A Progress Report on the Experimental Evaluation of Dry/Wet Air-Cooled Heat Exchangers

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FOREWORD

The Dry Cooling Enhancement project has been a Battelle program for about six years, jointly supported by the U.S. Department of Energy and the Electric Power Research Institute, at an approximate professional staff level of 12 man-years/year. Battelle Memorial Institute operates the Pacific Northwest Laboratory (PNL) for the U.S. Department of Energy. The objectives of the project are to develop, in conjunction with utilities in the cooling tower industry, dry cooling systems and dry/wet cooling systems that conserve scarce water resources and increase power plant siting flexibility with a minimum increase in the cost of generating electricity. The thrust of the project has included the following initial and ultimate objectives.

Initial Objectives

- Develop economic and performance models for cost optimization of total heat rejection systems using dry and dry/wet cooling.
- Analyze and disseminate operating experience on existing dry-cooled plant performance.
- Demonstrate certain features of existing technology equipment to provide confidence for specification by utilities.

Ultimate Objective

- Promote water conservation through industry use of dry cooling by developing and demonstrating the reliability of lower cost systems. The development of advanced dry/wet systems is also considered to be within this scope.

The depth and extent of this project can perhaps best be presented through a listing of the documents which have been issued by the staff involved in this project. Unless otherwise specified, these documents were prepared by the Pacific Northwest Laboratory, Richland, Washington.

Six documents have been written on the analysis of performance of existing dry-cooled plants and air-cooled heat exchanger equipment:

1. Dry Cooling Tower Project: Results of Industrial Contacts Through August 1972. BNWL-1878, November 1, 1974.
2. A Survey of Materials and Corrosion Performance in Dry Cooling Applications. A. B. Johnson, Jr., D. R. Pratt and G. E. Zima, BNWL-1948, March 1976.
3. European Dry Cooling Tower Operating Experience. J. G. DeSteeze and K. Simhan, BNWL-1995, March 1976.
4. Mathematical and Experimental Investigations on Dispersion and Recirculation of Plumes from Dry Cooling Towers at WYODAK Power Plant in Wyoming. Y. Onishi and D. S. Trent, BNWL-1982, February 1976.
5. Aluminum Alloy Performance Under Dry Cooling Tower Conditions. A. B. Johnson, Jr., S. Begaj, M. W. Martini, and R. P. May, PNL-02392, December 1977.
6. Aluminum Alloy Performance in Industrial Air-Cooled Applications. K. R. Wheeler, A. B. Johnson, Jr., and R. P. May, PNL-SA-8103, April 1980.

Five documents have addressed advanced dry/wet-cooled systems.

1. Preliminary Evaluation of Wet/Dry Cooling Concepts for Power Plants. W. V. Loscutoff, BNWL-1969.
2. Compatibility of Ammonia with Candidate Dry Cooling System Materials. D. R. Pratt, BNWL-1992, April 1976.
3. Scale Formation in Deluged Dry Cooling Systems. D. R. Pratt, BNWL-2070, March 1976.
4. Ammonia as an Intermediate Heat Exchange Fluid for Dry-Cooled Towers. R. T. Allemann, B. M. Johnson and G. C. Smith, BNWL-SA-5997, September 1976.
5. Cost Analysis of an Ammonia Dry Cooling System with Chicago Bridge and Iron Peak Shaving System. M. K. Drost and B. M. Johnson, PNL-3272, November 1980.

Three reports have been issued which consider the future need for any cooling and the potential benefit/cost ratio for a large-scale demonstration.

1. An Overview of Economic, Legal and Water Availability Factors Affecting the Demand for Dry and Wet/Dry Cooling of Thermal Power Plants. P. L. Hendrickson, BNWL-2268, September 1978.

2. Estimation of Benefits from Demonstrating Advanced Dry Cooling Technology: A Framework and Partial Analysis. J. W. Currie and J. T. Foley, BNWL-2182, April 1977.
3. Potential Use of Dry Cooling in Support of Advanced Energy Generation Systems. D. W. Mayer, E. M. Arnold and R. T. Allemann, PNL-3149, September 1979.

Cost optimization of dry-cooled and dry-heat-cooled rejection systems has been extensively studied in three reports.

1. A Review and Assessment of Engineering Economic Studies of Dry-Cooled Electrical Generating Plants. B. C. Fryer, BNWL-1976, March 1976.
2. Heat Transfer and Pressure Drop Characteristics of Dry Tower Extended Surfaces. Part I: Heat Transfer and Pressure Drop Data. Prepared for the Pacific Northwest Laboratory by PFR Engineering Systems, Incorporated, PFR 7-100, March 1976.
3. Heat Transfer and Pressure Drop Characteristics of Dry Tower Extended Surfaces. Part II: Data Analysis and Correlations. Prepared for the Pacific Northwest Laboratory by PFR Engineering Systems, Incorporated, PFR 7-102, June 1976.

A group of reports has been issued that: 1) develop cost optimization models for dry cooling systems based on available technology; and 2) compare the cost analysis of three systems with the projected cost of several advanced dry and dry/wet systems. Included in this group of reports are:

1. Dry Cooling of Power Generating Stations: A Summary of the Economic Evaluation of Several Advanced Concepts via a Design Optimization Study and a Conceptual Design and Cost Estimate. B. M. Johnson, R. T. Allemann, D. W. Faletti, B. C. Fryer, and F. R. Zaloudek, BNWL-2120, September 1976.
2. An Engineering and Cost Comparison of Three Different All-Dry Cooling Systems. B. C. Fryer, D. W. Faletti, Daniel J. Braun, David J. Braun, and L. E. Wiles, BNWL-2121, September 1976.
3. A Study of the Comparative Costs of Five Wet/Dry Cooling Tower Concepts. F. R. Zaloudek, R. T. Allemann, D. W. Faletti, B. M. Johnson, H. L. Parry, G. C. Smith, R. D. Tokarz, and R. A. Walter, BNWL-2122, September 1976.
4. Costs and Cost Algorithms for Dry Cooling Tower Systems. P. A. Ard, C. H. Henager, D. R. Pratt, and L. E. Wiles, BNWL-2123, September 1976.

5. A User's Manual for the BNW-I Optimization Code for Dry-Cooled Power Plants. David J. Braun, Daniel J. Braun, W. V. DeMier, D. W. Faletti, and L. E. Wiles, BNWL-2180, January 1977.
6. Comparative Cost Study of Four Wet/Dry Cooling Concepts That Use Ammonia as the Intermediate Heat Exchange Fluid. Daniel J. Braun, R. D. Tokarz, B. M. Johnson, R. T. Allemann, David J. Braun, D. W. Faletti, L. J. MacGowan, H. L. Parry, G. C. Smith, and F. R. Zaloudek, PNL-2661, May 1978.
7. An Engineering and Cost Analysis of a Dry Cooling System Augmented with a Thermal Storage Pond. M. K. Drost and R. T. Allemann, PNL-2745, September 1978.
8. A User's Manual for the BNW-II Optimization Code for Dry/Wet Cooled Power Plants. Daniel J. Braun, J. A. Bamberger, David J. Braun, D. W. Faletti, and L. E. Wiles, PNL-2674, Vol. 1, May 1978.
9. A Description and Cost Analysis of a Deluge Dry/Wet Cooling System. L. E. Wiles et al., PNL-2498, June 1978.

The primary focus of the dry cooling project has been to carry out a large-scale test of the system in the Advanced Concepts Test (ACT) facility built at the Kern Plant of Pacific Gas and Electric of Bakersfield, California.

Battelle performed the conceptual design study and cost estimates of the proposed test facility with assistance from an architect-engineering firm, and has managed the subsequent design and construction by the C. F. Braun Company of Alhambra, California. Documents which have been issued on the ACT facility design and the testing project include:

1. Advanced Concepts Facility--Measurements and Suggested Test Plan. F. R. Zaloudek, L. J. Brown and R. T. Allemann. Prepared for the Electric Power Research Institute by Battelle, Pacific Northwest Laboratories, Richland, Washington. Research Project RP422-3. EPRI-CS-1530, September 1980.
2. Development of an Advanced Concept Dry/Wet Cooling of Power Generating Plants. B. M. Johnson, ed. Prepared for the Electric Power Research Institute by Battelle, Pacific Northwest Laboratories, Richland, Washington. Research Project RP422-3. EPRI-CS-1668, February 1981.

One of the methods of dry/wet cooling under test at the ACT facility is deluge cooling. In order to study the feasibility and performance characteristics of deluge cooling, PNL has constructed a small-scale air heat transfer facility. This facility is unique because its flexibility allows a combined heat and mass transfer phenomena to be studied in a variety of configurations. Studies of "limits" of water chemistry to avoid corrosion and deposition are being carried on in conjunction with these heat transfer studies in separate facilities. Reports of this work include:

1. Analysis and Experimental Results from the Pacific Northwest Laboratory Augmented Dry Cooling Surface Test Project. H. L. Parry, L. J. MacGowan, D. K. Kreid, and B. M. Johnson. PNL-2746, November 1978.
2. Dry/Wet Performance of a Plate-Fin Air-Cooled Heat Exchanger with Continuous Corrugated Fins. S. G. Hauser, D. K. Kreid and B. M. Johnson. PNL-3428, February 1981.
3. Experimental Evaluation of a Dry/Wet Air-Cooled Heat Exchanger with Helically Wound Tubes. S. G. Hauser, R. L. Gruel, J. C. Huenefeld, E. J. Eschbach, B. M. Johnson and D. K. Kreid. PNL-4043, July 1982.
4. Deposition and Corrosion Phenomena on Aluminum Surfaces Under Deluged Dry Cooling Tower Conditions. K. R. Wheeler, R. P. May, J. G. Douglas, and J. H. Tylczak. Prepared for the Electric Power Research Institute by Battelle, Pacific Northwest Laboratories, Richland, Washington. EPRI-CS-1926, July 1981.

Finally, the application of dry and dry/wet cooling to other energy-related situations has been treated in these documents:

1. Potential Use of Dry Cooling in Support of Advanced Energy Generation Systems. D. W. Mayer, E. M. Arnold and R. T. Allemann. PNL-3149, September 1979.
2. Cost Analysis of an Ammonia Dry Cooling System with a Chicago Bridge and Iron Peak Shaving System. M. K. Drost and B. M. Johnson. PNL-3272, December 1980.
3. Cost Benefits from Applying Advanced Heat Rejection Concepts to a Wet/Dry Cooled Binary Geothermal Plant. D. W. Faletti. PNL-3419, March 1981.

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This report contains an account of the work sponsored by the Division of Advanced Systems and Materials Production of the United States Department of Energy (DOE) under the Dry Cooling Enhancement Program at the Pacific Northwest Laboratory (PNL). This work was completed under the direction of I. Helms and W. F. Savage, Project Officer and Manager, respectively, Advanced Concepts Evaluation Section of the Advanced Nuclear Systems and Projects Division; and B. M. Johnson, Manager of the Dry Cooling Enhancement Program at PNL.

SUMMARY

The ultimate goal of this project was to contribute to the development of improved cooling facilities for power plants. Accomplishing this goal would help to conserve increasingly scarce fresh water supplies in an environmentally compatible and economically viable manner. Specifically, the objective during FY-81 was to experimentally determine the thermal performance and operating characteristics of an air-cooled heat exchanger surface manufactured by the Unifin Company located in London, Ontario. The performance of the spiral-wound finned tube surface (Unifin) was compared with two inherently different plate-fin surfaces (one developed by the Trane Co. located in La Crosse, Wisconsin, and the other developed by the HÖTERV Institute in Hungary) which were previously tested as a part of the same continuing program.

Under dry operation the heat transfer per unit frontal area per unit inlet temperature difference (ITD) of the Unifin surface was 10% to 20% below that of the other two surfaces at low fan power levels. At high fan power levels the performances of the Unifin and Trane surfaces were essentially the same, which was 25% higher than the HÖTERV surface. The design of the Unifin surface created a much more tortuous path for air and deluge water causing significantly larger air-side pressure drop through the heat exchanger both in dry and deluge operation.

Generally higher overall heat transfer coefficients were calculated for the Unifin surface under deluged operation. They ranged from 2.0 to 3.5 Btu/hr-ft²-°F as compared to those calculated for the Trane and HÖTERV surfaces under similar conditions, which were usually less than 2.0 Btu/hr-ft²-°F. The heat transfer enhancement due to the evaporative cooling effect was also measurably higher with the Unifin surface as compared to the Trane surface. Both of these observations can be primarily attributed to the better wetting characteristics of the Unifin surface. The tortuous path provided by the spiral fins while causing high pressure drop also created good surface wetting characteristics.

If the thermal performance of the surfaces are compared at equal face velocities, the Unifin surface is as much as 35% better. This method of

comparison accounts for the wetting characteristics while neglecting the effect of pressure drop. Alternatively the surfaces when compared at equal pressure drop essentially the same thermal performance.

Several methods are used in the text for comparing the thermal performance and operating characteristics of each of the surfaces. No specific conclusions are merited which would indicate in general which surface is best. A much more detailed analysis of the specific applications and the corresponding economic factors would be necessary to identify completely the relative merits of each surface.

NOMENCLATURE

A	area
A_c	minimum flow area
A_f	frontal area
A_{io}	area tube wall area
A_x	tube side flow area
a	relative area A_i/A_s
Bi	Biot modulus
C	specific heat
D	characteristic dimension
D_h	hydraulic diameter
F_T	crossflow correction factor
f	fanning friction factor
f_o	effective friction factor (includes entrance and exit losses)
f_r	f evaluated at $Re = 1000$
G	mass flow rate per unit frontal area
g_c	gravitational constant
H	humidity ratio of moist air
h	film heat transfer coefficient
h_e	effective heat transfer coefficient
ITD	inlet temperature difference $(T_r - T_\infty)$,
i	enthalpy of moist air
j	Colburn factor without n
j_o	colburn factor including n
j_r	j_o evaluated at $Re = 1000$
K	parameter in equation for ξ_m
k	thermal conductivity
L	depth of heat exchanger
ℓ	length of fin
$\dot{m}$	mass flow rate
N	NTU rating of heat exchanger
P	static pressure

Pr	Prandtl number
Q	heat transfer rate
Q_d	heat transfer under dry operation
Q_{in}	heat input to system
Q_{req}	heat transfer from primary side to air
$Q_{peripheral}$	heat loss/gain to/from surroundings
Q_w	heat transfer under deluge operation
R	capacity ratio
Re	Reynolds number
r_b	outer tube radius
r_e	outer fin radius
s	surface area to volume ratio
T	temperature
t	thickness
U	overall heat transfer coefficient
v	specific volume
V_o	face velocity
x, y, z	coordinate directions
y_b	fin semi-thickness

GREEK LETTERS

β	dimensionless temperature range
Γ	driving potential ratio
δ	a ratio of property differences, also boundary layer thickness
Δ	difference
η	efficiency of finned surface
θ	nondimensional temperature difference
μ	dynamic viscosity
ν	kinematic viscosity
ξ	thermal resistance transformation parameter
π	3.14159
ρ	density

$\bar{\Sigma}$	overall mass transfer coefficient
σ	free area to frontal area ratio
σ_s	film mass transfer coefficient
Φ	relative humidity
ϕ	heat exchanger effectiveness
x	dimensionless axial coordinate

SUBSCRIPTS

a	air properties
ann	annubar
c	core (test section)
d	properties of deluge film (except Q_d)
dp	dew point
f	properties of fin (except A_f)
h	heater
i	inside surface
l_m	log mean temperature or other property
m	mass transfer parameters also mean value
o	overall property or parameter
p	primary side property or parameter
r	root conditions (except f_r, j_r)
s	surface condition or parameter (except σ_s)
t	tube wall property or parameter
v	vapor water property
w	liquid water property
ws	saturation property
∞	free stream condition
1	inlet condition
2	outlet condition

SUPERSCRIPTS

()*	denotes wet system parameter
()'	denotes saturation condition

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A PROGRESS REPORT ON THE EXPERIMENTAL EVALUATION OF DRY/WET AIR-COOLED HEAT EXCHANGERS

1.0 INTRODUCTION

This report describes work done in support of the Dry Cooling Enhancement (DCE) Program at the Pacific Northwest Laboratory (PNL).^(a) The principal goal of the work performed in the DCE program is to assist the development and demonstration of cooling systems that could eliminate or substantially reduce the consumption of water used for cooling electric power plants.

One of the more promising techniques being studied in this program is the dry/wet or "deluge" system, i.e., a normally dry air-cooled system is enhanced by thoroughly wetting the extended surface during periods of peak cooling demand. Part of the DCE program involved investigating on a small scale the phenomena of evaporative heat transfer from a heated surface. The primary goals of the work performed in this part of the DCE program were to:

- develop analytical means for predicting the rates of heat transfer and evaporation from a wet surface
- determine experimentally the performance and operating characteristics of heat exchangers in dry/wet operation.

A summary of the work done during FY-81 to evaluate a heat exchanger surface manufactured by the Unifin Company is included in this report. Typical of many commercial air-cooled heat exchangers, the Unifin surface is a spiral-wound finned tube heat exchanger (detail dimensions of the surface are given in Section 4.0). Various tests were performed to give an adequate evaluation of both the dry and the deluged performance of the surface. The results of this performance evaluation of the Unifin surface compared to the results of earlier work on the performance of two different types of plate-fin heat exchangers (the HÖTERV and Trane surfaces).

(a) The Pacific Northwest Laboratory is operated by the Battelle Memorial Institute, Pacific Northwest Division, under contract to the United States Department of Energy.

Section 2.0 discusses the conclusions of this work with appropriate recommendations. Section 3.0 briefly gives the background of the previous work done at PNL on enhanced cooling of air-cooled heat exchangers. Section 4.0 describes the test facility and the Unifin surface, as well as some operational characteristics. Finally, performance data are presented in Section 5.0, including a comparison with the performance data of other surfaces previously tested as part of this program.

2.0 CONCLUSIONS

Since the Unifin surface is not a plate-fin type of heat exchanger, an inherent difference exists in the fluid mechanics of the deluge operation. The two plate fin surfaces provide a rather direct path for the deluge water to flow from inlet to outlet, allowing a continuous thin film of water to form on the fins. The influence of air flowing over this thin film causes it to dry out in places resulting in a type of channeling and nonuniform wetting. The spiral-fin surface forces the water to be well distributed throughout the heat exchanger. Since the fins on neighboring tubes do not touch, the water drops from fin to fin as it traverses downward. This breakup of the deluge water allows the air to blow the droplets over the fins more randomly, resulting in a higher percentage of wetting. The phenomena is manifested in higher heat transfer coefficients and a higher surface heat flux.

The performance of the Unifin heat exchanger as compared to the Trane and HÖTERV heat exchangers is summarized as follows:

Pressure Drop

- At nominal dry operating conditions (2000 lb/hr-ft² air flow rate) the pressure drop of the Unifin surface was 45% greater than the Trane surface and 30% greater than the HÖTERV surface.
- At nominal deluged operating conditions (1000 lb/hr-ft² air flow rate) the pressure drop of the Unifin surface was 100% greater than the Trane surface and 50% greater than the HÖTERV surface.

Dry Heat Transfer

- At low fan power levels the Trane surface has about 10% better heat transfer per unit frontal area than the Unifin surface. Likewise the HÖTERV surface is about 20% better than the Unifin surface.
- At high fan power levels the heat transfer per unit frontal area is approximately equal for the Trane surface and the Unifin surface and both are 25% higher than the HÖTERV surface.

- At low fan power levels the heat transfer per unit volume is the same for the Unifin and HÜTERV surfaces and both are about 15% below the Trane surface.
- At high fan power levels the heat transfer per unit volume of the Trane surface is about 15% higher than the Unifin surface. The Unifin surface is about 40% higher than the HÜTERV surface.

Deluge Heat Transfer

- The heat transfer enhancement of the Unifin surface is about 15% higher than for the Trane surface.
- The thermal performance of the Unifin surface (i.e., Q/IT_0) is 8 to 10% higher than the Trane surface at the same fan power.

In general, the results indicate that the Unifin surface design causes both better surface wetting and higher air-side pressure drop. Based on equal fan power consumption in the dry mode, the Trane surface is slightly better, whereas, in the deluged mode, the Unifin surface is slightly better.

