,O,NP,O)
IF((LOCT(1).LT.1).OR.(LOCT(1).GT.NEQ))
   CALL TYPECK(4,INFO,O,NP,O)
IF((LOCT(2).LT.1).OR.(LOCT(2).GT.NEQ))
   CALL TYPECK(4,INFO,O,NP,O)
IF(CMIN.LE.0.) CALL TYPECK(4,INFO,0,NP,0)
IF(QHE(1).LT.0.) QHE(1)=0.
IF(QHE(2).LT.0.) QHE(2)=0.
IF(UAF.LT.0.) UAF=0.

C SET THE ADDITIONAL U PARAMETERS
DO 7 I=1,NEQ
   IF(UMODE) THEN
      USHORT(I) = PAR(LAST-IU+I)
   ELSE
      USHORT(I) = 0.0
   ENDIF
C CHECK TO BE SURE THAT TOTAL LOSS COEFFICIENT IS > 0
IF((ULOSS+USHORT(I)).LT.0) USHORT(I)=-ULOSS
7 CONTINUE

C SET THE COLLECTOR AND LOAD SIDE FLOW ENTRANCES
IF(TMODE) THEN
   INCOLL=INT(PAR(LAST-IU-ITANK+1)+0.1)
   INLOAD=INT(PAR(LAST-IU-ITANK+2)+0.1)
   IF((INCOLL.LT.1).OR.(INCOLL.GT.NEQ)) THEN
      CALL TYPECK(4,INFO,NI,NP,ND)
   ENDIF
   IF((INLOAD.LT.1).OR.(INLOAD.GT.NEQ)) THEN
      CALL TYPECK(4,INFO,NI,NP,ND)
   ENDIF
ENDIF

DO 71 II=1,2
   IF(TSET(II).GT.TBOIL) THEN
      TSET(II) = TBOIL
   IF(INFO(7).EQ.-1) THEN
      IF((PRWARN.EQ.'Y').OR.(PRWARN.EQ.'y')) THEN
         WRITE(LUW,1001) INFO(1)
ENDIF
IWARN=IWARN+1
ENDIF

END IF
71 CONTINUE

HNODE=ABS(PAR(6))/XNEQ
HIGH=0.
XAUX=0.
LOCFIN=MIN(LOC(1),LOC(2))

DO 27 J=1,NEQ
   H(J)=HNODE
   IF(HMODE) H(J)=PAR(5+J)
   IF(J.GT.LOCFIN) GO TO 27
   XAUX=XAUX+H(J)
27 HIGH=HIGH+H(J)

INIT = INFO(10) - 1
FINAL = INIT + NEQ
AVG = FINAL + NEQ
OLD = AVG + NEQ
IS=OLD+NEQ
MSCP=VOL*RHOF*CPF
UAETOP=(ULOSS+USHORT(1))*VOL/HIGH
UAEBOT=(ULOSS+USHORT(NEQ))*VOL/HIGH
PER=(4.*PI*VOL/HIGH)**0.5

30 CONTINUE

C SET INPUTS
TIN=XIN(1)
FLWSS=XIN(2)
TL=XIN(3)
FLWLL=XIN(4)
TENV=XIN(5)
TFLUE=XIN(6)
QBOOST(1)=QHE(1)
QBOOST(2)=QHE(2)
IF (INFO(3) .GT. 5) QBOOST(1) = QHE(1)*XIN(7)
IF (INFO(3) .GT. 6) QBOOST(2) = QHE(2)*XIN(8)
FLWS=FLWSS*CPF
FLWL=FLWLL*CPF
QVENT=0.

C ON THE FIRST CALL OF EACH TIME STEP, SAVE INITIAL VALUES
C OF NODE TEMPERATURES. THESE ARE THE FINAL VALUES FROM
C THE PREVIOUS CALL.
IF (INFO(7) .GT. 0) GO TO 40
37 DO 35 K = 1,NEQ
   KI = INIT + K
   KF = FINAL + K
35 \( S(KI) = S(KF) \)
\( \text{HTRON}(1) = \text{.FALSE.} \)
\( \text{HTRON}(2) = \text{.FALSE.} \)
\( \text{IF}(\text{OUT}(9) \geq 0 \land S(\text{LOCT}(1)+\text{INIT}) < \text{TSET}(1) - 0.01) \)
\( \text{HTRON}(1) = \text{.TRUE.} \)
\( \text{IF}(\text{OUT}(10) \geq 0 \land S(\text{LOCT}(2)+\text{INIT}) < \text{TSET}(2) - 0.01) \)
\( \text{HTRON}(2) = \text{.TRUE.} \)
40 \( \text{IF}(\text{NEQ} \neq 1) \) GO TO 200

C DETERMINE WHICH NODES RECEIVE ENTERING FLOWSTREAMS.
C EACH FLOWSTREAM GOES TO THE NODE WHICH IS CLOSEST IN
C TEMPERATURE.