3.0 BACKGROUND

Most of the power plants that have been built in recent years and that are planned for the near future will be equipped with evaporative cooling systems (or wet towers) for waste heat rejection. Evaporative cooling technology is a proven, versatile, reliable, and economical option. Therefore, it is likely to remain the prime means for large-scale heat rejection applications in most regions of the world for the foreseeable future. However, in arid regions where water supplies are limited, or in other areas where the growing demand for water from all sectors of society is approaching the available supply, alternative cooling systems must be considered.

3.1 ALTERNATIVES TO WET COOLING

The only practical alternatives to wet towers for the foreseeable future are dry-cooled systems, various dry/wet combinations, and hybrids. Dry cooling adds markedly to the cost of power produced when compared to a conventional wet-cooled plant which uses water at effectively no cost. The increased cost is not only because of the high capital cost of the dry systems, but also because of the inability to meet peak power demands during warm weather periods. On purely economic grounds, power plants can generally afford to pay more for water than competing users. However, the pressure to reduce water consumption for power plant cooling is likely to be more political than economic because of the volatile and highly politicized nature of water rights and water allocation issues.

In locations where limited quantities of water are available for power plant cooling, studies (Loscutoff 1976; Braun et al. 1978; Wiles et al. 1978) have shown that combinations of dry and evaporative cooled plants may be the short-term answer in lieu of paying the penalties of all-dry cooling. By using dry/wet cooling, with wet cooling used only during periods of high ambient temperature, the consumption of water can be greatly reduced and a cost savings may be realized due to a net reduction in the "size" of the cooling system.

In the method of dry/wet cooling that has been referred to here as the "deluge" concept, the performance of a normally dry-cooled heat exchanger is enhanced during periods of high ambient temperature and/or high cooling demand by "deluging" the surface with water. The air flowing through the system causes evaporation from the delugeate film, driving the air/water interface temperature toward the local wet bulb temperature and thereby increasing substantially the rate of heat rejection.

The primary potential benefits of deluge vis-à-vis all-dry cooling are a combination of the following:

- The condenser temperature (and thus the turbine back pressure) can be maintained lower than would be possible with an all-dry system under the same ambient conditions.
- The heat rejection capacity of the system is increased (or the "size" of the required system is reduced), compared to an all-dry system under similar conditions.

As a consequence of the above and other factors, the cost of building and operating a dry-cooled system with evaporative peaking can be substantially less than that of an all dry-cooled system of the same capacity, with only minimal consumption of water.

3.2 PREVIOUS WORK ON DRY/WET COOLING

The Water Augmentation Test Apparatus (WATA) facility was built in 1977 for the purpose of evaluating the heat and mass transfer performance of a variety of air-cooled heat exchanger surfaces. The WATA facility consists of a low-speed atmospheric wind tunnel, separate primary water and deluge water circulation loops, and the associated controls and instrumentation. Specifically, the objectives of the work done in connection with the WATA facility are as follows:

- 1) to determine all-dry (non-augmented) performance for comparison with other air-cooled heat exchanger surfaces
- 2) to establish the magnitude of the potential benefit of evaporative cooling

- 3) to measure dry/wet heat transfer performance and air-side pressure drop as they are affected by climate (air temperature and humidity), air-flow rate, and deluge flow rate
- 4) to compare measured performance to the performance predicted by analytical models developed at PNL, thus verifying and helping to define those models
- 5) to determine the physical operating limits of the deluged surface, particularly the limits of air flow and deluge flow such that a wetted surface is maintained.

When this program began, the information available on dry/wet cooling was extremely limited. The only known analytical work on deluge cooling at that time was a brief paper by Tasnadi (1972), which was translated from Hungarian. A limited amount of proprietary data were also made available by the HÖTERV Institute in Hungary from tests performed on a Heller-Forgo plate-fin heat exchanger. A subsequent literature review also identified an earlier analysis by Threlkeld (1962) that used a similar approach for calculating heat/mass transfer in finned air chillers. Although the general approach used in both of the preceding analyses was similar to that employed here (i.e., use of the enthalpy driving potential), current work at PNL (Kreid, Johnson and Faletti 1978; Hauser, Kreid and Johnson 1981) has expanded the theoretical basis of the model and has substantially increased its generality and applicability.

An initial report and paper (Parry et al. 1979; Kreid, Parry and MacGowan 1979) presented the dry and deluged performance evaluation of the "Forgo" plate-fin heat exchanger surface. This surface was developed by and manufactured for the HÖTERV Institute of Hungary and hereafter will be identified as the HÖTERV surface. In this design the flat aluminum plate fins contained a dense array of crimped perforations which acted as "turbulators" to enhance the surface heat transfer coefficient. The primary side of the heat exchanger consisted of six rows of tubes in a staggered array mounted with the tubes horizontal in crossflow. The 2'x6' test section was aligned at 15° from the vertical with deluge water trickled on the surface from top to bottom. The

test results demonstrated that wetting the surface of an air-cooled finned heat exchanger can greatly enhance heat transfer, especially under conditions of high ambient temperature and low ambient humidity. Heat transfer enhancement ratios (relative to dry operation at the same air-side conditions and pressure drop) of from 1.5 to 7 were measured for a wide range of operating conditions. The same report also presented an evaluation of the dry performance of two configurations of a chipped-(or skived-) fin heat exchanger surface manufactured under license from the Curtiss-Wright Company.

A subsequent report (Hauser, Kreid and Johnson 1981) provided the data and the theoretical basis for predicting the performance of another plate-fin heat exchanger, manufactured by the Trane Company principally for air conditioning service. The Trane design featured smooth wavy plate fins with the primary side consisting of only three staggered rows of tubes. The Trane surface gave higher heat transfer than the HÖTERV surface for the same fan power. This was due, at least in part, to the narrower depth of the Trane heat exchanger which resulted in a much lower pressure drop. The performance of these various surfaces will be shown in more detail in Section 5.0 and used to compare with the performance data measured for the Unifin surface.

4.0 TEST APPARATUS AND PROCEDURES

The experiments were conducted in a specially designed wind tunnel at the Pacific Northwest Laboratory, which is operated for the United States Department of Energy by Battelle Memorial Institute. For each test, air that was first heated and humidified to specified conditions was circulated at a controlled rate through a 2 ft x 6 ft heat exchanger module. Hot water was circulated through the heat exchanger tubes at high flow rates to maintain an essentially isothermal condition on the tube side. Deionized water, sprayed on the top of the vertically oriented fins, was collected at the bottom of the core and recirculated. Instrumentation measured flow rates and thermodynamic conditions in the air, in the core circulation water, and in the deluge water.

Measurements of the air-side pressure drop and heat rejection rate were made as a function of air-flow rate, air inlet temperature and humidity, deluge water flow rate, and the core inclination from the vertical. The data were reduced to determine an overall heat transfer coefficient and an effective deluge film convective coefficient.

4.1 DESCRIPTION OF FACILITIES

The simplified schematic diagram, Figure 4.1, illustrates the three principal flow loops in the WATA facility; namely, the wind tunnel, the primary water loop, and the deluge water loop.

4.1.1 Wind Tunnel

The low-speed, single pass, atmospheric wind tunnel was designed to provide a steady uniform flow of air at selected conditions of temperature, humidity, and flow rate. The dimensions of the rectangular test section module were 2 ft by 6 ft high. Air was brought into the wind tunnel by a centrifugal fan capable of producing face velocities up to 13 fps.

Air temperature and humidity at the inlet to the test section were controlled with a steam heat exchanger and steam-injection humidifier. The air then passed through a set of turning vanes, flow straighteners, screens, and through the test section where the primary heat and mass transfer (deluge

FIGURE 4.1. Schematic of WATA Test Apparatus

operation) occurred. Leaving the test section, the air entered a long straight cylindrical pipe containing an annubar flowmeter and was exhausted to the atmosphere.

4.1.2 Primary Water Loop

An electrically heated water circulation loop provided the heat that was rejected in the heat exchanger. The recirculating water was split, with approximately 10% of the flow passing through an electric heater capable of a variable output (SCR-controlled) of up to 135 kW. The heated stream was subsequently mixed with the remainder of the circulation water and passed through the test section in a cross-flow arrangement relative to the air. Because the flow rate was large, the temperature drop in the water across the test section was small, typically about 2°F or less.

4.1.3 Deluge Water Loop

This loop recirculated deionized water to wet the air-side surfaces of the heat exchanger. Water was sprayed or dripped directly onto the top surface of the fins. The deluge flow rate was always in excess of the rate of evaporation (roughly ten times) and the excess flow was collected at the bottom of the heat exchanger and recycled. Deionized water was used to eliminate the effect of scaling and corrosion on the heat transfer performance. These effects were studied separately (Wheeler et al. 1981).

In order to assess experimentally the combined effects of air drag, surface tension, and gravitational forces on the distribution of deluge water running down the surfaces of the fins, the wind tunnel was designed so that the entire rectangular section containing the heat exchanger assembly could be inclined at angles of 0 to 90° from the vertical. The principal disadvantage to this configuration was that the straight sections of rectangular duct upstream and downstream of the test section were relatively short compared to normal practice in wind tunnel design. To compensate for these deficiencies, an elaborate system of turning vanes, screens, and flow straighteners was devised to achieve acceptable uniformity of temperature, humidity and velocity distributions at the inlet to the heat exchanger.

4.1.4 Instrumentation

All dry bulb air and water temperatures were measured with Type T thermocouples. Where appropriate, average temperatures were computed from arrays of several thermocouples distributed across the air flowstream. Differential thermocouples were used to measure the water temperature differences across the circulation water heater and the heat exchanger.

The air-stream humidity was measured upstream and downstream of the heat exchanger using two General Eastern optical dewpoint hygrometers, supplied by continuous sample streams from the wind tunnel, using aspirating probes. Spatially averaged samples are obtained at both measurement locations using multiple sampling probe arrays analogous to the thermocouple arrays.

Two solid-state devices made by Thunder Scientific were also used to measure humidity. These units were used to compare the values of the dewpoint measurement.

The flow rates in the primary water and deluge water flow loops were measured with turbine meters. The total air flow was measured using an Annubar averaging pitot probe flowmeter that was installed downstream of the test section in a long straight section of cylindrical duct. The pressure drop across the heat exchanger, the static pressure in the duct and the ambient atmospheric pressure were measured with indirect liquid manometers. Most of the data were printed out on paper tape using Fluke Model 2420 automatic data logger, and manually transferred to the data reduction program.

A much more detailed description of the apparatus and instrumentation is given in the two previous reports (Hauser et al. 1981; Parry et al. 1979).

4.2 DESCRIPTION OF TEST SURFACES

Although the main intention of this report is to present the performance data of the Unifin surface, it is useful to compare this data with the other two surfaces tested in WATA. In order to facilitate comparison, the important design specifications for each of the three surfaces are presented in Table 4.1. The important dimensions are given for each surface in Figures 4.2,

TABLE 4.1. Heat Exchanger Specifications

	<u>HÖTERV</u>	<u>Trane</u>	<u>Unifin</u>
Total Surface, Area, ft ²	999.2	974.1	705.8
Frontal Area, ft ²	12.0	12.0	12.0
Fin Area, ft ²	942.3	931.0	643.9
Minimum Flow Area, ft ²	5.94	6.41	6.11
Hydraulic Diameter, ft	0.0127	0.0097	0.0145
Tubeside Flow Area, ft ²	0.440	0.073	0.15
Number of Tubes	180	144	202
Tube Passes	1	3	2
Tube O.D., in.	0.768	0.625	0.625
Tube Wall Thickness, in.	0.030	0.049	0.049
Heat Exchanger Depth, in.	5.9	4.5	5.0
Fin Thickness, in.	0.0170	0.0085	0.0080
Surface Density, ft ² /ft ³	155.2	216.0	140.9

FIGURE 4.2. HÖTERV Surface

4.3, and 4.4, and a sketch of the Unifin heat exchanger assembly is shown in Figure 4.5. Sketches of the HÖTERV assembly and the Trane assembly were given in the previous reports (Hauser et al. 1981; Parry et al. 1979).

4.3 TEST PROCEDURES

Both dry and deluged performance was determined for a range of environmental and operating conditions. The parameters of greatest interest were the air side pressure drop and the rate of heat rejection to the air stream.

Previous tests done with the Trane surface measured the peripheral heat loss/gain of the system as a function of the system to ambient temperature difference. (The result was a net addition of heat to the system due to the main circulation pump.) These adiabatic tests were assumed to be the same for the Unifin surface and were not redone.

A series of deluged performance tests were conducted to determine the independent effects of the following variables:

- air temperature
- air humidity

FIGURE 4.3. Trane Surface

- air-flow rate
- primary water temperature (usually fixed)
- deluge water-flow rate
- heat exchanger inclination

Table 4.2 gives the nominal conditions at which each of the tests were run for the Unifin surface. A few tests were duplicated to demonstrate repeatability or to check questionable results.

For each set of conditions, ample time was allowed to reach steady state. Once steady state was reached, four or more recordings of the entire data set were made at one minute intervals. These data were subsequently averaged, using a computerized data analysis procedure, to minimize the effects of short-term transients and randomness.

FIGURE 4.4. Unifin Surface

4.4 DATA REDUCTION AND ANALYSIS

Each data set was reduced in a computer program (Appendix B) to a number of important parameters used to characterize the heat exchanger performance.

4.4.1 Dry Performance

For dry operation the principal computed parameters were the friction factor f or loss coefficient f_o and the Colburn factor j_o .

The loss coefficient is defined as

$$f_o = \frac{2 g_c \rho_m \Delta P_c A_c^3}{\dot{m}_a^2 A_s} \quad (4.1)$$

The mass flow rate is calculated from an empirical formula supplied by the annubar manufacturer:

FIGURE 4.5. Unifin Test Section Assembly

$$\dot{m}_a = 90326 \sqrt{\Delta P_{\text{ann}}} \sqrt{\rho} \quad (4.2)$$

where ρ is computed from the condition of the air at the annubar. By subtracting the expansion and contraction losses, the loss coefficient becomes a standard fanning friction factor.

$$f = f_o - \frac{\sigma^2 A_c N^2 E}{A_s} \quad (4.3)$$

TABLE 4.2. Nominal Test Conditions

	<u>Upstream Air Condition (F)</u>		<u>Primary Temperature (°F)</u>	<u>Primary Flow Rate</u>		<u>Deluge Flow Rate (gpm)</u>	<u>Face Velocity (fpm)</u>	<u>Core Angle</u>
	<u>Dry Bulb</u>	<u>Relative Humidity (%)</u>		<u>(gpm)</u>	<u>(Tube FPS)</u>			
1.	90	50	120	300	4.37	2	270	20
2.	90	50	120	300	4.37	3	270	20
3.	90	50	120	300	4.37	5	270	20
4.	90	25	120	300	4.37	3	270	20
5.(a)	90	75	120	300	4.37	3	270	20
6.(a)	80	75	120	300	4.37	3	270	20
7.	80	50	120	300	4.37	3	270	20
8.(a)	80	25	120	300	4.37	3	270	20
9.	100	25	120	300	4.37	3	270	20
10.(a)	100	50	120	300	4.37	3	270	20
11.(a)	100	75	120	300	4.37	3	270	20
12.	90	50	120	300	4.37	2	180	20
13.	90	50	120	300	4.37	3	180	20
14.	90	50	120	300	4.37	5	180	20
15.	90	50	120	300	4.37	5	360	20
16.	90	50	120	300	4.37	3	360	20
17.	90	50	120	300	4.37	2	360	20
18.(a)	90	50	120	300	4.37	3	360	15
19.	90	50	120	300	4.37	3	180	15
20.(a)	90	50	120	300	4.37	3	270	15
21.	90	50	120	300	4.37	2	270	15
22.	90	50	120	300	4.37	5	270	15
DRY-1(a)	100	NA	120	300	4.37	NA	500	NA
DRY-2	100	NA	120	300	4.37	NA	360	NA
DRY-3	100	NA	120	300	4.37	NA	270	NA
DRY-4(a)	100	NA	120	300	4.37	NA	180	NA
DRY-5(a)	90	NA	120	300	4.37	NA	500	NA
DRY-6	90	NA	120	300	4.37	NA	180	NA

(a) Denotes conditions for which more than one test was run.

where

$$E = \frac{1.75}{\sigma_f^2} - \frac{2.75}{\sigma_f} + 1 \quad (4.3a)$$

and

$$\sigma_f = 1 - \alpha \cdot t_f \quad (4.3b)$$

The Colburn factor is defined as

$$j_o = \frac{h_o}{G_o C_a} Pr^{2/3} \quad (4.4)$$

where the average air-side coefficient h_o is calculated using

$$h_o = \frac{1}{U_o} - \frac{1}{h_p a_p} - \frac{t}{k_t a_t}^{-1} \quad (4.5)$$

The water-side heat transfer coefficient was calculated using the Dittus-Boelter correlation (Welty, Wicks and Wilson 1969) for turbulent flow of a liquid being cooled.

$$h_p = 0.023 \frac{k}{D_h} Re_p^{0.8} Pr_p^{0.3} \quad (4.6)$$

Under nominal operating conditions, the flow rate in the main circulation loop was 300 gpm which gives a velocity on the tubeside of the Unifin core of about 5 fps. At this velocity, the water is fully turbulent and h_p is calculated to be about 1800 Btu/hr-ft²-°F.

The overall heat transfer coefficient in Equation 4.5 was calculated from the measured heat input to the heat exchanger and the measured inlet and outlet temperatures on both the air side and the water side.

$$U_o = \frac{Q}{A_s \Delta T_m} \quad (4.7)$$

The plots of j_o and f , as a function of Re , can be used to calculate the total heat transferred per unit of frontal area as a function of the power expended per unit of frontal area.

The expression used for this calculation is the following (see Appendix E for derivation):

$$\frac{Q}{\Delta T_i A_f} = C_1 C_p \left\{ 1 - \exp \left[- C_2 P^{-n/(3-m)} \right] \right\} [P]^{1/(3-m)} \quad (4.8)$$

where

$$C_1 = \sigma \left[\frac{1000 \mu g_c}{2 L v^2 f_r} \left(\frac{D_h}{1000 \mu} \right)^{(1+m)} \right]^{1/(3-m)} \quad (4.9)$$

$$C_2 = \left(\frac{C_1}{\sigma C_p} \right)^{-n} \left(\frac{1000 \mu}{D_h} \right)^{(1+n)} \left(\frac{4 j_r L Pr^{-2/3}}{1000 \mu} \right) \quad (4.10)$$

P = power per square ft of frontal area; i.e., $\frac{P_{(total)}}{A_f}$

j_o = j-factor based on the air side heat transfer coefficient h_o

j_r = j_o evaluated at $Re = 1000$

f_r = f evaluated at $Re = 1000$

m, n = constants from the fit of f_o and j_o versus Re number; i.e.,

$f = f_r (Re/1000)^{-m}$

$j_o = j_r (Re/1000)^{-n}$

(The expression for j_o contains the influence of fin efficiency). All other terms are included in the nomenclature.

4.4.2 Deluge Performance

Previous analytical work (Hauser et al. 1981) has resulted in the development of a model which describes the heat and mass transfer from a wet-heated surface to a gas stream flowing over the surface. A detailed description of

this model is given in Appendix A. The model uses the heat/mass transfer analogy to relate wet heat transfer to dry heat transfer.

For deluged operation, four principal computed parameters are used to characterize the heat exchanger performance. These are the loss coefficient f_o^* , the overall wet heat transfer coefficient U_o^* , the deluge coefficient h_d^* , and the enhancement ratio Q_w/Q_d .

The loss coefficient is calculated similar to dry operation by using Equation 4.1 and the appropriate data from the deluge tests. A fanning friction factor may also be calculated from Equation 4.3 for deluge operation.

As shown in Appendix A, there are two fundamentally different, although equivalent, methods for calculating the overall heat transfer coefficient. The first method uses the log-mean enthalpy difference (analogous to the log-mean temperature difference) as the driving force to yield the following equation for U_o^* :

$$U_o^* = \frac{C_a Q_{rej}}{A \Delta i_{lm}} \quad (4.11)$$

The second method utilizes the NTU-effectiveness approach assuming a constant tube-side temperature to yield two alternative equations for calculating U_o^* :

$$U_o^* = - \frac{\dot{m}_a^* C_a}{A_s} \ln \left| 1 - \frac{Q_{rej}}{\dot{m}_a^* \Delta i_1} \right| \quad (4.12)$$

and

$$U_o^* = - \frac{\dot{m}_a^* C_a}{A_s} \ln \left| 1 - \frac{\Delta i_\infty}{\Delta i_1} \right| \quad (4.13)$$

Equation 4.12 depends on measurement of the heat rejection and the inlet enthalpy difference, both of which are accurate measurements. Equations 4.11 and 4.13 depend on the air stream enthalpy rise which is much more difficult

to measure precisely. Thus, Equation 4.12 is likely to have the least experimental uncertainty. Theoretically, all three of the above determinations of U_o^* should give the same value. A comparison of the three results gives a good check on the internal consistency of the data.

The model defines the overall heat transfer coefficient in terms of the series thermal resistances

$$\frac{1}{U_o^*} = \frac{\xi}{U_{pt}} + \frac{\xi}{h_d^*} + \frac{1}{h_s a_s^*} \quad (4.14)$$

where $\frac{\xi}{U_{pt}}$ is the primary side resistance including the tube wall

$\frac{\xi}{h_d^*}$ is the resistance due to the deluge film

and $\frac{1}{h_s a_s^*}$ is the resistance of the air

The method for determining each of the variables in Equation 4.14 is presented in Appendix A. The deluge coefficient h_d^* is a film coefficient similar to h_s in accounting for nonuniformities in geometry, flow rates, and surface conditions, as well as the effects of nonuniform wetting.

A more detailed explanation of the computational procedures is given in Appendix C with a sample calculation.

4.5 UNCERTAINTY ANALYSIS SUMMARY

Any experimentally derived value has an uncertainty associated with it. That uncertainty is caused by the inherent uncertainty in the measured variables and parameters used to derive the value. The purpose of an uncertainty analysis is to estimate the probable uncertainty in an experimentally derived value based on the uncertainty of the measurements used to derive it. In this way the expected magnitude of scatter or inconsistency in the experimental results can be estimated. If the actual observed inconsistency is appreciably

greater than that predicted, problems such as the existence of uncontrolled variables may exist in the experimental facility or procedure. This may also indicate the methods being used to analyze the data or the theory to which the results are being compared do not fully account for all the controlling variables. An uncertainty analysis aimed at predicting probable experimental uncertainty is particularly valuable in the WATA test program where the analytical methods being used to predict deluged performance are as yet unproven and may not fully account for all variables.

The probable uncertainty for the WATA test results presented in Section 5 is indicated by bars showing the uncertainty range for representative data points. Where data points are very close together, uncertainty bars are shown on representative points. Probable uncertainties (δR) have been determined using the method suggested by Kline and McClintock (1953) in which:

$$\delta R = \left[\left(\frac{\partial R}{\partial X_1} \delta X_1 \right)^2 + \left(\frac{\partial R}{\partial X_2} \delta X_2 \right)^2 + \dots + \left(\frac{\partial R}{\partial X_n} \delta X_n \right)^2 \right]^{1/2} \quad (4.15)$$

where R = the experimental result, a function of several variables (X_i)

δR = the probable uncertainty in R with 95% confidence limits

δX_i = the uncertainty in X_i where $X_i = \pm \delta X_i$ with 95% confidence limits

and where $\frac{\partial R}{\partial X_i}$ has been evaluated approximately using the expression

$$\frac{\partial R}{\partial X_i} \triangleq \lim_{\Delta X_i \rightarrow 0} \left[\frac{R(X_i + \Delta X_i) - R(X_i)}{\Delta X_i} \right] \quad (4.16)$$

$$\approx \frac{R(X_i + \delta X_i) - R(X_i)}{\delta X_i} \quad (4.17)$$

It should be noted that the determination of values for δX_i to be used in evaluating R requires some judgment on the part of the experimenters. Uncertainties in the individual variables measured in the WATA facility have

been estimated based on claims of instrument manufacturers, observed instrument fluctuation noted from repeated readings under steady-state conditions, and past experience. The major sources of uncertainty have been found to be the measurement of temperature rise across the circulation heater and the measurement of air dewpoint temperature upstream and downstream of the test cores. Uncertainty in the circulation heater temperature rise affects the uncertainty of both dry and deluged results, while dewpoint temperature uncertainty affects only deluged results. A complete calculation of the uncertainty for certain important parameters is given in Appendix D.

5.0 PERFORMANCE EVALUATION

A variety of tests were done to evaluate the performance of the Unifin heat exchanger under both dry and deluged operation. This section will present both pressure drop and heat transfer in several different ways to completely characterize the heat exchanger performance. Particular attention is given to certain key parameters such as psychrometric conditions, air-flow rate, and deluge flow rate. Because of the impact of this work on other parts of the Dry Cooling Enhancement Program, emphasis is given to the comparison of the Unifin surface with other surfaces which have already been tested.

5.1 PRESSURE DROP

Pressure taps 21 in. upstream and 21 in. downstream from the heat exchanger (4 at each location) measured the total pressure drop across the test section, including expansion and contraction losses. Figure 5.1 shows pressure drop as a function of air mass velocity under both dry and deluged operation (3 gpm deluge flow rate). Data points are shown for the Unifin surface only. The Unifin surface has a higher pressure drop under dry operation than either the HÖTERV or Trane surfaces. Although the hydraulic diameter of the Unifin surface is larger than the other two surfaces, the arrangement of tubes causes much larger expansion and contraction losses as the air follows a more tortuous path through the heat exchanger. It is interesting to notice that the shape of the "dry curves" for the Unifin and HÖTERV surfaces are very similar.

For similar reasons, during deluged operation the Unifin surface has a much larger pressure drop than the other two surfaces. Compared to the HÖTERV surface, the pressure drop through the Unifin surface is approximately 0.1 in. of water greater at any mass velocity. Compared to the Trane surface, the pressure drop through the Unifin surface is approximately 0.1 in. of water greater at low flow rates, less than $1000 \text{ lb/ft}^2\text{-hr}$. At higher flow rates the difference becomes much greater. For example, at $1500 \text{ lb/ft}^2\text{-hr}$ the pressure drop through the Unifin surface is 0.25 in. of water greater than the Trane surface, or more than double. The reason for this much larger pressure

FIGURE 5.1. Air-Side Pressure Drop Versus Air Flow Rate

drop under deluged operation is the pronounced breakup into droplets of the deluge water. The Trane surface has continuous straight fins for the deluge water to flow down. On the Unifin surface the deluge water must drop from one fin to the next causing more bridging between fins and much more restricted flow. A similar phenomena occurs with the HÖTERV surface because of the many perforations in the straight fins.

Tests were also done to investigate the effect of the deluge flow rate has on the performance of the Unifin surface. Figure 5.2 shows the pressure drop as a function of deluge flow rate at the different air velocities. The increase in pressure drop is very uniform at about 20% for a change in deluge flow rate from 2.0 gpm to 5.0 gpm.

FIGURE 5.2. Air-Side Pressure Drop Versus Deluge Flow Rate

Friction factors are shown in Figure 5.3 for each of the three surfaces. The curve shown for the Unifin surface is much greater in magnitude than the other two surfaces and seems to level off at higher velocities. The method used to determine f_0 was discussed in section 4.4 and a sample calculation is given in Appendix C.

5.2 DRY HEAT TRANSFER

Heat transfer tests were done for the Unifin surface under dry operation with the primary side temperature fixed at 120°F and ITD's of 20°F and 30°F. The air flow rates were varied between 740 lb/hr-ft² and 2300 lb/hr-ft². The high pressure drop of the Unifin surface prevented testing at higher flowrates due to limited fan capability.

$$\text{Re}(\text{HÖTERV}) = \text{Re}(\text{UNIFIN}) 0.876$$

$$\text{Re}(\text{TRANE}) = \text{Re}(\text{UNIFIN}) 0.669$$

FIGURE 5.3. Friction Factor and Colburn Factor Versus Air Mass Velocity

Three heat transfer coefficients are shown in Figure 5.4 as a function of air mass velocity. They are: U_o , the overall heat transfer coefficient; h_o , the film coefficient with fin efficiency included; and h_s , the film coefficient without the fin efficiency. The uncertainty in U_o was estimated (Appendix D) as 18% and is shown with error bars in Figure 5.4.

FIGURE 5.4. Dry Heat Transfer Coefficients Versus Air Mass Velocity

The film coefficient h_s , excluding the effects of fin efficiency, was used to present the data in the form of a standard Colburn factor, j_s , Figure 5.3. In this form the dry heat transfer performance of the three surfaces appears very similar for the range of Re shown.

Recent work by Johnson et al.^(a) has shown that the most useful method of comparing the performance of various surfaces for dry operation is in terms of heat transfer per ITD per unit area (or volume) versus power. The justification for this type of plot will not be discussed in this report. Given the friction factor and the Colburn factor (Figure 5.3) for each surface as a function of the Reynolds number, the plots shown in Figure 5.5 are generated. These plots give a more informative way of comparing the performance of the three surfaces of interest in this report.

5.3 DELUGED HEAT TRANSFER

A variety of tests were performed with the Unifin surface under deluged conditions. The tests provide a basic understanding of the performance of the Unifin surface as affected by psychrometric conditions of the air, air flow rate, deluge flow rate and the inclination of the test section. Several computed parameters are used to show graphically the dependence of heat exchanger performance on the factors mentioned above. Three of these parameters U_o^* , h_d^* , and Q_w/Q_d were briefly discussed in Section 4.4.2. The fourth parameter, Q/ITD , is shown in Equation A.67, to be represented by the theoretical expression

$$\frac{Q}{ITD} = (\dot{m}_a \phi^* C_a) \Gamma \quad (5.1)$$

where Γ is the nondimensional inlet driving potential

$$\Gamma = \frac{(i_p' - i_\infty)_1}{C_a (T_p - T_\infty)_1} \quad (5.2)$$

(a) In an article to be published in Heat Transfer Engineering.

FIGURE 5.5a. Heat Transfer Per Unit Area Versus Fan Power for Dry Operation

FIGURE 5.5b. Heat Transfer Per Unit Volume Versus Fan Power for Dry Operation

The quantity Γ incorporates the effects of the psychrometric conditions where increasing Γ corresponds to increasing ITD and/or decreasing humidity. The quantity in parentheses in Equation 5.1 is independent of changes in Γ and is thus constant with respect to air psychrometric conditions.

5.3.1 Psychrometric Effects

As just described, theoretically Q/ITD should be a linear function of Γ for tests where $\dot{m}_a$, $\dot{m}_d$, and θ are fixed. Figure 5.6 shows a plot of the data from tests where $\dot{m}_a = 270$ fpm, $\dot{m}_d = 3$ gpm, and $\theta = 20^\circ$. The data supports the theory reasonably well in falling along a straight line. Also shown in Figure 5.6 is a line representing the data from previous tests on the Trane surface which was taken at the same $\dot{m}_a$ and $\dot{m}_d$ but at $\theta = 25^\circ$. Comparing the

FIGURE 5.6. Normalized Heat Transfer Versus Driving Potential

Trane line with the upper Unifin line shows that the Unifin surface gives much higher heat transfer operating at the same air mass velocities. The lower Unifin line shows a lower heat transfer rate at equal pressure drop. What these two lines do not show is that the Unifin surface has a higher pressure drop and a higher heat flux than the Trane surface under these conditions. A more equitable comparison might be the middle Unifin line which is at the same fan power as the Trane data. Consequently, for conditions of equal fan power, the amount of heat transfer by the Unifin surfaces is about 10% higher. At this condition, since the Unifin heat exchanger has 30% less surface area, the heat flux is 40% higher.

Another method of evaluating the performance of a finned surface under deluged operation is shown in Figure 5.7. This is the heat transfer enhancement, deluged compared to dry, at the same pressure drop: Q_w/Q_d . The theoretical expression for Q_w/Q_d is a linear function of Γ (see A.69), which Figure 5.7 supports. Similar to Figure 5.6, the two lines in Figure 5.7 represent data for the Unifin and Trane surfaces which were taken at the same air flow rate. Since the theoretical expression for Q_w/Q_d contains the ratios of

FIGURE 5.7. Enhancement Ratio Versus Driving Potential

the mass flow rate and the effectiveness, both of which are slightly dependent on mass flow rate, the slope of the lines in Figure 5.7 will change slightly at different mass flow rates. At the particular air flow rate at which the data were taken, the enhancement is better for the Unifin surface than for the Trane surface. This indicates that the fins are more uniformly wetted on the Unifin surface yielding a higher heat flux as noted above. Figure 5.7 should not necessarily be interpreted as indicating that the Unifin surface has better performance. Performance of the heat exchangers is better described in terms of heat transfer and pressure drop as shown in other figures (such as Figure 5.6).

Also of interest is the dependence of U_o^* and h_d^* on the psychrometric air conditions. Theoretically both U_o^* and h_d^* are independent of air conditions. The data shown in Figure 5.8 for U_o^* support the theory since U_o^* is essentially independent of the driving force. Although the scatter in the values of h_d^* is quite large, the uncertainty is also quite large and no strongly justifiable conclusions can be made from the data shown in Figure 5.8. The magnitude of U_o^* is nearly a factor of two larger than the corresponding values for the Trane surface. Likewise, the magnitude of h_d^* is about a factor of three larger than the corresponding values for the Trane surface. This supports the contention that the Unifin surface was wetted more uniformly than the Trane surface.

5.3.2 Distribution Effects

Three other parameters affect the performance of heat exchangers during deluged operation primarily by changing the distribution of the deluge water on the surface. The influence of these three parameters, m_a^* , m_d^* , and ϕ are shown in a variety of ways in Figures 5.9 through 5.13. Several interesting observations can be made from these figures.

1. Figure 5.9 indicates that at the higher deluge flow rates (3.0 gpm and 5.0 gpm), intermediate air flow rates around 1200 lb/hr-ft² yield the best surface wetting. At higher air flow rates, the effect of deluge flow rate disappears. Visual inspection of the deluge water distribution during testing confirmed these indications since at the highest air flow rates most of the water was

FIGURE 5.8. Deluged Heat Transfer Coefficients Versus Driving Potential

blown off of the heat exchanger surface downstream, thus the performance would tend to be independent of $\dot{m}_d^*$.

2. Figure 5.10 indicates that the wetting of the surface is better when the heat exchanger is more vertical (15° versus 20°). This is true at all air flow rates. A possible explanation for this is that the configuration of the Unifin surface is such that it cannot hold the water well in the more inclined positions. If this is true, it must be difficult to use this type of surface in an A-frame configuration because the small apex angles which give the best heat transfer performance would leak to high pressure drop from the acceleration/deceleration of the air passing through the A-frame.

FIGURE 5.9. Deluged Heat Transfer Coefficients Versus Air Mass Velocity and Deluge Flow Rate

FIGURE 5.10. Deluged Heat Transfer Coefficients Versus Air Mass Velocity and Angle of Inclination

3. Figure 5.11 supports the conclusions in 1 and 2 by indicating increased wetting at higher deluge flow rates and at smaller angles of inclination. This data was all taken at intermediate air flow rates of about 1200 lb/hr-ft^2 .
4. Figures 5.12 and 5.13 show almost no dependence of Q_w/Q_d on $\dot{m}_d^*$. This would indicate that increasing $\dot{m}_d^*$ increases the pressure drop in direct proportion to the increase in heat transfer. Q_w/Q_d is slightly better at $\theta = 15^\circ$ and at intermediate $\dot{m}_a^* = 1200 \text{ lb/ft}^2\text{-hr}$. This agrees with previous conclusions.

FIGURE 5.11. Deluged Heat Transfer Coefficients Versus Deluge Flow Rate and Angle of Inclination

FIGURE 5.12. Enhancement Ratio Versus Air Mass Velocity and Deluge Flow Rate

FIGURE 5.13. Enhancement Ratio Versus Deluge Flow Rate and Angle of Inclination

5.4 TEST DETAILS

A total of 6 different test conditions dry and 22 different test conditions deluged were done to characterize the performance of the Unifin dry/wet heat exchanger. The important results of these tests are given in Tables 5.1 and 5.2. Several of the tests were done more than once to check repeatability and/or questionable results. These duplicate tests are also listed in Tables 5.1 and 5.2 as X-1, X-2, etc.

The adjustments on the controls in the WATA facility are not sensitive enough to exactly match the desired conditions (see Table 3.1). For example, if the desired test condition for Test #1 were; inlet air dry bulb = 90°F, relative humidity = 50%, primary water temperature = 120°F, face velocity = 270 fpm, etc.; the actual test conditions vary slightly; inlet air dry bulb = 89.8°F, relative humidity = 42.4%, etc. This inability to set the test conditions exactly introduces small uncertainties in the results as was seen in several of the plots of the data shown earlier in this section. Although an uncertainty analysis was done for several important parameters, these "control" uncertainties were not included in the estimates.

Close examination of the results given in Tables 5.1 and 5.2 reveals some interesting details which will be briefly discussed here.

- The heat input is calculated by measuring the flow rate and ΔT through the main heaters (see Section 5.5.2 of Hauser et al. 1981). This value is compared with the change in heat content of the air across the test section. The difference in these two numbers aids in determining the quality of the test. Of course, if everything were measured exactly, these two numbers would be equal. The percentage difference is given in Table 5.2 for the deluged tests. For all tests except two, the percentage difference is negative ranging from -1.0 to -54% with the majority being closer than 20%. Because of the negative bias and because the tests which were repeated generally showed the same percentage difference, the error does not seem to be random but is probably some type of systematic error. Further investigation failed to uncover the cause of this error,

TABLE 5.1. Experimental Values for Dry Operation

Dry Test	Upstream Dry Bulb, °F	Downstream Dry Bulb, °F	Primary Water Temperature, °F	Heat Input, Btu/hr	Drop-Core, in. H ₂ O	Face Velocity		V_{o_s}	h_{o_s}	h_{s_s}	f_o , ND	j_o , ND
						fpm	lb/hr-ft ²	Btu/ hr-ft ² -°F	Btu/ hr-ft ² -°F	Btu/ hr-ft ² -°F		
1-1	100.7	113.0	120.2	85024	0.35	490.9	2033	9.9	11.2	12.9	.057	.0092
1-2	99.7	112.8	120.2	90251	0.35	488.0	2041	10.1	11.5	13.3	.056	.0094
2	99.9	114.0	120.3	71023	0.19	356.1	1475	8.5	9.4	10.6	.058	.0107
3	101.7	115.4	120.2	52582	0.10	267.3	1104	7.5	8.2	9.2	.055	.0125
4-1	103.9	117.0	120.3	32786	0.04	180.3	742	5.7	6.2	6.8	.034	.0140
4-2	101.6	117.0	120.8	39382	0.06	175.6	732	5.9	6.4	7.0	.075	.0147
5	92.6	110.2	120.3	124318	0.35	484.8	2039	10.3	11.7	13.6	.056	.0096
6	89.7	116.6	122.3	67196	0.06	170.8	727	6.2	6.7	7.3	.076	.0155