\( \text{LOCFIN} = \text{MIN}(\text{LOC}(1), \text{LOC}(2)) \)

C SKIP AHEAD IF MODE 2
\( \text{IF}(\text{MODE} \neq 2) \) GO TO 45

C INLET LOCATION IS JUST BELOW TOP HEATER ELEMENT
\( \text{NCOLL} = \text{MIN}(\text{LOCFIN}+1, \text{NEQ}) \)
\( \text{NLOAD} = \text{NEQ} \)
\( \text{IF}(\text{TMODE}) \) THEN
\( \text{NCOLL} = \text{INCOLL} \)
\( \text{NLOAD} = \text{INLOAD} \)
ENDIF

GO TO 65

45 \( \text{IF}(\text{INFO}(7) \geq \text{NSTK}) \) GO TO 60
\( \text{D1} = \text{ABS}(S(\text{AVG}+1) - \text{TIN}) \)
\( \text{NCOLL} = 1 \)
\( \text{D2} = \text{ABS}(S(\text{AVG}+1) - \text{TL}) \)
\( \text{NLOAD} = 1 \)
\( \text{DO } 52 \ J = 2, \text{NEQ} \)
\( K = \text{AVG} + J \)
\( \text{IF}(\text{ABS}(S(K) - \text{TIN}) \geq \text{D1}) \) GO TO 50
\( \text{NCOLL} = J \)
\( \text{D1} = \text{ABS}(S(K) - \text{TIN}) \)
50 CONTINUE
\( \text{IF}(\text{ABS}(S(K) - \text{TL}) \geq \text{D2}) \) GO TO 52
\( \text{NLOAD} = J \)
\( \text{D2} = \text{ABS}(S(K) - \text{TL}) \)
52 CONTINUE
\( \text{IF}(\text{TIN} \geq \text{S(\text{AVG}+1)}) \) \( \text{NCOLL} = 1 \)
\( K = \text{AVG} + \text{NEQ} \)
\( \text{IF}(\text{TL} \geq \text{S(K)}) \) \( \text{NLOAD} = \text{NEQ} \)
\( S(\text{IS}+2) = \text{NCOLL} \)
\( S(\text{IS}+3) = \text{NLOAD} \)
GO TO 65
60 \( \text{NCOLL} = S(\text{IS}+2) \)
\( \text{NLOAD} = S(\text{IS}+3) \)
CALCULATE TEMPERATURES OF EACH NODE. THE FINAL AND AVERAGE TEMPERATURE OF NODE K IS STORED IN S(FINAL+K) AND S(AVG+K). HEAT EXCHANGE BETWEEN NODES IS BASED ON THE AVERAGE NODE TEMPERATURES OVER THE TIMESTEP. TEMPERATURES ARE EVALUATED STARTING WITH THE NODE WHICH HAS THE LARGEST ENTERING FLOWSTREAM FLOW RATE.

65

\[
\text{NSTART} = \text{NCOLL}
\]

\[
\text{IF} (\text{FLWL} .GT. \text{FLWS}) \text{NSTART} = \text{NLOAD}
\]

\[
\text{NODE} = \text{NSTART}
\]

\[
\text{DIRECT} = 1
\]

\[
\text{DO 90} \text{ N = 1,NEQ}
\]

ENTERING FLOWSTREAMS

\[
\text{FL1} = 0.
\]

\[
\text{FL2} = 0.
\]

\[
\text{T1} = 0.
\]

\[
\text{T2} = 0.
\]

\[
\text{IF (NODE .NE. NCOLL) GO TO 70}
\]

\[
\text{T1} = \text{TIN}
\]

\[
\text{FL1} = \text{FLWS}
\]