TABLE 5.2. Experimental Values for Deluged Operation

Test	Upstream Air Condition		Primary Water Temperature, °F	Face Velocity		Heat Input, Btu/hr	Heat Output Airside, Btu/hr	In-Out, In %	Pressure Drop, in. H ₂ O	U _o *	U _o ² *	U _o ³ *	Q _w /Q _d	h _d *
	Dry Bulb, °F	Relative Humidity, %		fpm	lb/hr-ft ²									
1	89.8	42.4	120.4	261.9	1091	486223	448186	-7.8	.34	2.83	2.61	2.36	3.81	63.4
2-1	89.9	47.3	119.7	258.5	1066	505073	430321	-14.8	.36	3.06	2.91	2.36	3.96	99.7
2-2	89.4	43.5	120.2	261.6	1091	530775	453868	-14.5	.36	3.16	2.98	2.43	4.04	118.3
3	89.0	44.3	119.7	261.8	1091	537883	462897	-13.9	.40	3.30	3.11	2.55	3.92	143.1
4	88.5	24.1	119.0	260.4	1099	535079	499008	-6.7	.34	3.11	2.84	2.60	4.20	78.5
5-1	90.0	73.9	119.5	264.0	1082	473870	333016	-29.7	.36	3.16	3.21	2.04	3.76	917.6
5-2	90.7	76.3	120.6	261.4	1084	469553	333700	-28.9	.36	3.11	3.09	2.00	3.68	223.2
5-3	89.2	75.5	120.0	266.0	1098	512810	335272	-34.6	.38	3.37	3.51	2.00	3.81	-NA-
6-1	79.9	72.6	119.7	259.4	1091	518686	437706	-15.6	.35	3.14	3.00	2.41	3.09	122.0
6-2	79.6	76.3	119.5	255.9	1090	511729	431322	-15.7	.36	3.21	3.01	2.41	2.98	117.6
6-A	79.7	75.0	120.1	161.9	688	399082	311388	-22.0	.18	2.48	2.54	1.79	3.13	79.2
7	80.1	50.4	119.5	255.3	1092	524402	494341	-5.7	.34	3.11	2.84	2.63	3.19	80.5
8-1	82.1	31.2	119.4	257.8	1108	535654	524970	-2.0	.36	3.08	2.74	2.68	3.36	65.13
8-2	78.6	26.4	118.7	252.7	1096	574628	533368	-7.2	.36	3.36	3.00	2.74	3.35	95.0
9	101.7	25.1	120.8	265.5	1091	532744	456781	-14.3	.36	3.09	2.91	2.39	6.53	106.7
9-A	96.0	24.5	120.6	165.8	689	411347	361768	-12.1	.18	2.46	2.39	2.01	5.29	62.9
10-1	101.7	49.6	120.9	267.8	1066	436838	344902	-21.0	.36	2.83	2.74	2.04	5.31	85.2
10-2	99.8	49.6	120.0	273.2	1106	463839	350294	-24.5	.39	3.04	3.00	2.10	5.18	130.3
11-1	101.4	75.7	120.7	267.8	1078	381866	196993	-48.4	.36	2.82	3.04	1.34	4.61	222.5
11-2	101.5	74.9	120.4	274.0	1094	416053	189272	-54.5	.37	3.14	3.48	1.30	5.09	-NA-
12	89.6	49.8	120.3	171.6	717	321768	324848	+1.0	.18	1.96	1.80	1.85	3.31	29.2
13	89.4	51.1	120.3	171.7	717	358178	311500	-13.0	.19	2.15	2.09	1.75	3.58	40.9
14	86.4	49.6	120.0	170.6	717	377822	344329	-8.9	.23	2.34	2.22	1.98	3.19	44.5
15	90.6	53.9	120.0	349.8	1454	532164	490552	-7.8	.51	3.23	2.93	2.69	3.68	83.7
16	89.2	51.1	119.8	351.1	1461	528855	512877	-3.0	.47	3.19	2.87	2.77	3.62	75.8
17	90.1	50.7	119.7	352.6	1466	527331	491452	-6.8	.44	3.18	2.93	2.67	3.85	84.4
18-1	91.3	50.5	119.3	350.5	1459	545948	464578	-14.9	.44	3.37	3.15	2.57	4.21	131.4
18-2	90.4	49.9	119.9	350.8	1464	553868	494133	-10.8	.43	3.30	3.05	2.65	4.08	110.7
19	90.3	52.1	120.0	177.4	739	402881	314691	-21.9	.22	2.55	2.57	1.83	3.93	75.4
20-1	91.0	50.8	120.2	263.1	1097	518445	417602	-19.5	.34	3.19	3.07	2.32	4.25	168.2
20-2	91.3	50.3	120.1	261.7	1091	511427	414255	-19.0	.34	3.15	3.05	2.32	4.26	151.7
21-1	90.7	48.4	120.4	262.0	1095	418880	478742	+14.3	.30	2.82	2.67	2.26	4.07	73.4
22	90.7	49.3	119.7	259.7	1084	554581	422928	-23.7	.38	3.50	3.46	2.39	4.37	-NA-

although it is likely an error in dewpoint measurement. Tests 11-1 and 11-2 were included in the various plots of the data but must be highly suspect as to their validity. to uncover the cause of this error. Tests 11-1 and 11-2 were included in the various plots of the data but must be highly suspect as to their validity.

- The pressure drops listed in Table 5.1 at the lowest velocities (Test 4-1, 4-2, and 6) are very uncertain because of the difficulty in reading the manometer at these low values. Separate tests were done to measure the pressure drop as a function of air flow rate used in computing the friction factors shown in Figure 5.3.
- The results of these tests indicate that the model used to calculate h_d^* (Appendix A) may not be entirely satisfactory. The film coefficient accounting for the heat transfer to the air is evidently enhanced by the water droplets.

6.0 REFERENCES

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APPENDIX A

FORMULATION OF A HEAT/MASS TRANSFER MODEL

APPENDIX A

FORMULATION OF A HEAT/MASS TRANSFER MODEL^(a)

A.1 ABSTRACT

An approximate analytical model is described that may be used to calculate the rates of heat transfer and evaporation from the surface of a wet finned heat exchanger. The model is analogous to the conventional expression for convective heat transfer. However, the driving potential is an enthalpy difference instead of a temperature difference. The overall heat transfer coefficient can be computed in the usual manner using a simple transformation of variables. The value of the model is that wet heat exchanger performance can be estimated accurately using dry performance data.

A.2 BACKGROUND AND INTRODUCTION

Most of the power plants that have been built in recent years and that are planned for the near future will be equipped with evaporative cooling systems (or wet towers) for waste heat rejection. Evaporative cooling technology is a proven, versatile, reliable, and economical option. Therefore, it is likely to remain as the prime means for large-scale heat rejection applications in most regions of the world for the foreseeable future. However, in arid regions where water supplies are limited, or in other areas where the growing demand for water from all sectors of society is approaching the available supply, alternative cooling systems must be considered.

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A.2.1 Alternatives to Wet Cooling

The only practical alternatives to wet towers for the foreseeable future are dry-cooled systems and various dry/wet combinations and hybrids. Dry cooling adds markedly to the cost of power produced when compared to a conventional wet-cooled plant to which water is effectively free. This is not only because of the high capital cost, but also because of the inability to meet peak demand during warm weather periods. On purely economic grounds, power plants could generally afford to pay more for water than competing users. However, the pressure to reduce water consumption for power plant cooling is likely to be more political than economic because of the volatile and highly politicized nature of water rights and water allocation issues.

In locations where limited quantities of water are available for power plant cooling, studies (Loscutoff 1976; Braven et al. 1978; Wiles et al. 1978) have shown that combinations of dry-cooled and evaporatively cooled plants may be the answer before totally dry-cooled plants are required. By using dry/wet cooling, with wet cooling for peaking only, the consumption of water can be greatly reduced and savings may be realized due to a net reduction in the "size" of the cooling that has been referred to here as the "deluge" concept, the performance of a normally dry-cooled heat exchanger is enhanced during periods of high ambient temperature and/or high cooling demand by "deluging" the surface with water. The air flowing through the system causes evaporation from the delugeate film driving the air/water interface temperature toward the local wet bulb temperature and thereby increasing substantially the rate of heat rejection.

The primary potential benefits of deluge vis-à-vis all-dry cooling are a combination of the following:

- The condenser temperature (and thus the turbine back pressure) can be maintained lower than would be possible with an all-dry system under the same ambient conditions.
- The heat rejection capacity of the system is increased (or the "size" of the required system is reduced), compared to an all-dry system under similar conditions.

As a consequence of the above and other factors, the cost of building and operating a dry-cooled system with evaporative peaking can be substantially less than that of an all dry-cooled system of the same capacity, with only minimal consumption of water.

A.2.2 Previous Work on Dry/Wet Cooling

When this study began, the information available on dry/wet cooling was extremely limited. The only known analytical work on deluge cooling at that time was a brief paper by Tasnadi (1972), which was translated from Hungarian. A limited amount of proprietary data were also made available by the HOTERV Institute in Hungary from tests performed on a Heller-Forgo plate fin heat exchanger. A subsequent literature review also identified an earlier analysis by Threlkeld (1962) that used a similar approach for calculating heat/mass transfer in finned air chillers with condensation. Although the general approach used in both of the preceding analyses was similar to that employed here (i.e., use of the enthalpy driving potential), the present work has expanded the theoretical basis of the model and has substantially increased its generality and applicability.

Some of the early developments of the present work were reported in the final report of a project in which several heat exchangers were tested in dry and dry/wet operation (Parry et al. 1979) and in two papers presented at meetings (Kreid et al. 1978; Kreid et al. 1979). New information and insights gained in the course and subsequent experiments performed with a wavy plate fin heat exchanger (Hauser et al. 1980) were instrumental in development of the recent improvements in the predictive accuracy of the model. In particular, the heat transfer analogy was extended to allow calculation of the mass transfer at the surface. A detailed development of the refined and extended deluge model analysis is given by Hauser et al. (1980). A summary of the results of that analysis will be given here.

A.3 ANALYSIS

The approximate formulation referred to here as the deluge model is based on the physical similarity and resulting mathematical analogies between heat

transfer and mass transfer. To highlight the similarities, an outline of the analogous steps in the parallel developments of equations for dry convective heat transfer, wet convective/evaporative heat transfer, and mass transfer will be presented in tables such as Table A.1.

In the analysis, mass/energy balances are first performed on a differential surface area, from which differential equations for the distribution of temperature, enthalpy, and humidity are derived. These equations are then extended to include effects of internal thermal resistance below the surface. Finally, the "local" differential equations are integrated across the depth of a wet, finned tube condenser to obtain expressions for the heat transfer and evaporation rates and equations for the distributions of temperature, enthalpy, and humidity in the system.

A.3.1 The Surface Heat/Mass Flux Analogy

The analysis of the heat and mass transfer from an element of wetted surface is based on the control volume shown in Figure A.1. Equations (A.1), (A.2), (A.3), and (A.4) in Table A.1 contain expressions for the energy and mass balances for the control volume for dry and wet operations.

Equation (A.2) is based on the assumption that the processes of heat transfer and mass transfer are separable. The first term on the right in Equation (A.2) is the sensible heat transfer which is assumed to be given by the equation for dry heat transfer at the same air side conditions [i.e., by Equation (A.1)]. The second term on the right in Equation (A.2) is the latent heat term given by the product of the evaporation rate (assumed to be governed by diffusion, as given by Fick's law) times the latent heat of evaporation at the surface. Evaporation is assumed to be diffusion limited so that the surface air may be assumed saturated at the surface temperature.

For water vapor at the temperature and pressures of interest in this situation, it may be assumed that $Le \approx 1$, where the convective Lewis number is defined by

$$Le = \frac{h_s}{\sigma_s C_a} \approx 1 \quad (A.13)$$

TABLE A.1. Summary of Equations for the Surface Mass/Energy Balance

<u>Dry Heat Transfer</u>	<u>Wet Heat Transfer</u>	<u>Wet Mass Transfer</u>
$\frac{dQ}{dA_s} = h_s (T_s - T_\infty) \quad (A.1)$	$\frac{dQ}{dA_s} \approx h_s (T_s - T_\infty) + \lambda_s \sigma_s (H'_s - H_\infty) \quad (A.2)$ $\approx \sigma_s [C_a (T_s - T_\infty) Le + \lambda_s (H'_s - H_\infty)] \quad (A.3)$	$\frac{d\dot{m}_w}{dA_s} = \sigma_s (H'_s - H_\infty) \quad (A.4)$
	$\frac{dQ}{dA_s} \approx \frac{h_s}{C_a} (i'_s - i_\infty) \quad (A.5)$	$\frac{d\dot{m}_w}{dA_s} \approx \frac{h_s}{C_a} (H'_s - H_\infty) \quad (A.6)$
$\frac{dQ}{dA_f} = G_a C_a dT_\infty \quad (A.7)$	$\frac{dQ}{dA_f} \approx G_a di_\infty \quad (A.8)$	$\frac{d\dot{m}_w}{dA_f} \approx G_a dH_\infty \quad (A.9)$
$\frac{dT_\infty}{(T_s - T_\infty)} = \left(\frac{h_s D}{G_a C_a \delta} \right) dx \quad (A.10)$	$\frac{di_\infty}{(i'_s - i_\infty)} = \left(\frac{h_s D}{C_a G_a \delta} \right) dx \quad (A.11)$	$\frac{dH_\infty}{(H'_s - H_\infty)} = \left(\frac{h_s D}{G_a C_a \delta} \right) dx \quad (A.12)$

FIGURE A.1. Control Volume, General Boundary Layer

With this assumption, the term in brackets on the right in Equation (A.3) is very nearly equal to the difference between the enthalpy of moist air at the surface and free stream conditions. Equations (A.5) and (A.6) then follow directly from Equations (A.3) and (A.4) with the assumption $Le \approx 1$.

The mass/energy balances for the air stream are given in Equations (A.7), (A.8), and (A.9). When combined with Equations (A.5), and (A.6), these yield analogous differential equations for the distribution of temperature on a dry surface [Equation (A.10)] and the distributions of enthalpy and humidity on a wet surface [Equations (A.11) and (A.12), respectively]. In principle, Equations (A.10), (A.11), and (A.12) may be integrated if the relevant variables can be given as functions of the dimensionless distance X .

A.3.2 Extension to Finned Surfaces

The surface heat/mass transfer analysis has been extended to the treatment of heat transfer from finned surfaces, as illustrated by Figure A.2, with introduction of the overall heat and mass transfer coefficients. The results of the analysis are summarized in Table A.2. Equations (A.14), (A.15), and (A.16) are the analogous equations for heat/mass transfer based on overall coefficients and overall driving potentials from the primary (tube) side to

FIGURE A.2. Representative Segment of a Plate Fin Assembly

the free stream (air side) conditions. The analogous expressions assumed for U_o , U_o^* , and $U_m^* C_a \Sigma_o^*$ are given in Equations (A.17), (A.18), and (A.19), respectively.

The thermal resistance $1/U_o$ is given by the sum of the net inside thermal resistance $1/U_{pt}$ plus the thermal resistance of the finned surface $1/h_s a_s$). The relative effective surface area a_s is given by

$$a_s = \frac{A_{st} + \eta_f A_{sf}}{A_{st} + A_{sf}} \quad (\text{A.32})$$

The fin efficiency is defined in the conventional manner where the actual fin area A_{sf} is replaced by an equivalent area $\eta_f A_{sf}$, which, when assumed to

TABLE A.2. Summary of Model Development for Extended Surface

Dry Heat Transfer	Wet Heat Transfer	Wet Mass Transfer
$\frac{dQ}{dA_s} = U_o(T_p - T_\infty) \quad (A.14)$	$\frac{dQ}{dA_s} = U_o^* \left(\frac{i'_p - i_\infty}{C_a} \right) \quad (A.15)$	$\frac{d\dot{m}_w}{dA_s} = \Sigma_m^* (H'_p - H_\infty) \quad (A.16)$
$U_o = \left[\frac{1}{U_{pt}} + \frac{1}{h_s a_s} \right]^{-1} \quad (A.17)$	$U_o^* = \left[\xi \left(\frac{1}{U_{pt}} + \frac{1}{h_d a_s^*} \right) + \frac{1}{h_s a_s^*} \right]^{-1} \quad (A.18)$	$C_a \Sigma_m^* = \left[\xi_m \left(\frac{1}{U_{pt}} + \frac{1}{h_d a_s^*} \right) + \frac{1}{h_s a_m^*} \right]^{-1} \quad (A.19)$
$U_{pt} = \left[\frac{1}{h_p a_p} + \frac{t}{k a_t} \right]^{-1} \quad (A.20)$	$\xi = \frac{(i'_p - i'_r)}{C_a (T_p - T_r)} \quad (A.21)$	$\xi_m \approx \xi \quad (A.22)$
$\eta_f = f \left(\frac{r_i}{r_o}, Bi_f^{1/2} \right) \quad (A.23)$	$\eta_f^* = f \left(\frac{r_i}{r_o}, Bi_f^{*1/2} \right) \quad (A.24)$	$\eta_m^* = f \left(\frac{r_i}{r_o}, Bi_m^{*1/2} \right) \quad (A.25)$
$Bi_f = \frac{l_f^2}{ky_b} h_s \quad (A.26)$	$Bi_f^* = \frac{l_f^2}{ky_b} \left[\frac{1}{h_d} + \frac{1}{\xi h_s} \right]^{-1} \quad (A.27)$	$Bi_m^* = \frac{l_f^2}{ky_b} \left[\frac{1}{h_d} + \frac{1}{\xi_m h_s} \right]^{-1} \quad (A.28)$
$\frac{dT_\infty}{(T_p - T_\infty)} = \left(\frac{U_o A_s}{\dot{m}_a C_a} \right) dx \quad (A.29)$	$\frac{di_\infty}{(i'_p - i_\infty)} = \left(\frac{U_o^* A_s}{\dot{m}_a C_a} \right) dx \quad (A.30)$	$\frac{dH_\infty}{(H'_p - H_\infty)} = \left(\frac{\Sigma_o^* A_s}{\dot{m}_a} \right) dx \quad (A.31)$

be uniformly at the fin root temperature T_r , will dissipate the same amount of heat as the actual finned surface. The fin efficiency can be given by a function [Equation (A.23)] or a graph (Figure A.3) as a function of the heat exchanger geometry and operating conditions. The function given in Figure A.3 for cylindrical fins may also be used for plate fins if the equivalent fin length l_f and equivalent radius r_o are computed such that the resulting area is equal to the actual area of the plate fin configuration.

Equation (A.18) for U_o^* is analogous to Equation (A.17) for U_o . However, an additional thermal resistance has been added to account for the deluge water film ($1/h_d a_s^*$) and the sum of the resistances from the tube side to the air/water interface has been multiplied by a resistance transformation parameter, ξ .

FIGURE A.3. Fin Efficiency Solution for an Annular Fin (also used to model plate fin with equivalent radius)

The deluge film convective heat transfer coefficient h_d is an effective or "lumped" parameter that incorporates the effects of nonuniform water flow, air flow, and surface wetness. The technique employed for determining this parameter from experimental data is described in the subsequent discussion.

The relative effective surface area for wet operation a_s^* is analogous to a_s . The wet fin efficiency may be computed from the function or graph for the dry efficiency by the transformation of variables given by Equations (A.24) and (A.27). Detailed development and discussion of the methods described here for evaluation of the wet surface efficiency η_f^* are provided by Hauser et al. (1980).

Transformation of the inside thermal resistances is required to match the internal expression for Q based on the temperature driving potential to the surface heat flux based on the enthalpy potential. The heat transfer through the sum of the thermal resistances up to the fin root (including the effective resistance of the deluge water film) can be given by

$$\frac{dQ}{dA_s} = \frac{(T_p - T_r)}{\left(\frac{1}{h_p a_p} + \frac{t}{k a_t} + \frac{1}{h_d a_s^*} \right)} \quad (A.33)$$

$$= U_{pr} (T_p - T_r) \quad (A.34)$$

Converting this to the equivalent enthalpy difference, we obtain the needed transformation parameter ξ as follows:

$$U_{pr}(T_p - T_r) = U_{pr} \left[\frac{C_a(T_p - T_r)}{(i_p' - i_r')} \right] \left(\frac{i_p' - i_r'}{C_a} \right) \quad (A.35)$$

$$\frac{dQ}{dA_s} = \frac{U_{pr}}{C_a} (i_p' - i_r') \quad (A.36)$$

where the resistance transformation parameter ξ is defined by Equation (A.21). A physical interpretation for this parameter and means for its evaluation are provided in the next section.

An overall mass transfer coefficient, $U_m^* = C_a \Sigma_m^*$, is defined by Equation (A.19) in a manner completely analogous to the expression for U_o^* , Equation (A.18). It is assumed that the actual internal resistance to heat transfer can be converted to an "effective" resistance to mass transfer by multiplication with a resistance transformation parameter ξ_m . This is completely analogous to the transformation wherein the equation for U_o was obtained from the expression for U_o . The efficiency η_m^* and area A_m^* are then determined by the transformation of variables given by Equations (A.25) and (A.28). In many cases it may be assumed that $\xi_m \approx \xi$ and thus $\eta_m^* \approx \eta_f^*$, which greatly simplifies the model. A general rigorous technique for computing ξ_m is described briefly in a subsequent discussion.

When the equations for dry surface heat flux [Equation (A.14)] and wet surface heat/mass flux [Equations A.15) and (A.16)] are equated to the respective heat/mass gains of the air [Equations (A.1), (A.5), and (A.6)], expressions [Equations A.29), (A.30), and (A.31)] are obtained that describe the air property variations with distance in the airflow direction. For suitable conditions these expressions can be integrated to obtain the distributions of temperature, enthalpy, and humidity in a heat exchanger.

A.3.3 Analysis of a Wetted Plate-Fin Condensor

The expressions in Equations (A.29), (A.30), and (A.31) will now be integrated for a plate fin condensor in crossflow. The principal assumptions are:

- The primary or tube side temperature is constant.
- The air mass flux G_o is uniform in any plane normal to the flow.
- The NTU parameters N , N^* and N_m^* are constant.

The results of the integrations are summarized in Table A.3.

Equations (A.40), (A.41), and (A.42) give the distributions of the air temperature in a dry heat exchanger and the air enthalpy and humidity in a wet heat exchanger. In addition, using the computed values of i_∞ , H_∞ , and psychrometric charts, the distribution of the dry bulb temperature in a wet core can also be predicted.

The total heat/mass transfer from the heat exchanger is obtained by a second integration where the results are given in terms of the effectiveness

TABLE A.3. Summary of Model Equations for a Deluged Condenser in Crossflow

Dry Heat Transfer	Wet Heat Transfer	Wet Mass Transfer
$\theta(x) = \frac{T_{p1} - T_{\infty}(x)}{T_{p1} - T_{\infty 1}} \quad (A.37)$	$\theta^*(x) = \frac{i'_{p1} - i'_{\infty}(x)}{i'_{p1} - i'_{\infty 1}} \quad (A.38)$	$\theta_m^*(x) = \frac{H'_{p1} - H(x)}{H'_{p1} - H_{\infty 1}} \quad (A.39)$
$= e^{-Nx} \quad (A.40)$	$= e^{-N^*x} \quad (A.41)$	$= e^{-N_m^*x} \quad (A.42)$
$N = \frac{U_o A_s}{\dot{m}_a C_a} \quad (A.43)$	$N^* = \frac{U_o^* A_s}{\dot{m}_a C_a} \quad (A.44)$	$N_m^* = \frac{\Sigma_o^* A_s}{\dot{m}_a} \quad (A.45)$
$\dot{Q}_o = \dot{m}_a C_a \phi (T_{p1} - T_{\infty 1}) \quad (A.46)$	$\dot{Q}_o = \dot{m}_a \phi^* (i'_{p1} - i'_{\infty 1}) \quad (A.47)$	$\dot{m}_o = \dot{m}_a \phi_m^* (H'_{p1} - H_{\infty 1}) \quad (A.48)$
$\phi = 1 - e^{-N} \quad (A.49)$	$\phi^* = 1 - e^{-N^*} \quad (A.50)$	$\phi_m^* = 1 - e^{-N_m^*} \quad (A.51)$
$\dot{Q}_o = U_o A_s \Delta T_{lm} \quad (A.52)$	$\dot{Q}_o = U_o^* A_s \frac{\Delta i_{lm}}{C_a} \quad (A.53)$	$\dot{m}_o = \Sigma_o^* A_s \Delta H_{lm} \quad (A.54)$
$\Delta T_{lm} = \frac{(T_p - T_{\infty})_2 - (T_p - T_{\infty})_1}{\ln \left \frac{(T_p - T_{\infty})_2}{(T_p - T_{\infty})_1} \right } \quad (A.55)$	$\Delta i_{lm} = \frac{(i'_p - i'_{\infty})_2 - (i'_p - i'_{\infty})_1}{\ln \left \frac{(i'_p - i'_{\infty})_2}{(i'_p - i'_{\infty})_1} \right } \quad (A.56)$	$\Delta H_{lm} = \frac{(H'_p - H_{\infty})_2 - (H'_p - H_{\infty})_1}{\ln \left \frac{(H'_p - H_{\infty})_2}{(H'_p - H_{\infty})_1} \right } \quad (A.57)$

ϕ , ϕ^* and ϕ_m^* . The analogous expressions for heat and mass transfer are given by Equations (A.46), (A.47), and (A.48) where the corresponding expressions for the effectiveness are given by Equations (A.49), (A.50), and (A.51). The advantage to this approach is that the heat/mass transfer is given in terms of the inlet conditions without the need of the outlet properties.

Equations (A.29), (A.30), and (A.31) can also be integrated to obtain alternate equivalent expressions for the heat and mass transfer, Equations (A.52), (A.53), and (A.54). A disadvantage of this approach is that both inlet and outlet conditions appear in the log mean property differences defined by Equations (A.55), (A.56), and (A.57). This generally requires an iterative solution technique. In addition, a crossflow correction would normally be required.

In principle, similar results could be obtained for other heat exchanger geometries. However, in most cases, the results would probably be more complex and, in many cases, closed form solutions could probably not be found. However, in these cases, existing numerical solutions or graphs of $\phi(R, N)$ for dry operation can be used to compute $\phi^*(R^*, N^*)$ and $\phi_m^*(R_m^*, N_m^*)$ by transformation of variables. The capacity ratio $R = \dot{M}_a C_a / \dot{M}_p C_p$ can be transformed to R^* and R_m by multiplication with ξ or ξ_m , respectively. The solution for ϕ in a condenser is a special case, where the dependence on R cancels out.

A.3.4 Evaluation T_r , ξ , and h_d

To apply the deluge model for prediction of heat/mass transfer, values of ξ and h_d must be specified. An explicit expression [Equation (A.21)] has been derived for ξ . However, precise evaluation of ξ requires knowledge of the root temperature T_r , which varies with core design and operating conditions in a manner that is not easily predicted.

When all relevant operating and inlet conditions are known, including h_d , the value of the average or effective root temperature T_r can be estimated. Since the heat that is transferred from the system passes through a series resistance path, the "local" rate of heat transfer can be given by the following equivalent expressions:

$$\frac{dQ}{dA_s} = \frac{\left(\frac{i'_p - \bar{T}_r}{C_a} \right)}{\xi \left(\frac{1}{h_p a_p} + \frac{t}{k a_t} + \frac{1}{h_d a_s^*} \right)} = U_{pr} (T_p - T_r) \quad (A.58)$$

$$\frac{dQ}{dA_s} = h_s a_s^* \left(\frac{i'_r - \bar{T}_\infty}{C_a} \right) \quad (A.59)$$

where U_{pr} is the inverse of the net internal resistance

$$U_{pr} = \left(\frac{1}{h_p a_p} + \frac{t}{k a_t} + \frac{1}{h_d a_s^*} \right)^{-1} \quad (A.60)$$

Equation (A.58) gives the heat transfer from the tube side to the root and Equation (A.59) gives the heat transfer from the root to the air. Equations (A.58) and (A.59) constitute a set of two simultaneous equations that can be solved for the two unknown dQ/dA_s and T_r . Unfortunately, these equations cannot be solved explicitly because of the complex dependence of a_s^* on T_r . Thus, an iterative solution is required.

In analyzing the experimental results, both h_d and T_r were unknown, but the heat transfer from the system Q_0 was measured. If we assume the average value of dQ/dA_s is Q_0/A_s and use the measured average value of T_∞ , Equations (A.58) and (A.59) can be solved simultaneously to obtain average values of T_r and h_d for each test.

The value of ξ is determined in the iterative solution for T_r . The results of this computation for the tests performed with the wavy plate fin heat exchanger (Hauser et al. 1980) are shown in Figure A.4. It is apparent that ξ depends rather dramatically on the operating conditions and especially on the primary fluid temperature T_p . It is thus important that this variable be evaluated as accurately as possible.

FIGURE A.4. Computation of ξ as a Function of Representative Operating Conditions

A.3.5 Development of an Expression for ξ_m

To calculate ξ , we take the ratio of the local heat transfer to the local mass transfer on a wet finned surface as follows:

$$\frac{\dot{dQ}}{d\dot{m}} = \frac{U_{pr} A_s}{\xi} \left(\frac{i_p' - i_r'}{c_a} \right) \bigg/ \frac{U_{pr} A_s}{\xi_m} \left(\frac{H_p' - H_r'}{c_a} \right) \quad (A.61)$$

$$= \frac{\xi_m}{\xi} \left(\frac{i_p' - i_r'}{H_p' - H_r'} \right) \quad (\text{A.62})$$

From Equations (A.46) and (48) the ratio of overall heat transfer to mass transfer can be given in terms of the effectiveness, as follows:

$$\frac{Q}{\dot{m}} = \frac{\phi^*(i_p' - i_\infty)_1}{\phi_m^*(H_p' - H_\infty)_1} \quad (\text{A.63})$$

Assuming average values are used in Equation (A.62) for properties, Equations (A.62) and (A.63) may be equated to yield

$$\frac{\xi_m \phi_m^*}{\xi \phi^*} = \left(\frac{H_p' - H_r'}{i_p' - i_r'} \right) / \left(\frac{H_p' - H_\infty}{i_p' - i_\infty} \right)_1 = \delta \quad (\text{A.64})$$

The ratio of property differences defined by δ can be evaluated for a given case by techniques described in the preceding analysis. An iterative solution can then be used to compute ξ/ξ_m .

For conditions where $\xi/\xi_m \approx 1$ and $\phi^*/\phi_m^* \approx 1$, an approximate closed form solution can be obtained by expansion of Equation (A.64) in a series. The result may be given as follows:

$$\frac{\xi_m}{\xi} \approx 1 - \left(\frac{1-\delta}{1-K} \right) \quad (\text{A.65})$$

where K is defined by

$$K = \frac{\xi_m U_m^*}{U_{pr}} \left(\frac{N^*}{e^{N^*} - 1} \right) \quad (\text{A.66})$$

The relevant derivations are given by Hauser et al. (1980). Evaluation of these results for conditions relevant to cooling tower operation indicates

that ξ_m and ξ may be assumed equal with an accuracy of about $\pm 10\%$. This is probably within the overall accuracy of the deluge model formulation.

A.3.6 Prediction of Heat Transfer Correlations

The problem of presenting wet heat transfer data is more complicated than for dry heat transfer because several additional variables must be accounted for. A number of means were devised for handling and presenting the data that are unique to the case of wetted heat exchangers. The theoretical bases for these correlations are provided by Hauser et al. (1980).

One of the standard quantities used in characterizing heat exchanger performance is the heat transfer per unit ITD. This parameter may be computed using the deluge model to obtain a simple correlation in terms of the normalized driving potential for heat transfer. The result is

$$\frac{Q}{M_a C_a (T_p - T_\infty)_1} = \phi^* \Gamma \quad (\text{A.67})$$

where the normalized inlet driving potential for heat transfer Γ is defined by

$$\Gamma = \frac{(i_p' - i_\infty)_1}{C_a (T_p - T_\infty)_1} \quad (\text{A.68})$$

This parameter is a purely thermodynamic variable that depends only on the inlet conditions (T_p, ITD, ϕ) . Values of Γ as a function of conditions for a range of variables of interest in this study are presented in Figure A.5.

Another similar correlation can be derived using the primary to fin root temperature difference. The theoretical correlation in this case is

$$\frac{(T_p - T_r)}{(T_p - T_\infty)_1} = \left(\frac{C_a m_a \phi^*}{U_{pr} A_s} \right) \Gamma$$

FIGURE A.5. Evaluation of the Inlet Driving Potential Ratio for Representative Operating Conditions

There are a number of potential comparisons between wet and dry heat transfer that could be made to assess the enhancement due to deluge. One of the most useful comparisons has been found to be the ratio of wet to dry heat transfer for the same core temperature and inlet air conditions and the same air-side pressure drop. In this case, the predicted condition is

$$\frac{Q_w}{Q_d} = \frac{m_a^* \phi^* (i_p - i_\infty)_1}{m_a \phi C_a (T_p - T_\infty)_1} = \left(\frac{m_a^*}{m_a} \right) \left(\frac{\phi^*}{\phi} \right) \Gamma \quad (\text{A.69})$$

The heat rejected by a wetted heat exchanger can be given as the sum of the heat required to evaporate the water (the latent heat) and that which raises the temperature of the moist air (the sensible heat). The ratio of the latent heat to the total heat transfer Q_v/Q_o can be obtained from the deluge model. The result is

$$\frac{Q_v}{Q_o} = \left(\frac{\phi_m^*}{\phi^*} \right) \Gamma_m \approx 1 - \left(\frac{K(\delta - 1)}{(\delta - K)} \right) \Gamma_m \quad (\text{A.70})$$

where Γ_m is defined by

$$\Gamma_m = \frac{\lambda_s (H'_p - H_\infty)}{(i'_p - i_\infty)} \quad (\text{A.71})$$

The parameter Γ_m is the ratio of the driving potential for mass transfer to the driving potential for heat transfer on a wet surface. This parameter is analogous to Γ and has been evaluated in a similar manner. The results are plotted in Figure A.6.

All of these techniques were used to reduce and correlate the data in the Trane core experiments (Hauser et al. 1980).

A.4 CONCLUSIONS

Approximate models have been developed for predicting the rates of heat transfer and evaporation from the surfaces of a wet finned heat exchanger. The models are based on the physical similarity that exists between heat and mass transfer phenomena and the mathematical analogies that result.

The potential value of this technique is that by means of a simple transformation of variables, predictions of both heat transfer and evaporation

FIGURE A.6. Calculation of Γ_m as a Function of Representative Operating Conditions

rates from a wetted heat exchanger can be made using conventional dry surface heat transfer formulations. In addition, the models are based primarily on heat transfer performance data obtained for the same surface operated dry.

The primary difficulty in using these models is that they are dependent upon the effective deluge film heat transfer coefficient, h_d . No satisfactory means have been identified for predicting this parameter from either

first principles or from existing empirical correlations. It thus appears that direct experimental determinations of h_d are required to fully utilize the predictive capabilities of the model. Once a number of heat exchanger types have been tested for a representative range of conditions, suitable correlations for h_d might be developed.

A.5 REFERENCES

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APPENDIX B

DATA ANALYSIS PROGRAM

APPENDIX B

DATA ANALYSIS PROGRAM

GLOSSARY FOR DATA ANALYSIS PROGRAM

AC	core frontal area, ft^2
AMFR	air mass flow rate, lb/hr
ANDLP	annubar pressure drop, in. H_2O
AP	reduced mean tube wall area
AS	total surface area - airside, ft^2
ASX	reduced surface area for heat transfer, air side
AV	frontal air velocity, ft/s
BETA	critical temperature of water - T , $^{\circ}\text{R}$
BX	(Biot Number) $^{1/2}$ for heat transfer
CA	average moist heat capacity based on dew point data, $\text{Btu/lbm}^{\circ}\text{F}$
CA1	upstream moist heat capacity based on dew point data, $\text{Btu/lbm}^{\circ}\text{F}$
CAA	average moist heat capacity based on relative humidity data, $\text{Btu/lbm}^{\circ}\text{F}$
CA1A	upstream moist heat capacity based on relative humidity data, $\text{Btu/lbm}^{\circ}\text{F}$
DELH	log mean enthalpy difference based on dew point data, Btu/lbm
DELHA	log mean enthalpy difference based on relative humidity data, Btu/lbm
DELTA	core pressure drop, in. H_2O
DH1	$i'_{p1} - i_{\infty 1}$, based on dew point data, Btu/lbm
DH1A	$i'_{p1} - i_{\infty 1}$, based on relative humidity data, Btu/lbm
DH2	$i'_{p2} - i_{\infty 2}$, based on dew point data, Btu/lbm
DH2A	$i'_{p2} - i_{\infty 2}$, based on relative humidity data, Btu/lbm
DTFC	temperature drop across core, $^{\circ}\text{F}$
DTFH	temperature gain across heaters, $^{\circ}\text{F}$
DTMVC	temperature drop across core, mV
DTMVH	temperature gain across heaters, mV

ETAF	experimental fin efficiency for heat transfer
ETAFX	predicted fin efficiency for heat transfer
GAMA	nondimensional inlet driving potential based on dew point data
GAMAA	nondimensional inlet driving potential based on relative humidity data
H1	$i_{\infty 1}$ inlet air enthalpy, dew point data, Btu/lbm
H1A	$i_{\infty 1}$ inlet air enthalpy, relative humidity data, Btu/lbm
H2	$i_{\infty 2}$ outlet air enthalpy, dew point data, Btu/lbm
H2A	$i_{\infty 2}$ outlet air enthalpy, relative humidity data, Btu/lbm
H12	i'_{p1} saturated air enthalpy at inlet water temperature, Btu/lbm
H22	i'_{p2} saturated air enthalpy at outlet water temperature, Btu/lbm
HA	air side surface heat transfer coefficient at same pressure as wet test, $\text{Btu/hr-ft}^2\text{-}^\circ\text{F}$
HD	deluge coefficient, $\text{Btu/hr-ft}^2\text{-}^\circ\text{F}$
HDX	apparent surface heat transfer coefficient, $\text{Btu/hr-ft}^2\text{-}^\circ\text{F}$
HE	effective surface heat transfer coefficient, $\text{Btu/hr-ft}^2\text{-}^\circ\text{F}$
HP	primary side heat transfer coefficient, $\text{Btu/hr-ft}^2\text{-}^\circ\text{F}$
HS	surface heat transfer coefficient, air side, $\text{Btu/hr-ft}^2\text{-}^\circ\text{F}$
HS1	saturation air enthalpy at inlet, Btu/lbm
HS2	saturation air enthalpy at outlet, Btu/lbm
HSD	enthalpy based on fin root temperature, Btu/lbm
HTIN	heat from heater plus pump heat input plus ambient losses, Btu/hr
HTOUT	heat gain by air based on data, Btu/hr
HTOUTA	heat gain by air based on data, Btu/hr
HTREJ	$\text{HTIN}-\text{QD}$
P	barometric pressure, in. Hg., atm.
P1	barometric pressure, psf
PHEE	heat exchanger effectiveness
PHI1	relative humidity, air inlet
PHI2	relative humidity, air outlet
PW1	partial pressure at $T_{\infty 1}$
PW1A	partial pressure based on inlet relative humidity

PW1	saturation pressure at T_{p1}
PW2	partial pressure at $T_{\infty 2}$
PW2A	partial pressure based on exit relative humidity
PW2	saturation pressure at T_{p2}
PWS1	saturation pressure at $T_{\infty 1}$
PWS1	saturation pressure at T_{d1}
PWS2	saturation pressure at $T_{\infty 2}$
PWSD	saturation pressure at fin root temperature
Q	pump heat input, plus ambient losses, Btu/hr
QD	deluge water heat input, Btu/hr
RE	Reynolds number
RENO	Reynolds number at same core pressure drop
T	dummy variable temperature
T1	$T_{\infty 1}$ - air inlet temperature, °F
T1	T_{p1} - core water inlet temperature, °F
T_2	$T_{\infty 2}$ - air outlet temperature, °F
T_2	T_{p2} - core water outlet temperature, °F
T_3	air inlet dew point temperature, °F
T_3	deluge outlet temperature, °F
T_4	air outlet dew point temperature, °F
T_5	deluge inlet temperature, °F
T_{10A}	deluge weigh tank temperature, °F
T_{10B}	storage auxiliary tank temperature, °F
T_{11}	core storage tank temperature, °F
TA	T_{p1} , core water inlet temperature, °C
TAI	$T_{\infty 1}$, air inlet temperature, °F
TAIR	ambient room temperature, °F
TB	deluge inlet temperature, °C
TC	core water outlet temperature, °C
TDB1	measured inlet air temperature, °F
TDB2	measured outlet air temperature, °F
THK	tube wall thickness, ft
TMEA1	measured relative humidity, inlet
TMEA2	measured relative humidity, outlet