70

\[
\text{IF (NODE .NE. NLOAD) GO TO 72}
\]

\[
\text{IF (FLWL .LE. 0.) GO TO 72}
\]

\[
\text{T1} = (\text{FL1} \ast \text{T1} + \text{FLWL} \ast \text{TL})/(\text{FL1} + \text{FLWL})
\]

\[
\text{FL1} = \text{FL1} + \text{FLWL}
\]

FLOW FROM ADJACENT NODES

72

\[
\text{IF (NODE .EQ. NSTART) GO TO 80}
\]

\[
\text{IF (DIRECT .EQ. 1) GO TO 76}
\]

NET FLOW UP FROM NODE BELOW

\[
\text{FL2} = 0.
\]

\[
\text{IF (NLOAD .GT. NODE) FL2 = FLWL}
\]

\[
\text{IF (NCOLL .LE. NODE) FL2 = FL2 - FLWS}
\]

\[
\text{T2} = S(AVG+NODE+1)
\]

\[
\text{IF(NODE .LT. LOCFIN) T2 = S(AVG+NODE+1)}
\]

\[
\text{GO TO 80}
\]

NET FLOW DOWN FROM NODE ABOVE

76

\[
\text{FL2} = 0.
\]

\[
\text{IF (NCOLL .LT. NODE) FL2 = FLWS}
\]

\[
\text{IF (NLOAD .GE. NODE) FL2 = FL2 - FLWL}
\]

\[
\text{T2} = S(AVG+NODE-1)
\]

\[
\text{IF(NODE.LE.(LOCFIN+1)) .AND. LOCFIN.GT.0) .}
\]

\[
\text{T2=S(AVG+NODE-1)}
\]

LOSSES TO ENVIRONMENT

80

\[
\text{UA} = (\text{ULOSS+USHORT(NODE)}) \ast \text{PER*H(NODE)}
\]

\[
\text{UAFI=0.}
\]

\[
\text{IF(NODE.LE.LOCFIN .AND. OUT(8).LE.0.)}
\]

\[
\text{UAFI=UAF*H(NODE)/XAUX}
\]

\[
\text{IF(TPLUE.GT.0.1)}
\]

\[
\text{UAFI=CMIN*(1-EXP(-UAF/CMIN))H(NODE)/XAUX}
\]
IF(NODE.EQ.1) UA = UA + UAETOP
IF(NODE.EQ.NEQ) UA = UA + UAEBOT

C CAPACITANCE OF EACH NODE
MCPN=MSCP*H(NODE)/HIGH

C FIND TEMPERATURE OF NODE AT END OF TIMESTEP AND AVERAGE
C NODE TEMPERATURE OVER TIMESTEP.
KI = INIT + NODE
KF = FINAL + NODE
KAVG = AVG + NODE
AA = -(FL1 + FL2 + UA + UAFI)/MCPN

BB=(FL1*T1+FL2*T2+UA*TENV+UAFI*TFLUE+QB(NODE,1)+QB(NODE,2))/
   . MCPN
TI = S(KI)
CALL DIFEQ(TIME,AA,BB,TF,TAVG)
S(KF) = TF
S(KAVG) = TAVG

C MAINTAIN U AND V ARRAYS FOR AUXILIARY HEATER ALGORITHM.
C IF AUX HEAT IS ADDED TO NODE, THE FINAL NODE TEMPERATURE
C IS INCREASED BY AUX*U(NODE) AND THE AVERAGE TEMPERATURE
C IS INCREASED BY AUX*V(NODE).
IF(ABS(AA) .GT. 0.) GO TO 84
U(NODE) = DELT/MCPN
V(NODE) = U(NODE)/2.
GO TO 85
84 U(NODE) = (EXP(AA*DELT)-1.)/AA/MCPN
V(NODE) = ((EXP(AA*DELT)-1.)/AA/DELT-1.)/AA/MCPN

C GO ON TO NEXT NODE. SWITCH DIRECTIONS AND START GOING UP
C IF NODE=NEQ
85 NODE = NODE + DIRECT
IF (NODE .LE. NEQ) GO TO 90
DIRECT = -1
NODE = NSTART - 1
90 CONTINUE