TR	fin root temperature, °F
TT	tube wall thermal conductivity, Btu/hr-ft-°F
TVA1	absolute outlet air temperature, °R
TVA2	absolute outlet air temperature, °R
U	overall dry heat transfer coefficient, Btu/hr-ft ² -°F
VA1	specific volume of air at inlet, ft ³ /lb
VA2	specific volume of air at outlet, ft ³ /lb
V01	overall wet heat transfer coefficient, Btu/hr-ft ² -°F
V02	overall wet heat transfer coefficient, Btu/hr-ft ² -°F
V03	overall wet heat transfer coefficient, Btu/hr-ft ² -°F
V01A	overall wet heat transfer coefficient based on relative humidity data, Btu/hr-ft ² -°F
V02A	overall wet heat transfer coefficient based on relative humidity data, Btu/hr-ft ² -°F
V03A	overall wet heat transfer coefficient based on relative humidity data, Btu/hr-ft ² -°F
W1	humidity ratio of inlet air, lb/lb
W1A	humidity ratio of inlet air, based on relative humidity data, lb/lb
W12	humidity ratio of air at T _{p1} , lb/lb
W2	humidity ratio of outlet air, lb/lb
W2A	humidity ratio of outlet air based on relative humidity data, lb/lb
W22	humidity ratio at air at T _{p2} , lb/lb
WD	humidity ratio based on fin root temperature
WDLG	flow rate of deluge water - gpm
WDLP	flow rate of deluge water, lb/hr
WDLPA	flow rate of deluge water not evaporated, lb/hr
WDLPE	flow rate of deluge water evaporated, lb/hr
WHLG	flow rate of heater loop water, gpm
WHLP	flow rate of heater loop water, lb/hr
WMLG	flow rate of main loop water, gpm
WMLP	flow rate of main loop water - lb/hr
WS1	humidity ratio of saturated inlet air, lb/lb

WS2	humidity ratio of saturated outlet air, lb/lb
WW1	weight of water at T_{p1} , lb/gal
WW2	weight of water at T_{d1} , lb/gal
WW3	weight of water at T_{p2} , lb/gal
X	(Biot number) ^{1/2} , experimental
X1	thermal resistance of core water, $\text{hr-ft}^2\text{-}^\circ\text{F/Btu}$
X2	thermal resistance of tube wall, $\text{hr-ft}^2\text{-}^\circ\text{F/Btu}$
XNTU	experimental NTU for dry heat transfer
XXX	Q_w/Q_d
Y	fin effectiveness ratio
ZT	ξ , transformation parameter

```

DIMENSION DATE(3),TIME(3),CORE(3),IRON(3)
DIMENSION ARAYA(4)
DIMENSION ARRAY(4,2)
DIMENSION ARKAY1(4,6)
DIMENSION ARKAY2(4,6)
DIMENSION ARKAY3(4)
DIMENSION ARKAY4(4)
DIMENSION ARKAY6(4)
DIMENSION ARKAY8(4)
DIMENSION ARKAY9(4)
DIMENSION ARAY10(4,2)
DIMENSION ARAY11(4)

C
C
C      READ IN NUMBER OF TEST CASES

      ICON12 = 0
      READ(5,50) ICOUNT
50     FORMAT(I10)
25     CONTINUE
      ICON12 = ICON12 + 1

C
C
C      READ      -      DATE,TIME,CORE,RUN

      READ(5,100) DATE,TIME,CORE,IRON
100    FORMAT(4(2A4,A2))

C
C
C      READ      -      AIR FLOW RATE AND BAROMETRIC PRESSURE

      READ(5,200) ANDLP,P
200    FORMAT(2E10.0)

C
C
C
C      READ DELTA T'S      (HEATER AND CORE)      AND DELTA P

      READ(5,20) DTMVH,DTMVC,DELTA
20     FORMAT(3E10.0)

C
C
C      READ WATER FLOW RATES ,MAIN, HAEATER, AND DELUGE

      READ(5,30) WMLG,WHLG,WDLG
30     FORMAT(3E10.0)

C
C
C      READ MEASURED HUMIDITIES AND DRY BULB TEMPERATURES

      READ(5,40) TMEAL,TMEA2,TDB1,TDB2
40     FORMAT(4E10.0)
C

```



```

C      READ AIR TEMPERATURES - INSIDE AMBIENT
C
C      READ(5,300) (ARAYA(I),I=1,4)
C
C      READ AIR TEMPERATURES - AMBIENT AND ANNUBAR
C
C      READ(5,300)((ARRAY(I,J),I=1,4),J=1,2)
300    FORMAT(4E10.0)
C
C      READ AIR TEMPERATURES - CORE INLET
C
C      READ(5,400)((ARRAY1(I,J),I=1,4),J=1,6)
400    FORMAT(4E10.0)
C
C      READ AIR TEMPERATURES - CORE OUTLET
C
C      READ(5,500)((ARRAY2(I,J),I=1,4),J=1,6)
500    FORMAT(4E10.0)
C
C      READ DEW POINT TEMPERATURES - CORE INLET AND CORE OUTLET
C
C      READ(5,600)(ARRAY3(I),I=1,4)
600    FORMAT(4E10.0)
C      READ(5,600) (ARRAY4(I),I=1,4)
C
C      CALCULATE AVERAGE OF AIR TEMPERATURE - INSIDE AMBIENT
C      TAIR=0.
C      DO 700 I = 1,4
C      ARAYA(I) = ARAYA(I)*1.8+32.
C      TAIR = ARAYA(I) + TAIR
700    CONTINUE
C      TAIR = TAIR/4.
C      DO 800 J=1,4
C      DO 800 I=1,4
C      ARRAY(I,J)=ARRAY(I,J)*1.8+32.
800    CONTINUE
C
C      CALCULATE AVERAGE OF AIR TEMPERATURE - CORE INLET
C
C      T1 = 0.
C      DO 2000 J=2,4
C      DO 2000 I=1,4
C      ARRAY1(I,J) = ARRAY1(I,J)*1.8 + 32.
C      T1=ARRAY1(I,J)+T1
2000   CONTINUE
C      T1=T1/12.
C      TAI=T1
C

```

```

C      CALCULATE AVERAGE OF AIR TEMPERATURE - CORE OUTLET
C
      T2 = 0.
      DO 2100 J=2,4
      DO 2100 I=1,4
      ARRAY2(I,J) = ARRAY2(I,J)*1.8+32.
      T2=ARRAY2(I,J)+T2
2100   CONTINUE
      T2=T2/12.
      DO 2101 I=1,4
      ARRAY1(I,1)=ARRAY1(I,1)*1.8+32.
      ARRAY1(I,5)=ARRAY1(I,5)*1.8+32.
      ARRAY1(I,6)=ARRAY1(I,6)*1.8+32.
      ARRAY2(I,1) = ARRAY2(I,1)*1.8+32.
      ARRAY2(I,5)=ARRAY2(I,5)*1.8+32.
      ARRAY2(I,6)=ARRAY2(I,6)*1.8+32.
2101   CONTINUE
      TDB1=TDB1*1.8+32.
      TDB2=TDB2*1.8+32.

C
C      CALCULATE AVERAGE OF DEW POINT TEMPERATURE - CORE INLET
C
      T3 = 0.
      DO 2200 I=1,4
      T3 = ARRAY3(I)+T3
2200   CONTINUE
      T3 = T3/4.

C
C      CALCULATE AVERAGE OF DEW POINT TEMPERATURE - CORE OUTLET
C
      T4 = 0.
      DO 2300 I=1,4
      PB = P
      T4 = ARRAY4(I)+T4
2300   CONTINUE
      T4 = T4/4.

C
C      CALCULATE PARTIAL PRESSURE
C
      A = 3.2437814
      B = 5.86826*10.E-4
      C=1.1702379*10.E-9
      D=2.1878462*10.E-4
      T = (T3-32)/1.8+273.17
      BETA = 647.27-T
      PW = -(BETA/1*((A+B*BETA+C*BETA**3)/(1.0+D*BETA)))
      PW1=10**PW*210.167

```

```

C
C SECOND PARTIAL PRESSURE
C
  T = (T4-32)/1.8+273.17
  BETA = 647.27-T
  PW = -(BETA/T*((A+B*BETA+C*BETA**3)/(1.0+D*BETA)))
  PW2 = 10**PW*218.167

C
C THIRD PARTIAL PRESSURE
C
  T = (T1-32.)/1.8+273.17
  BETA=647.27-T
  PW = -(BETA/T*((A+B*BETA+C*BETA**3)/(1.0+D*BETA)))
  PWS1 = 10**PW*218.167

C
C FOURTH PARTIAL PRESSURE
C
  T = (T2-32.)/1.8+273.17
  BETA = 647.27-T
  PW = -(BETA/T*((A+B*BETA+C*BETA**3)/(1.0+D*BETA)))
  PWS2 = 10**PW*218.167

C
C CALCULATE HUMIDITY RATIOS
C
  P = P * .033421
  W1 = 0.62198 * PW1/(P-PW1)
  W2 = 0.62198 * PW2/(P-PW2)
  WS1=0.62198*PWS1/(P-PWS1)
  WS2=0.62198*PWS2/(P-PWS2)

C
C CALCULATE RELATIVE HUMIDITY
C
  PHI1 = (PW1/PWS1) * 100.
  PHI2 = (PW2/PWS2) * 100.

C
C CONVERT TO ABSOLUTE TEMPERATURE INLET AND OUTLET DRY BULBS
C
  TVA1 = T1+459.7
  TVA2 = T2+459.7

C
  P1 = P*2116.22
C CALCULATE SPECIFIC VOLUMES
C
  VA1 = 53.352 * TVA1*(1.0+1.6078*W1)/P1
  VA2 = 53.352*TVA2*(1.0+1.6078*W2)/P1
C

```

```

C      CALCULATE MOIST HEAT CAPACITY
C
      W10 = (W1+W2)/2.
      CA = 0.240+0.444*W10
      CA1=0.240 + 0.444*W1
C
C      CALCULATE ENTHALPYS
C
      H1 = 0.240*T1+W1*(1061.+0.444*T1)
      H2 = 0.240*T2+W2*(1061.+0.444*T2)
      HS1 = 0.240*T1+WS1*(1061.+0.444*T1)
      HS2 = 0.240*T2+WS2*(1061.+0.444*T2)
C
C      CALCULATE AIR FLOW RATE - LB/HR (AMFR)
C
      AMFR = 90326*SQRT(ANDLP)*SQRT(1.0/VA2)
C
C      CALCULATE AIR FLOW RATE - FPS (AV)
C
      AC = 12.0
      AV=AMFR*VA1/(3600.*AC)
      AVM=AV*60.
C
C
C      COMPUTE DELTA T'S TO (F)
C
      DTFH = DTMVH/.023
      DTFC = DTMVC/.023
C
C      CALCULATE FIRST HEAT OUTPUT. AIR
C
      HTOUT = AMFR * (H2-H1)
C
C      CALCULATE AIR ENTHALPIES BASED ON REL. HUM.
C
      PW1A=TMEA1/100*PWS1
      PW2A=TMEA2/100*PWS2
      W1A=0.62198*PW1A/(P-PW1A)
      W2A=0.62198*PW2A/(P-PW2A)
      H1A=0.240*TDB1+W1A*(1061.+0.444*TDB1)
      H2A=0.240*TDB2+W2A*(1061.+0.444*TDB2)
      CA1A=0.240+0.444*W1A
      W10A = (W1A + W2A)/2
      CAA = 0.240 + 0.444*W10A
      HTOUTA=AMFR*(H2A-H1A)
C

```

```

C
C   WRITE IT ALL OUT
C
C   WRITE(6,5000)
5000  FORMAT(1H1,52X,'WATA HEAT TRANSFER TESTS'///)
C
C   WRITE(6,5010) DATE,TIME,CORE,IRUN
5010  FORMAT(26X,'DATE-',2A4,A2,8X,'TIME-',2A4,A2,8X,'CORE-',
1      2A4,A2,8X,'RUN-',2A4,A2//)
C
C   WRITE(6,5020) ANDLP,AMFR,AV,AVM
5020  FORMAT(23X,'AIR FLOW RATE',2X,F4.2, 2X,'IN. H2O',7X,F7.0,
1      2X,'LB/HR',7X,F4.2,2X,'FPS',5X,F5.1,2X,'FPM'//)
C
C   WRITE(6,5050)
5050  FORMAT(1X,'AIR TEMPERATURES-'//)
C   WRITE(6,5055) APAYA
5055  FORMAT(7X,'INSIDE AMBIENT ( )',2X,4(F5.1,2X),'DEGREES F',26X,
1      '--- CALCULATED VALUES ---',/)
C
C   WRITE(6,5060) (ARRAY(I,1),I=1,4),H1
5060  FORMAT(6X,'OUTSIDE AMBIENT ( )',2X,4(F5.1,2X),'DEGREES F',/
1      89X,'ENTHALPY(1)',9X,F5.2 ,2X,'BTU/LB'//)
C
C   WRITE(6,5070) (ARRAY(I,2),I=1,4),W1
5070  FORMAT(14X,'ANNUBAR ( )',2X,4(F5.1,2X),'DEGREES F',/
1      89X,'HUMIDITY RATIO',6X, F5.4 ,2X,'LB/LB')
C
C   WRITE(6,5080) (ARRAY1(I,1),I=1,4),TMEAL,PHI1
5080  FORMAT(11X,'POSITION 1 ( ) ',4(F5.1,2X),'DEGREES F',10X,
1      'MEASURED',/,78X,F5.2,6X,'RELATIVE HUMIDITY(1)',F5.2 ,2X,'PER
2      2CENT'//)
C
C   WRITE(6,5090) (ARRAY1(I,2),I=1,4),H2
5090  FORMAT(11X,'POSITION 2 ( ) ',4(F5.1,2X),'DEGREES F',/89X,
1      'ENTHALPY(2)',9X,F5.2 ,2X,'BTU/LB'//)
C   WRITE(6,5100) (ARRAY1(I,3),I=1,4),T1,W2
5100  FORMAT(4X,'CORE POSITION 3 ( ) ',4(F5.1,2X),'DEGREES F',
1      4X,'T = ',F5.1,/, 4X,'INLET',80X,'HUMIDITY RATIO(2)',3X,
2      F5.4 ,2X,'LB/LB'//)
C
C   WRITE(6,5110) (ARRAY1(I,4),I=1,4),TMEAL2,PHI2
5110  FORMAT(11X,'POSITION 4 ( ) ',4(F5.1,2X),'DEGREES F',10X,
1      'MEASURED',/78X,F5.2,6X,'RELATIVE HUMIDITY(2)',F5.2 ,2X,
2      'PERCENT'//)

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```

C
WRITE(6,5120)(ARRAY1(I,5),I=1,4),VA1
5120 FORMAT(11X,'POSITION 5 ( ) ',4(F5.1,2X),'DEGREES F',/
1 89X,'VOLUME(1)',11X,F5.2,2X,'FT3/LB'//)
WRITE(6,5130)(ARRAY1(I,6),I=1,4),TDB1
5130 FORMAT(11X,'POSITION 6 ( ) ',4(F5.1,2X),'DEGREES F',5X,
1'TDB1= ',F5.1//)
C
C WRITE OUT AIR TEMPERATURES - CORE OUTLET
C
WRITE(6,5140)(ARRAY2(I,1),I=1,4),HS1
5140 FORMAT(11X,'POSITION 1 ( ) ',4(F5.1,2X),'DEGREES F',/
C
1 89X,'ENTHALPY(1)',9X,F5.2,2X,'BTU/LB'//)
C
WRITE(6,5150)(ARRAY2(I,2),I=1,4),WS1
5150 FORMAT(11X,'POSITION 2 ( ) ',4(F5.1,2X),'DEGREES F',/
1 89X,'HUMIDITY RATIO(1)',3X,F5.4,2X,'LB/LB'//)
C
WRITE(6,5160)(ARRAY2(I,3),I=1,4),T2,HS2
5160 FORMAT(4X,'CORE',3X,'POSITION 3 ( ) ',4(F5.1,2X),
1 'DEGREES F',4X,'T = ',F5.1,/4X,'OUTLET',79X,'ENTHALPY(2)',
2 9X,F5.2,2X,'BTU/LB'//)
C
WRITE(6,5170)(ARRAY2(I,4),I=1,4),WS2
5170 FORMAT(11X,'POSITION 4 ( ) ',4(F5.1,2X),'DEGREES F',/
1 89X,'HUMIDITY RATIO(2)',3X,F5.4,2X,'LB/LB'//)
C
C
C
C
WRITE(6,5180)(ARRAY2(I,5),I=1,4),VA2
5180 FORMAT(11X,'POSITION 5 ( ) ',4(F5.1,2X),'DEGREES F',/
1 89X,'VOLUME(2)',11X,F5.2,2X,'FT3/LB'//)
C
WRITE(6,5190)(ARRAY2(I,6),I=1,4),TDB2
5190 FORMAT(11X,'POSITION 6 ( ) ',4(F5.1,2X),'DEGREES F',5X,
1'TDB2= ',F5.1//)
C
WRITE(6,5200) CA,HTOUT,HTOUTA
5200 FORMAT(74X,'HEAT CAPACITY',1X,F7.4,1X,'BTU/LBM F',/
1 74X,'HEAT OUTPUT TO AIR',F9.0,1X,'BTU/HR (DEW POINT)',/
2 74X,'HEAT OUTPUT TO AIR',F9.0,1X,'BTU/HR (REL. HUM.)',/)
C
WRITE(6,5210)(ARRAY3(I),I=1,4),T3
5210 FORMAT(4X,'DEW POINT TEMPERATURES-',/,11X,'CORE INLET ( )',
1 2X,4(F5.1,2X),'DEGREES F',4X,'AVE INLET',3X,F5.1,/)
WRITE(6,5230)(ARRAY4(I),I=1,4),T4
5230 FORMAT(
2 10X,'CORE OUTLET ( ) ',4(F5.1,2X),'DEGREES F',4X,
3 'AVE OUTLET',2X,F5.1//)

```

```

C
C
C
C
C      READ WATER TEMPERATURES (CORE INLET)
C
      READ(5,5500)(ARRAY6(I),I=1,4)
5500  FORMAT(4E10.0)
C
C      READ DELUGE INLET TEMPERATURES
C
      READ(5,5520)(ARRAY8(I),I=1,4)
5520  FORMAT(4E10.0)
C
C      READ DELUGE OUTLET TEMPERATURES
C
      READ(5,5530)(ARRAY9(I),I=1,4)
5530  FORMAT(4E10.0)
C
C      READ STORAGE TEMPERATURES
C
      READ(5,5540)((ARRAY10(I,J),I=1,4),J=1,2)
5540  FORMAT(4E10.0)
      READ(5,5545)(ARRAY11(I),I=1,4)
5545  FORMAT(4E10.0)
C
C      CALCULATE AVERAGE OF CORE WATER INLET POSITIONS
C
      T1 = 0.
      DO 5550 I=1,4
      ARRAY6(I)=ARRAY6(I)*1.8+32.
      T1=ARRAY6(I)+T1
5550  CONTINUE
      T1=T1/4.
C
C      CALCULATE AVERAGE CORE OUTLET TEMPERATURE
C
      T2 = T1 - DTFC
C
C      CALCULATE AVERAGE OF DELUGE INLET TEMPERATURES
      T5 = 0.
      DO 5560 I = 1,4
      ARRAY8(I) = ARRAY8(I)*1.8+32.
      T5 = ARRAY8(I)+T5
5560  CONTINUE
      T5 = T5/4.
C

```

```

C      CALCULATE AVERAGE OF DELUGE OUTLET TEMPERATURES
C

```

```

      T3=0.
      DO 5570 I=1,4
      ARRAY9(I) = ARRAY9(I)*1.8+32.
      T3=ARRAY9(I)+T3
5570   CONTINUE
      T3 = T3/4.

```

```

C
C      CALCULATE AVERAGE STORAGE TEMPERATURES
C

```

```

      DO 5575 J=1,2
      DO 5575 I=1,4
      ARAY10(I,J)=ARAY10(I,J)*1.8+32.
5575   CONTINUE
      DO 5577 I=1,4
      ARAY11(I)=ARAY11(I)*1.8+32.
5577   CONTINUE
      T10A=0.
      T10B=0.
      T11 =0.
      DO 5580 I=1,4
      T10A=ARAY10(I,1)+T10A
      T10B=ARAY10(I,2)+T10B
      T11 =ARAY11(I)+T11
5580   CONTINUE
      T10A=T10A/4
      T10B=T10B/4
      T11=T11/4

```

```

C
C      COMPUTE WATER FLOW RATES, GPM TO LB/HR
C

```

```

      TA = (T1-32.)/1.8
      TB = (T5-32.)/1.8
      TC = (T2-32.)/1.8
      A = -3.1538 * 10.E-6
      B = -5.7575 * 10.E-5
      C = 8.3462
      WW1 = A*TA**2+B*TA+C
      WW2 = A*TB**2+B*TB+C
      WW3 = A*TC**2+B*TC+C
      WMLP = 60.*WW1*WMLG
      WHLP = 60.*WW3*WHLG
      WDLP = 60.*WW2*WDLG

```



```

C
C      COMPUTE HEAT INPUT AND OUTPUT
C
      WDLPE=AMFR*(w2-w1)
      WDLPA=WDLP-WDLPE
      QD=WDLPA*(T3-T5)
      Q = 25600. - 187.5 * (T1-TAIR)
      HTIN = DTFH*WHLF+Q
      HTREQ=HTIN-QD
C
      WRITE(6,5030) WMLG,WHLG,WDLG,WMLP,WHLF,WDLF
5030  FORMAT(22X,'WATER FLOW RATE-',2X,'MAIN',2X,F5.1,3X,'GPM',
1      6X,'HEATER',2X,F4.1,4X,'GPM',8X,'DELUGE',2X,F3.1,2X,'GPM'//
2      40X,'LOOP',2X,F7.0,1X,'LB/HR',4X,'LOOP',4X,F6.0,2X,
3      'LB/HR',6X,'LOOP',1X,F7.0,1X,'LB/HR'///)
C
      WRITE(6,5040) DTMVH,DTFH,HTIN,DTMVC,DTFC,QD
5040  FORMAT(22X,'HEATER IN/OUT DELTA T ', F5.3,2X,'MV',
1      2X,F5.2,1X,'F',11X,'HEAT INPUT=',2X,F7.0,2X,'BTU/HR'//
$ 22X,'CORE IN/OUT DELTA T ',3X,
2      F5.3,2X,'MV',2X,F5.2,1X,'F'
3      11X,'CORE HEAT =',2X,F7.0,2X,'BTU/HR'///)
C
C      CALCULATE PARTIAL PRESSURE
C
      A = 3.2437814
      B = 5.86826*10.E-4
      C=1.1702379*10.E-9
      D=2.1878462*10.E-4
      T = (T1-32.)/1.8+273.17
      BETA=647.27-T
      PW = -(BETA/T*((A+B*BETA+C*BETA**3)/(1.0+D*BETA)))
      PW1=10**PW*218.167
C
C      SECOND PARTIAL PRESSURE
C
      T = (T2-32.)/1.8+273.17
      BETA = 647.27-T
      PW = -(BETA/T*((A+B*BETA+C*BETA**3)/(1.0+D*BETA)))
      PW2=10**PW*218.167
C
C      THIRD PARTIAL PRESSURE
C
      T = (T3-32.)/1.8+273.17
      BETA = 647.27-T
      PW = -(BETA/T*((A+B*BETA+C*BETA**3)/(1.0+D*BETA)))
      PWS1= 10**PW*218.167
C

```

```

C      CALCULATE HUMIDITY RATIOS -- PRIMARY SIDE
C
      W12 = 0.62198 * PW1/(P-PW1)
      W22 = 0.62198 * PW2/(P-PW2)
C
C      CALCULATE ENTHALPIES -- PRIMARY SIDE
C
      H12 = 0.240*T1+W12*(1061.+0.444*T1)
      H22 = 0.240*T2+W22*(1061.+0.444*T2)
C
C      CALCULATE HEAT OUTPUT -- PRIMARY SIDE
C
      DTFC=DTMVC/.024
      HTOUT=DTFC*WMLP
C
C      CALCULATE NONDIMENSIONAL INLET DRIVING POTENTIAL
C
      GAMA=(H12-H1)/(CA1*(T1-TA1))
      GAMAA=(H12-H1A)/(CA1A*(T1-TDB1))
C
C      WRITE IT ALL OUT
C
      WRITE(6,6000)
6000  FORMAT(1X,'WATER TEMPERATURES-'//)
C
      WRITE(6,6010)(ARRAY6(I),I=1,4),T1,H12
6010  FORMAT(4X,'CORE POSITION ( ) ',4(F5.1,2X),
1      'DEGREES F',5X,'T= ',F5.1,12X,'ENTHALPY(1)',9X,F5.1,2X,
2      'BTU/LB'//,4X,'INLET'//)
C
      WRITE(6,6020)W12
6020  FORMAT(89X,'HUMIDITY RATIO(1)',3X,F5.4,2X,'LB/LB'//)
C
      WRITE(6,6030) T2,H22,W22
6030  FORMAT(4X,'CORE POSITION ( ) ',28X,'DEGREES F',
1      5X,'T= ',F5.1,12X,'ENTHALPY',12X,F5.1,2X,'BTU/LB'//,4X
2      'OUTLET',/89X,'HUMIDITY RATIO(1)',3X,F5.4,2X,'LB/LB'//)
C
      WRITE(6,6040)(ARRAY8(I),I=1,4),T5,HTOUT
6040  FORMAT(3X,'DELUGE POSITION ( ) ',4(F5.1,2X),'DEGREES F',5X,
A      'T= ',F5.1,12X,
1      'CORE HEAT OUTPUT ',1X,F7.0,2X,'BTU/HR'//3X,'INLET'//)
      WRITE(6,6050)(ARRAY9(I),I=1,4),T3,H1,H1A
6050  FORMAT(3X,'DELUGE POSITION ( ) ',4(F5.1,2X),
1      'DEGREES F',5X,'T= ',F5.1,12X,'INLET AIR',12X,F5.2,1X,
2      '(DEW POINT)'//,3X,'OUTLET',80X,'ENTHALPY (BTU/LBM)',3X,F5.2,
3      1X,'(REL. HUM.)'//)
      WRITE(6,6060)(ARAY10(I,1),I=1,4),T10A,CA1,(ARAY10(I,2),I=1,4),
1      T10B,CA1A

```

```

6060  FORMAT(1X,'DELUGE WEIGH TANK  ( ) ',4(F5.1,2X),'DEGREES F',
      1  5X,'T= ',F5.1,12X,'HEAT CAPACITY -',6X,F5.4,1X,'(DEW POINT)'/
      11X,'STORAGE AUX. TANK  ( ) ',4(F5.1,2X),'DEGREES F',
      1  5X,'T= ',F5.1,12X,'INLET (BTU/LBM F)',4X,F5.4,1X,'(REL. HUM.)'/)
      WRITE(6,6065) GAMA,GAMAA
6065  FORMAT(89X,'NONDIMEN. INLET',6X,F5.2,1X,'(DEW POINT)'/,
      1  89X,'DRIVING POTENTIAL',4X,F5.2,1X,'(REL. HUM.)')
      WRITE(6,6080) (ARAY11(I),I=1,4),T11
6080  FORMAT( 1X,'CORE STORAGE TANK  ( ) ',4(F5.1,2X),'DEGREES F',
      1  5X,'T= ',F5.1/)
      WRITE(6,6070) PH,DELTA
6070  FORMAT(1X,'AIR PRESSURES-',20X,'BAROMETRIC ',F5.2,
      1  'IN. HG.',10X,'CORE DELTA P -',F4.2,'IN. H2O'/)

C
C
C  ADDITIONAL CALCULATIONS
C  OVERALL HEAT TRANSFER COEFFICIENTS
C
C  CALCULATION BASED ON DEW POINT DATA
C
AS=705.783
DH1 = H12-H1
DH2 = H22-H2
DELH = (DH1-DH2)/ALOG(DH1/DH2)
VO1=CA*HTREJ/(AS*DELH)
DUMMY=ABS(1.-(HTREJ/(AMFR*DH1)))
VO2 = -AMFR*CA/AS *ALOG (DUMMY)
DUMMY = ABS(1-(H2-H1)/DH1)
VO3 = -AMFR*CA/AS*ALOG (DUMMY)
RE=0.0517*AMFR
HP=11.680648*WMLG**0.8

C
C  CALCULATION BASED ON RELATIVE HUMIDITY DATA
C
DH1A = H12 - H1A
DH2A = H22 - H2A
DELHA = (DH1A-DH2A)/ALOG(DH1A/DH2A)
VO1A = CAA*HTREJ/(AS*DELHA)
DUMMY=ABS(1.-(HTREJ/(AMFR*DH1A)))
VO2A= -AMFR*CAA/AS*ALOG(DUMMY)
DUMMY=ABS(1.-(H2A-H1A)/DH1A)
VO3A=-AMFR*CAA/AS*ALOG(DUMMY)

C
C
      WRITE(6,9999) VO1,VO2,VO3
9999  FORMAT(5X,'UO1 = ',F5.2,5X,'UO2 = ',F7.2,5X,
      1  'UO3 = ',F5.2,5X,'(DEW POINT)')
      WRITE(6,9797) VO1A,VO2A,VO3A

```

```

9797  FORMAT(5X,'U01 = ',F5.2,5X,'U02 = ',F7.2,5X,
1      'U03 = ',F5.2,5X,'(REL. HUM.)')
C
      WRITE(6,9898) RE,HP
9898  FORMAT(5X,'RE = ',F7.2,5X,'HP = ',F7.2/)
C
C
C      WET HEAT TRANSFER/ DRY HEAT TRANSFER
C
C
      AP=55.74
      APT=40.9219
      TT=128.
      THK = 0.0041
      X1 = AS/(HP*AP)
      X2 = AS*THK/(TT*APT)
      HA=18.0911*DELTA**0.3279
      X=0.1513*SQRT(HA)
      ETAF=-0.419*X+1.125
      Y=705.7829/(61.8738+ETAF*643.9091)
      U=1.0/(Y/HA+X1+X2)
      RENO=2485.4291*DELTA**0.59566
      XNTU=154.7*U/RENO
      PHEF=1.-EXP(-XNTU)
      Q=PHEE*RENO*+.56226*(T1-TAI)
      XXX = HTIN/Q
      WRITE(6,61234) XXX
61234  FORMAT ( 5X,'QWET/QDRY = ', F6.3)
C
C
C      THE FOLLOWING CALCULATES HDX AND TR TO MATCH HEAT TRANSFER
C      PATHS FROM INSIDE TO OUTSIDE SURFACES
C
C
      HDX1=10000
      HS=HA
      ASX=0.1
      TR=T3
89     CONTINUE
      T=(TR-32.)/1.8+273.17
      BETA=647.27-T
      PW = -(BETA/T*((A+B*BETA+C*BETA**3)/(1.0+D*BETA)))
      PWSO=10**PW*218.167
      WD=0.62198*PWSO/(P-PWSO)
      HSD=0.240*TR+WD*(1061.+0.444*TR)
      ZT=(H12-HSD)/(CAA*(T1-TR))
      WRITE(6,7511) ZT,HSD,TR,HS,X1,X2

```

```

7511 FORMAT(3X,'ZT=',F6.3,3X,'HSD=',F10.2,3X,'TR=',F6.2,3X,'HS=',F10.2,
$3X,'X1=',F10.7,3X,'X2=',F10.7)
87 CONTINUE
HD=1./(1./(ZT*V02A)-1./(HS*ASX*ZT)-X1-X2)
HDX=HD/ASX
IF(ABS((HDX1-HDX)/HDX1).LE.0.0005) GO TO 88
HE=1./(1./HDX+1./(ZT*HS))
WRITE(6,7512) ASX,HDX,HDX1
7512 FORMAT(2X,'ASX=',F5.3,3X,'HDX=',F10.2,3X,'HDX1=',F10.2)
BX=0.1513*SORT(HE)
ETAFX=-0.419*BX+1.125
ASX=(61.8/38+ETAFX*643.9091)/705.7829
HDX1=HDX
GO TO 87
88 CONTINUE
TRX=TR
TX=HTPEJ*(1./HD+X1+X2)/AS
TR=T1-TX
IF(ABS((TR-TRX)/TRX).GE.0.0005) GO TO 89
WRITE(6,7500) ASX,HD,ZT,TR,TX
7500 FORMAT(5X,'ASX = ',F6.4,5X,'HD = ',F6.2,5X,'ZETA = ',F6.3,5X,
$ 'TR = ',F6.2,5X,'TP - TR = ',F6.2)
C
C
C
IF(ICONT.NE.ICONT2) GO TO 25
END
MAP,

```


APPENDIX C

SAMPLE CALCULATIONS

APPENDIX C

SAMPLE CALCULATIONS

PRESSURE DROP CALCULATIONS

The following calculations are an example of the reduction from pressure drop data to a friction factor, f_o , for the Unifin heat exchanger.

$$f_o = \frac{2g_c \rho_m \Delta P_c A_c^3}{\dot{m}_a^2 A_s}$$

where $A_c = 6.11 \text{ ft}^2$
 $A_s = 705.78 \text{ ft}^2$

To complete the calculation of f_o , data from dry test 1 are used.

$$\Delta P_c = 0.35 \text{ in. of water}$$

$$\Delta P_{ann} = 1.08 \text{ in. of water}$$

$$T_{\infty 1} = 100.7^\circ\text{F}$$

$$T_{\infty 2} = 113.0^\circ\text{F}$$

$$\dot{m}_a = 90326 \sqrt{\rho_{ann}} \sqrt{\Delta P_{ann}}$$

$$\rho_{ann} = 0.067544 \text{ lbm/ft}^3$$

$$\dot{m}_a = 24,400 \text{ lbm/hr}$$

$$\rho_m = 0.0702 \text{ lbm/ft}^3$$

Therefore

$$f_o = \frac{2(3600)^2 (32.174) (0.0702) \left(\frac{0.35}{12}\right) (62.37) (6.11)^3}{(24400)^2 (705.78)}$$

$$= \frac{\left(\frac{\text{sec}^2}{\text{hr}^2}\right)\left(\frac{1\text{bm ft}}{1\text{bf sec}^2}\right)\left(\frac{1\text{bm}}{\text{ft}^3}\right)\left(\frac{1\text{bf}}{\text{ft}^2}\right)(\text{ft}^2)^3}{\left(\frac{1\text{bm}^2}{\text{hr}^2}\right)\text{ft}^2}$$

$$f_o = 0.0578$$

DRY HEAT TRANSFER CALCULATIONS

Dry heat transfer data reduction is illustrated by the subsequent calculation using the data of dry test 1.

$$T_{\infty 1} = 100.7^\circ\text{F}$$

$$T_{\infty 2} = 113.0^\circ\text{F}$$

$$T_{p1} = 120.2^\circ\text{F}$$

$$\Delta T_p = \frac{13\mu\text{V}}{23 \frac{\mu\text{V}}{^\circ\text{F}}} = 0.6^\circ\text{F}$$

$$T_{p2} = T_{p1} - \Delta T_p = 119.6^\circ\text{F}$$

$$\text{LMTD} = \frac{(T_{p1} - T_{\infty 2}) - (T_{p2} - T_{\infty 1})}{\ln\left(\frac{T_{p1} - T_{\infty 2}}{T_{p2} - T_{\infty 1}}\right)} = 12.12^\circ\text{F}$$

The amount of heat transferred from the water to the air, Q_{rej} , is calculated as follows:

$$Q_{\text{rej}} = Q_{\text{heater}} + Q_{\text{peripheral}}$$

$$Q_{\text{heater}} = \dot{m}_h C_p \Delta T_h$$

$$C_p = 1.00 \text{ Btu/lbm-}^\circ\text{F}$$

$$\dot{m}_h = (\text{gpm}) \left(\frac{\text{min}}{\text{hr}} \right) \left(\frac{\text{ft}^3}{\text{gal}} \right) \left(\frac{\text{lb}}{\text{ft}^3} \right)$$

$$= (25)(60)(0.1337)(61.655)$$

$$\dot{m}_h = 12,365 \text{ lb/hr}$$

$$\Delta T_h = \frac{120 \mu\text{V}}{23 \frac{\mu\text{V}}{^\circ\text{F}}} = 5.22^\circ\text{F}$$

$$Q_{\text{heater}} = \left(\frac{\text{lb}}{\text{hr}} \right) \left(\frac{\text{Btu}}{\text{lb}^\circ\text{F}} \right) (^\circ\text{F})$$

$$= (12365)(1)(5.22)$$

$$Q_{\text{heater}} = 64,500 \text{ Btu/hr}$$

$$Q_{\text{peripheral}} = 19,800 \text{ [from empirical data]}$$

$$Q_{\text{rej}} = 84,300 \text{ Btu/hr}$$

The overall heat transfer coefficient, U_o , is finally calculated by

$$U_o = \frac{Q_{\text{rej}}}{A_s \times \text{LMTD}} = \frac{84300 \frac{\text{Btu}}{\text{hr}}}{(706 \text{ ft}^2)(12.12^\circ\text{F})} = 9.85 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

The air velocity across the face of the core, V_o , is calculated by

$$V_o = \frac{\dot{m}_a}{\rho_a A_f}$$

$$\dot{m}_a = 24400 \frac{\text{lb}}{\text{hr}}$$

$$\rho_a = 0.0710 \text{ lb/ft}^3 (\text{ @ } 100^\circ\text{F})$$

$$V_o = \frac{(1\text{b/hr})}{(1\text{b/ft}^3)(\text{ft}^2)(\frac{\text{sec}}{\text{hr}})} = \frac{(24400)}{(0.0710)(12)(3600)} = 8. \frac{\text{ft}}{\text{sec}}$$

The air mass velocity G_o is calculated by

$$G_o = \dot{m}_a / A_f = 24400 / 12 = 2033 \text{ lb/ft}^2 \cdot \text{hr}$$

The interior heat transfer coefficient is calculated using the Dittus-Boelter equation for smooth walled tubes being cooled.

$$h_p = 0.023 \frac{k_w}{D_h} \text{Re}_{Dh}^{0.8} \text{Pr}^{0.3}$$

Properties of water were evaluated at 120°F :

$$k_w = 0.369 \frac{\text{Btu}}{\text{hr ft}^\circ\text{F}}$$

$$\text{Pr}_w = 3.75$$

$$\nu_w = 0.622 (10^{-5}) \frac{\text{ft}}{\text{sec}^2}$$

The hydraulic diameter, D_h , of a circular pipe is given by the outer diameter less twice the thickness of the tube wall.