IF(NEQ.EQ.1) GO TO 200

C ** Absolute Checks on Temperature Instabilities for
C Both Modes
150 NEQEND = NEQ - 1.
DO 155 JJ=1,NEQEND
   IF(S(FINAL+JJ).LT.S(FINAL+JJ+1)) GO TO 165
155 CONTINUE

GO TO 400

165 TMIX = 0.
HMIX = 0.
DO 175 LL=JJ,NEQ
   TMIX = TMIX + S(FINAL+LL)*H(LL)
   HMIX = HMIX + H(LL)
   IF(LL.EQ.NEQ) GO TO 185
   IF(TMIX/HMIX.GT.S(FINAL+LL+1)) GO TO 185
175 CONTINUE

185 TMIX = TMIX/HMIX
DO 195 KK=JJ,LL
   QTMIX = (TMIX - S(FINAL+KK))/U(KK)
   S(FINAL+KK) = TMIX
   S(AVG+KK) = S(AVG+KK) + QTMIX*V(KK)
195 CONTINUE
GO TO 150

C CHECK TO SEE IF TANK TEMP. IS GREATER THAN BOILING TEMP.
C
400 DO 405 JJ=1,NEQ
   IF(S(FINAL+JJ).GT.TBOIL) THEN
      MCPN=MSCP*H(JJ)/HIGH
      QVENT=QVENT+MCPN*(S(FINAL+JJ)-TBOIL)
      S(AVG+JJ)=S(AVG+JJ) - (S(FINAL+JJ)-TBOIL)/2.
      S(FINAL+JJ)=TBOIL
   ENDIF
405 CONTINUE

C AUXILIARY HEATER
C DO TOP ELEMENT FIRST, THEN BOTTOM ELEMENT
C
4009 DO 4009 II=1,NEQ
      QB(II,1)=0.
      QB(II,2)=0.
4009 CONTINUE
DO 4006 II=I1,I2,I3
   QBOOST(II)=(1-FRACT)*QBOOST(II)
   QBTOT(II)=O.O
   KLOCT = LOCT(II) + FINAL
   IF(INFO(7).LT.NSTK) GO TO 95
   IF(OUT(8+II).GT.0) GO TO 100
4006 GO TO 4006
IF (S(KLOCT).GE.TSET(II) .OR. QBOOST(II).LE.0 .OR. TIME.EQ.TIMEO) GO TO 4006
IF(.NOT.(HTRON(II)).AND. S(KLOCT).GE.(TSET(II)-TDB(II))) GO TO 4006


DO 130 J = 1,LOC(II)
   TH = TSET(II)
   KF = FINAL + J - 1
   IF (J .GT. 1) TH = AMIN1(TSET(II),S(KF))
   QBTOT(II) = 0.
   IF(J.GT.1) QB((J-1),II)=0.0
   DO 126 K = J,LOC(II)
      KF = FINAL + K
      QB(K,II) = AMAX1(0.,(TH - S(KF))/U(K))
   END
   QBTOT(II) = QBTOT(II) + QB(K,II)
   IF (QBTOT(II) .LE. QBOOST(II)) GO TO 135

130 CONTINUE

QB(LOC(II),II) = QBOOST(II)
QBTOT(II) = QBOOST(II)
J = LOC(II) + 1

C HEAT NODES J,J+1,...,LOC(II) TO TEMPERATURE TH.
C DIVIDE REMAINING AUXILIARY HEAT BETWEEN NODES J-1,J,...,LOC(II) (PROVIDED J .GT. 1)

135 QBADD = 0.
IF (J .LT. 2) GO TO 138
J = J - 1
QBADD = (QBOOST(II) - QBTOT(II))

138 QBTOT(II) = 0.
SUMUJ = 0.
DO 139 K = J,LOC(II)
   SUMUJ=SUMUJ+1./U(K)
139 CONTINUE

DO 140 K = J,LOC(II)
   QB(K,II) = QB(K,II) + QBADD/(SUMUJ*U(K))
   QBTOT(II) = QBTOT(II) + QB(K,II)
140 CONTINUE

C DETERMINE IF 2ND ELEMENT IS ALLOWED TO COME ON IN TIMESTEP
IF (AUXMOD.EQ.1) THEN
   FRACT=QBTOT(II)/QBOOST(II)
ELSE
   FRATC=0
ENDIF
CHECK TO SEE IF TANK TEMP. IS GREATER THAN THE BOILING TEMPERATURE.

DO 4005 JJ=1,NEQ
   IF(S(FINAL+JJ).GT.TBOIL) THEN
   MCPN=MSCP*H(JJ)/HIGH
   QVENT=QVENT+MCPN*(S(FINAL+JJ)-TBOIL)
   S(AVG+JJ)=S(AVG+JJ) - (S(FINAL+JJ)-TBOIL)/2.
   S(FINAL+JJ)=TBOIL
   ENDIF
CONTINUE
CONTINUE
CONTINUE
LOOPED=.TRUE.
GO TO 65

C ONE NODE TANK EQUATIONS
200 UA = (ULOSS+USHORT(1))*PER*HIGH + UAETOP + UAEBOT
UAFI=0.
C
IF(OUT(8).LE.O.) UAFI=UAF
IF(TFLUE.GT.0.1) UAFI=CMIN*(1-EXP(-UAFI/CMIN))
MCPN=MSCP
AA = -(FLWS + FLWL + UA+UAFI)/MCPN
BB = (FLWS*TIN + FLWL*TL + UA*TENV+UAFI*TFLUE)/MCPN
TI = S(INIT+1)
CALL DIFFEQ(TIME,AA,BB,TI,TF,TAVG)
S(FINAL+1) = TF
S(AVG+1) = TAVG

C CHECK TO SEE IF TANK TEMP. IS GREATER THAN THE BOILING TEMPERATURE.
IF(S(FINAL+1).GT.TBOIL) THEN
   QVENT=MCPN*(S(FINAL+1)-TBOIL)
   S(AVG+1)=S(AVG+1) - (S(FINAL+1)-TBOIL)/2.
   S(FINAL+1)=TBOIL
ENDIF

C AUXILIARY HEATER
DO 210 II=1,2
   QBTOT(II) = 0.
   IF(INFO(7).LT.NSTK) GO TO 202
   IF(OUT(8+II).GT.0) GO TO 204
   GO TO 210
202 IF (S(FINAL+1).GE.TSET(II))
   1 .OR.QBOOST(II).LE.0..OR.TIME.EQ.TIME0) GO TO 210
   IF(.NOT.(HTRON(II)).AND.
   1 S(FINAL+1).GE.(TSET(II)-TDB(II))) GO TO 210
204 IF(ABS(AA) .GT. 0.) GO TO 205
   U(1) = DELT/MCPN
   V(1) = U(NODE)/2.
   GO TO 207
205 U(1) = (EXP(AA*DELT)-1.)/AA/MCPN
V(1) = (((EXP(AA*DELT)-1.)/AA/DELT-1.)/AA/MCPN
QBTOT(II) = (TSET(II) - S(FINAL+1))/U(1)
QBTOT(II) = AMIN1(QBTOT(II),QBOOST(II))
S(FINAL+1) = QBTOT(II)*U(1) + S(FINAL+1)
S(AVG+1) = QBTOT(II)*V(1) + S(AVG+1)

CONTINUE

C CHECK TO SEE IF TANK TEMP. IS GREATER THAN THE BOILING 
C TEMPERATURE.
IF(S(FINAL+1).GT.TBOIL) THEN
  QVENT=QVENT+MCPN*(S(FINAL+1)-TBOIL)
  S(AVG+1)=S(AVG+1) - (S(FINAL+1)-TBOIL)/2.
  S(FINAL+1)=TBOIL
ENDIF

TF = S(FINAL+1)
QENV = UA*(S(AVG+1)-TENV)+UAFI*(S(AVG+1)-TFLUE)
QENV = UA*(S(AVG+1)-TENV)
QIN = UAFI*(TFLUE-S(AVG+1))

DTDT(1) = (TF - TI)/DELT
GO TO 320

C DETERMINE ENERGY FLOWS, CHANGE IN INTERNAL ENERGY, 
C AND AVERAGE DERIVATIVES
300  TF = 0.
   QENV = 0.
   QIN = 0.
DO 310 K = 1,NEQ
   KI = K + INIT
   KF = K + FINAL
   KAVG = K + AVG
   KOLD = K + OLD
   S(KOLD) = S(KAVG)
   TF = TF + S(KF)*H(K)
   UA = (ULOSS+USHORT(K))*PER*H(K)
   IF(K.EQ.1) UA = UA + UAETOP
   IF(K.EQ.NEQ) UA = UA + UAEBOT
   UAFI=0.
   IF(K.LE.LOCFIN .AND. OUT(8).LE.0.) UAFI=UAFI*H(K)/XAUX
   IF(TFLUE.GT.0.1)UAFI=CMIN*(1-EXP(-UAFI/CMIN))*H(K)/XAUX
   QENV = QENV + UA*(S(KAVG)-TENV) + UAFI*(S(KAVG)-TFLUE)
   QENV = QENV + UA*(S(KAVG)-TENV)
   QIN = QIN + UAFI*(TFLUE-S(KAVG))
310  DTDT(K) = (S(KF) - S(KI))/DELT
   TF = TF/HEIGHT
320  QTANK = FLWL*(S(AVG+1) - TL)
   QIN=FLWS*(TIN-S(AVG+NEQ))
   DELAU=(TF-S(IS+1))*MSCP

C SET OUTPUTS
OUT(1)=S(AVG+NEQ)
OUT(2)=FLWSS
OUT(3)=S(AVG+1)
OUT(4)=FLWLL
OUT(5)=QENV+QVENT
OUT(6)=QTANK
OUT(7)=DELAU
OUT(8)=QBTOT(1)+QBTOT(2)
OUT(9)=QBTOT(1)
OUT(10)=QBTOT(2)
OUT(11)=QIN
OUT(12)=TF
N=NEQ-1
IF(N.LT.2) RETURN 1
DO 350 I=2,N
350 OUT(11+I)=S(AVG+I)
RETURN 1

1001 FORMAT(/2X,'***** WARNING *****',/2X,'THE SET POINT
1 TEMPERATURE OF THE UNIT ',I2,' TYPE 4 STORAGE TANK IS
1 HIGHER THAN'/2X,'THE BOILING TEMPERATURE. THE
1 BOILING TEMPERATURE WILL BE USED AS THE SET',
1 /2X,'POINT.')

END
APPENDIX E

TRNSYS SIMULATION OUTPUT
Active Desuperheater A2
TRNSYS - A TRANSIENT SIMULATION PROGRAM
FROM THE SOLAR ENERGY LAB AT THE UNIVERSITY OF WISCONSIN
VERSION 14.1  LATE 1993

* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *
* * * *
* DHP2-07YR
* DESUPERHEATER/HEAT PUMP (W/5KW STRIP) - ORLANDO - YR *
* (SI Units)
* FEB 1995 *
* *
* * * * * * * * * * * * * * * * * * * * * * * * * * * * * * *

ASSIGN DHP207YR.LST 6
ASSIGN WEATHER\ORLANDO\ORLANDO.TMY 10
ASSIGN INPUT\38YK036.ACS 11
ASSIGN INPUT\38YK036.HPC 12
ASSIGN DHP207YR.PLT 13

SIMULATION 1.000E+00 8.759E+03 1.000E-01

LIMITS 100 50
WIDTH 72
NOLIST

DIFFERENTIAL EQUATIONS SOLVED BY MODIFIED EULER

*****WARNING*****
UNIT 35 TYPE35 REQUIRES THE SUBROUTINE TALF
MAKE SURE THAT THIS SUBROUTINE IS LINKED IN TO AVOID PROBLEMS.

*****WARNING*****
UNIT 36 TYPE35 REQUIRES THE SUBROUTINE TALF
MAKE SURE THAT THIS SUBROUTINE IS LINKED IN TO AVOID PROBLEMS.

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13.00  .06  I**
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17.00  .04  I*
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20.00  .10  I***
21.00  .14  I***
22.00  .19  I*****
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** TOTAL 4.04 **

** KWUNIT : MIN= 0.00  MAX= 24.00 **

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**ENERGY BALANCE 1:** \(\text{KWUNIT} + \text{KWHAUX} - \text{KWLOAD} - \text{KWLOSS} - \Delta T\)

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**IN INTERVALS OF 1.000 MONTH**

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LIST OF REFERENCES


Bell, K.J., 1972, "Temperature Profiles in Pure Component Condensers with Desuperheating and/or Subcooling," 71st National Meeting of the American Institute of Chemical Engineers, Dallas, Texas.