$$D_h = .D. - 2t_t = 0.625 \text{ in.} - 2(0.049 \text{ in.}) = 0.527 \text{ in.}$$

$$D_h = 0.0439 \text{ ft}$$

Velocity of the water through a tube is given by

$$V = \frac{\dot{V}}{A_x} = \frac{\text{ft}^3/\text{sec}}{\text{ft}^2}$$

where $A_x = (\text{number of tubes})(\text{cross sectional area/tube})$

$$\dot{V} = \frac{(\text{gpm})(\text{ft}^3/\text{gal})}{(\text{sec}/\text{min})} = \frac{(300)(0.1337)}{60} = 0.669 \frac{\text{ft}^3}{\text{sec}}$$

$$A_x = \frac{(\text{in}^2)}{\text{in}^2/\text{ft}^2} = (101)\left(\frac{1}{4}\pi [0.625 - 2(0.049)]^2\right) = 0.153 \text{ ft}^2$$

$$\text{Thus } V = \frac{0.669}{0.153} = 4.37 \text{ ft/sec}$$

The Reynolds number based on the hydraulic diameter is therefore

$$\text{Re}_{D_h} = \frac{V D_h}{\nu} = \frac{(\text{ft}/\text{sec})(\text{ft})}{\text{ft}^2/\text{sec}} = \frac{(4.37)(0.0439)}{0.622(10^{-5})}$$

$$\text{Re}_{D_h} = 30,840$$

Next, h_p is calculated:

$$h_p = 0.023 \frac{0.369}{0.0439} (30,840)^{0.8} (3.75)^{0.3}$$

$$\frac{\frac{\text{Btu}}{\text{hr-ft-}^\circ\text{F}}}{\text{ft}} = \frac{\text{Btu}}{\text{hr-ft}^2\text{-}^\circ\text{F}}$$

$$h_p = 1122 \frac{\text{Btu}}{\text{hr-ft}^2\text{-}^\circ\text{F}}$$

The outside heat transfer coefficient which includes fin efficiency is given by

$$h_o = \left[\frac{1}{U_o} - \frac{A_s}{h_p A_i} - \frac{A_s t_t}{A_{io} k_f} \right]^{-1}$$

where $k_f = 128 \text{ Btu/hr-ft-}^\circ\text{F}$

$$t_t = 0.0041 \text{ ft}$$

$$A_i = 55.74 \text{ ft}^2$$

$$A_o = 66.10 \text{ ft}^2$$

$$A_{io} = 1/2(A_i + A_o) = 60.92 \text{ ft}^2$$

$$A_s = 706 \text{ ft}^2$$

$$h_o = \left[\frac{1}{9.85} - \frac{706}{(1122)(55.74)} - \frac{706(0.0041)}{(60.92)(128)} \right]^{-1}$$

$$\left[\frac{\frac{1}{\frac{\text{Btu}}{\text{hr-ft}^2\text{-}^\circ\text{F}}}}{\frac{\text{ft}^2}{\left(\frac{\text{Btu}}{\text{hr-ft}^2\text{-}^\circ\text{F}}\right)\text{ft}^2}} - \frac{\text{ft ft}^2}{\text{ft}^2 \left(\frac{\text{Btu}}{\text{hr-ft-}^\circ\text{F}}\right)} \right]^{-1}$$

$$h_o = 11.13 \frac{\text{Btu}}{\text{hr-ft}^2\text{-}^\circ\text{F}}$$

If the fin efficiency, η_f , is extracted, the air side heat transfer coefficient, h_s , is obtained:

$$h_s = \left(\frac{A_s}{A_{st} + n_f A_{sf}} \right) h_o$$

$$A_{st} = 61.87 \text{ ft}^2$$

$$A_{sf} = 643.91 \text{ ft}^2$$

The fin efficiency is a function of both r_e/r_b and $Bi^{1/2}$, where

$$Bi^{1/2} = \ell_f \left(\frac{h_s}{k_f y_b} \right)^{1/2} \quad k_f = 128 \frac{\text{Btu}}{\text{hr-ft-}^\circ\text{F}} \quad y_b = 0.004 \text{ in.}$$

$$r_e/r_b = d_e/d_b = 1.375/0.625 = 2.2$$

$$\ell_f = \frac{1}{2} (d_e - d_b) = 0.375 \text{ in.}$$

$$\ell_f = 0.03125 \text{ ft}$$

The iterative solution for h_s and n_f is initiated by using h_o as the first guess of h_s :

$$Bi^{1/2} = 0.03125 \left(\frac{11.13}{128(0.000333)} \right)^{1/2} = 0.50$$

$$(\text{ft}) \left(\frac{\text{Btu/hr-ft}^2\text{-}^\circ\text{F}}{\frac{\text{Btu}}{\text{hr-ft-}^\circ\text{F ft}}} \right)^{1/2}$$

$$n_f = 0.88 \quad [\text{From Figure A.3}]$$

After additional iterations the final answers are

$$n_f = 0.87$$

$$h_s = 12.59 \frac{\text{Btu}}{\text{hr-ft}^2-\text{°F}}$$

Heat transfer is also characterized by a Colburn factor, j_o , which is given by

$$j_o = \frac{h_o}{G_o C_{p a}} \text{Pr}_a^{2/3}$$

$$G_o = \dot{m}_a / A_c$$

$$A_c = 6.11 \text{ ft}^2$$

Air properties evaluated at 105°F (approximate average of inlet and outlet air temperature across the core 1).

$$C_{p a / 105} = 0.24 \text{ Btu/lbm°F}$$

$$\text{Pr}_{a/105} = 0.705$$

$$j_o = \frac{11.13 (0.705)^{2/3}}{\left(\frac{24400}{6.11}\right) (0.240)} = 0.0092$$

$$\frac{\frac{\text{Btu}}{\text{hr-ft}^2-\text{°F}}}{\left(\frac{\frac{\text{lbm}}{\text{hr}}}{\text{ft}^2}\right)\left(\frac{\text{Btu}}{\text{lbm}\cdot\text{°F}}\right)}$$

WET HEAT TRANSFER

A sample of all wet heat transfer calculations follows. All calculations were performed on the computer. Data from Test 2-1 is used in the sample calculations.

Note that each input temperature has four values which are recorded at one-minute intervals. An average of these four numbers is used for calculations.

Air Temperatures

Inlet dry bulb temperatures are averaged over six locations. Outlet dry bulb temperatures are averaged over five locations. The sixth is interfered with by the deluge inlet water.

$$\begin{aligned} T_{\infty 1} &= 89.9^{\circ}\text{F} \\ T_{\infty 2} &= 105.8^{\circ}\text{F} \end{aligned} \quad \left[\begin{array}{l} 1 = \text{inlet} \\ 2 = \text{outlet} \end{array} \right]$$

The dew point temperatures are

$$\begin{aligned} T_{d_{p1}} &= 73.3^{\circ}\text{F} \\ T_{d_{p2}} &= 96.0^{\circ}\text{F} \end{aligned}$$

The calculation of air properties follows the outline given in the ASHRAE handbook (1977). The saturation pressures for the above four temperatures are calculated using the Keenan-Keeyes formula

$$\log_{10} \left(\frac{P_{ws}}{218.167} \right) = -\frac{\beta}{T} \left(\frac{a + b\beta + c\beta^3}{1 + d\beta} \right)$$

where P_{ws} = saturation pressure, atm

$$b = 647.27 - T$$

T = absolute temperature, K

$$a = 3.2437814$$

$$b = 5.86826 \times 10^{-3}$$

$$c = 1.1702379 \times 10^{-8}$$

$$d = 2.1878462 \times 10^{-3}$$

The saturation pressures must be calculated for all four temperatures given above.

$$P_{ws}(T_{\infty 1}) = 0.0475 \text{ atm}$$

$$P_{ws}(T_{\infty 2}) = 0.0769 \text{ atm}$$

$$P_{ws}(T_{d_{p1}}) = 0.0277 \text{ atm}$$

$$P_{ws}(T_{d_{p2}}) = 0.0573 \text{ atm}$$

The barometric pressure (P) is 29.32 in. Hg

$$= 0.980 \text{ atm.}$$

The humidity ratios may be calculated

$$H = 0.62198 \frac{P_{ws}}{P - P_{ws}}$$

$$H(T_{\infty 1}) = 0.0317$$

$$H(T_{\infty 2}) = 0.0530$$

$$H(T_{d_{p1}}) = 0.0180$$

$$H(T_{d_{p2}}) = 0.0387$$

The relative humidities are calculated.

$$\phi_1 = \frac{P_{ws}(T_{dp1})}{P_{ws}(T_{\infty1})} \times 100$$

$$\phi_2 = \frac{P_{ws}(T_{dp2})}{P_{ws}(T_{\infty2})} \times 100$$

$$\phi_1 = 58.3$$

$$\phi_2 = 74.5$$

The specific volumes are calculated:

$$v = \frac{R_a T}{P} (1 + 1.6078 H(T_{dp}))$$

$$\text{where } R_a = 53.352 \frac{\text{ft-lbf}}{\text{lbm-}^\circ\text{R}}$$

$$T_{\infty} = T_{\infty}(^\circ\text{F}) + 459.7$$

$$P = P(\text{in. Hg}) \times 70.73, \text{ lbf/ft}^2$$

$$v_1 = 14.45 \text{ ft}^3/\text{lbm}$$

$$v_2 = 15.45 \text{ ft}^3/\text{lbm}$$

Calculated heat capacity C_a :

$$C_a = 0.240 + \frac{0.444}{2} (H(T_{dp1}) + H(T_{dp2}))$$

$$= 0.240 + \frac{0.444}{2} (0.01556 + 0.03650)$$

$$= 0.253 \text{ Btu/lbm-}^\circ\text{F}$$

The enthalpies are calculated: $[T] \text{ } ^\circ\text{F}$

$$i(x_i)_1 = 0.240 T_{\infty i} + H(x)(1061 + 0.444 T_{\infty i})$$

$$\begin{aligned} i(T_{\infty 1}) &= 0.240 (89.9) + 0.0318 (1061 + 0.44 \times 89.9) \\ &= 56.5 \text{ Btu/lb dry air} \end{aligned}$$

Likewise, $i(T_{\infty 2}) = 84.1 \text{ Btu/lb dry air}$

$$i(T_{dp1}) = 41.4 \text{ Btu/lb dry air}$$

$$i(T_{dp2}) = 68.3 \text{ Btu/lb dry air}$$

The air flow rate is calculated from the pressure drop across the annubar (ΔP_{am}).

$$\begin{aligned} \dot{m}_a &= 90326 \sqrt{\Delta P_{ann}} \sqrt{1/v_2} \\ &= 90326 \sqrt{0.31} \sqrt{1/15.45} \\ &= 12,795 \text{ lb/hr} \end{aligned}$$

The air frontal velocity is calculated:

$$V_o = \frac{\dot{m}_a \times V_1}{A_f \times 3600}$$

$$A_f = \text{frontal area, } 12 \text{ ft}^2$$

$$V_o = \frac{12795 \times 14.45}{12.0 \times 3600}$$

$$V_o = 4.28 \text{ fps}$$

Heat change in air:

$$\begin{aligned}Q &= \dot{m}_a(i(T_{dp2}) - i(T_{dp1})) \\&= 12795 (65.8 - 37.3) \\Q &= 364,658 \text{ Btu/hr}\end{aligned}$$

The mass flow rates of water must be calculated at three different places: 1) deluge water inlet, 2) core water inlet, 3) heater inlet. The temperatures involved are

$$\begin{aligned}\text{deluge inlet} &= 102.6^\circ\text{F} \\ \text{deluge outlet} &= 108.1^\circ\text{F} \\ \text{core inlet} &= 119.7^\circ\text{F} \\ \text{core outlet} &= 116.4^\circ\text{F}\end{aligned}$$

The core outlet temperature is calculated by subtracting the core temperature drop from the inlet temperature. The temperature for the heater inlet is assumed to be the same as the core outlet temperature.

The densities of the water at the three different temperatures are

$$\begin{aligned}\text{WW1)} &= 8.3359 \text{ lb/gal} \\ \text{WW2)} &= 8.3391 \text{ lb/gal} \\ \text{WW3)} &= 8.3366 \text{ lb/gal}\end{aligned}$$

The mass flow rates in the three different streams become

$$\begin{aligned}\dot{m} &= \left(\frac{\text{gal}}{\text{min}}\right)\left(\frac{\text{lb}}{\text{gal}}\right)\left(\frac{\text{min}}{\text{hr}}\right) \\ \dot{m}_w^1 &= 300 \times 8.3359 \times 60 = 150046 \text{ lb/hr} \\ \dot{m}_w^2 &= 3.0 \times 8.3391 \times 60 = 1501 \text{ lb/hr} \\ \dot{m}_w^3 &= 25.0 \times 8.3366 \times 60 = 12505 \text{ lb/hr}\end{aligned}$$

The heat input to the deluge water is

$$Q_d = \dot{m}_w^1 \times [T_d(\text{out}) - T_d(\text{in})] (C_p)$$

$$= 1501 (1)(108.1-102.6) \left[\frac{\text{lb}}{\text{hr}} \frac{\text{Btu}}{\text{lb} \cdot ^\circ\text{F}} ^\circ\text{F} \right]$$

$$Q_d = 8256 \text{ Btu/hr}$$

The heat loss across the core is

$$Q_{\text{loss}} = \dot{m}_w^2 \times [T_c(\text{in}) - T_c(\text{out})] (C_p)$$

$$= (150046)(1)(119.7 - 116.4)$$

$$= 495152 \text{ Btu/hr}$$

The heat input to the heaters is

$$Q_h = \dot{m}_w^3 C_p \Delta T_h$$

$$= 12505(1) \frac{895}{23} \left[\frac{(\text{lb/hr})(\text{Btu/lbm} \cdot ^\circ\text{F})(\mu\text{V})}{(\mu\text{V}/^\circ\text{F})} \right]$$

$$= 486608 \text{ Btu/hr}$$

The actual heat transferred from the primary fluid to the air stream is

$$Q_{\text{heaters}} + Q_{\text{pump}} - Q_{\text{deluge}} - Q_{\text{surrounding}}$$

The combination of $Q_{\text{pump}} - Q_{\text{surrounding}}$ was measured experimentally and is given by the following equation:

$$Q_{\text{pump}} - Q_{\text{surrounding}} = 25600 - 187.5 (T_c - T_{\text{amb}})$$

For this case $T_{amb} = 81.7^{\circ}\text{F}$

$$T_c \text{ (in)} = 119.7^{\circ}\text{F}$$

$$Q_{\text{pump}} - Q_{\text{surrounding}} = 18475 \text{ Btu/hr}$$

Therefore, the total heat transferred is

$$Q_{\text{rej}} = 486608 + 18475 - 8256$$

$$Q_{\text{rej}} = 496827 \text{ Btu/hr}$$

The properties of saturated air are also calculated at the two primary side temperatures.

$$T_{p1} = 119.7^{\circ}\text{F}$$

$$T_{p2} = 116.4^{\circ}\text{F}$$

$$P_{ws}(T_{p1}) = 0.114513 \text{ atm}$$

$$P_{ws}(T_{p2}) = 0.104425 \text{ atm}$$

$$H(T_{p1}) = 0.082294$$

$$H(T_{p2}) = 0.074180$$

$$i(T_{p1}) = 120.42 \text{ Btu/lbm}$$

$$i(T_{p2}) = 110.47 \text{ Btu/lbm}$$

The overall wet transfer coefficient can be calculated from experimental values three different ways:

$$U_{o1}^* = \frac{C_a Q_{\text{rej}}}{A_s} \frac{\epsilon_n \frac{i(T_{p1}) - i(T_{dp1})}{i(T_{p2}) - i(T_{dp2})}}{i(T_{p1}) - i(T_{dp1}) - i(T_{p2}) + i(T_{dp2})}$$

$$U_{02}^* = - \frac{\dot{m}_a C_a}{A_s} \ell_n \left| 1 - \frac{Q_{rej}}{m_a (i(T_{p1}) - i(T_{dp1}))} \right|$$

$$U_{03}^* = - \frac{\dot{m}_a C_a}{A_s} \ell_n \left| 1 - \frac{i(T_{dp2}) - i(T_{dp1})}{i(T_{p1}) - i(T_{dp1})} \right|$$

$$U_{01}^* = \frac{0.253 \times 496827}{706} \ell_n \frac{120.42 - 41.4}{110.47 - 68.3}$$

$$= 3.03 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

$$U_{03}^* = - \frac{12,795 \times 0.253}{706} \ell_n \left| 1 - \frac{496827}{12,795(120.42 - 41.4)} \right|$$

$$= 3.10 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

$$U_{03}^* = - \frac{12,795 \times 0.253}{706} \ell_n \left| 1 - \frac{68.3 - 41.4}{120.42 - 41.4} \right|$$

$$= 1.91 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

The calculation of Reynolds number is as follows:

$$Re = \frac{\text{Air Mass Velocity} \times \text{Hydraulic Diameter}}{\text{Dynamic Viscosity}}$$

$$\text{Viscosity} = 0.0470 \text{ lb/ft-hr}$$

$$\text{Air Mass Velocity } G_o = \frac{\dot{m}_a}{A_{min}}$$

$$G_o = \frac{12795}{6.11} = 2094 \frac{\text{lb}}{\text{hr-ft}^2}$$

$$\text{Hydraulic Diameter} = 0.01455 \text{ ft}$$

$$Re = \frac{0.01455 \times 2094}{0.0470} \frac{[\text{ft}][\text{lb/hr}]}{[\text{lb/ft-hr}][\text{ft}^2]}$$

$$= 648.3$$

The primary side heat transfer coefficient is calculated using the Dittus-Boelter equation and is the same as calculated for dry heat transfer. The amount of deluge water evaporated is:

$$\begin{aligned}\dot{m}_w &= \dot{m}_a [H(T_{dp2}) - H(T_{dp1})] \\ &= 13000 [0.0387 - 0.018] \\ &= 269.1 \text{ lb/hr}\end{aligned}$$

Additional calculations:

$$\begin{aligned}\Gamma &= \frac{i(T_{p1}) - i(T_{dp1})}{C_a [T_{p1} - T_1]} \\ &= \frac{120.42 - 41.4}{0.253(119.7 - 89.9)} \\ &= 10.48\end{aligned}$$

Other Performance

As discussed in Appendix A, the following equations must be solved simultaneously to determine h_d^* , ξ , and T_r .

$$\begin{aligned}\xi &= \frac{i(T_{p1}) - i(T_r)}{C_a (T_{p1} - T_r)} \\ h_d^* &= \left[\frac{1}{U_o^*} - \frac{1}{h_s a_s^*} - \frac{1}{U_{pt}} \right]^{-1} \\ T_r &= T_{p1} - \frac{Q_{rej}}{A_s} \left(\frac{1}{h_d^*} + \frac{1}{U_{pt}} \right)\end{aligned}$$

These equations must be solved by iteration. Since $T_r \sim T_d$, the initial assumption will be that $T_r = T_d$.

$$T_r = T_d = 108.1^\circ\text{F}$$

Using the same procedure for calculating air properties previously presented; the saturation pressure, humidity ratio, and enthalpy are found

$$P_{ws}(T_r) = 0.0823 \text{ atm}$$

$$H(T_r) = 0.057$$

$$i(T_r) = 89.16 \text{ Btu/lbm}$$

$$\xi = \frac{120.42 - 89.16}{0.253(119.7 - 108.1)} = 10.65$$

$$h_e = \left(\frac{1}{h_d} + \frac{1}{h_s} \right)^{-1}$$

$$h_d = \frac{h_d^*}{a_s^*}$$

The value of a_s^* must also be determined by iteration. Assume $a_s^* = 0.62$

$$h_d^* = \left(\frac{1}{10.65 \times 2.87} - \frac{1}{10.65 \times 0.62 \times 12.94} - 0.01168 \right)^{-1}$$

$$= 50.39 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

$$h_d = \frac{50.39}{0.62} = 81.28 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

$$h_e = \left(\frac{1}{81.28} + \frac{1}{10.65 \times 12.94} \right)^{-1} = 51.13 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

$$Bi^{1/2} = \ell f (h_e / k_f y_b)^{1/2}$$

$$Bi^{1/2*} = 0.03/25 \left(\frac{51.13}{128 \times 0.00033} \right)^{1/2} (\text{ft}) \frac{\text{Btu/hr ft}^2\text{-}^\circ\text{F}}{(\text{Btu/hr-ft-}^\circ\text{F})(\text{ft})}^{1/2}$$

$$Bi^{1/2*} = 1.08$$

$$\eta_f^* = 0.68 \text{ (From Figure A.3)}$$

$$a_s^* = \frac{A_{st} + \eta_f^* A_{sf}}{A_s} = \frac{61.87 + 0.68 (643.91)}{705.78}$$

$$a_s^* = 0.71$$

The next value of h_d^* may be calculated.

$$h_d^* = \left(\frac{1}{10.65 \times 2.87} - \frac{1}{10.65 \times 12.94 \times 0.71} - 0.01168 \right)^{-1}$$

$$= 92.29 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

Using this value of h_d^* a new T_r is calculated.

$$T_r = T_{p1} - \frac{Q_{rej}}{A_s} \left(\frac{1}{h_d} + \frac{1}{U_{pt}} \right)$$

$$= 119.7 - \frac{496827}{706} \left(\frac{1}{91.29} + 0.01168 \right)$$

$$T_r = 103.85^\circ\text{F}$$

A new value of ξ is calculated and the process is repeated until the exact values are converged upon. They are

$$T_r = 103.95^\circ\text{F}$$

$$\xi = 10.08$$

$$h_d^* = 92.05 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

$$a_s^* = 0.635$$

The following is a calculation of Q_w/Q_d . Q_w was already determined:

$$Q_w = Q_{rej} = 496827 \text{ Btu/hr}$$

Q_d is the estimated heat transferred under dry operation at the same pressure drop as under wet operation. An empirical equation was developed for the dry heat transfer coefficient with fin efficiency excluded.

$$\begin{aligned} h_s &= 18.0911 \Delta P^{0.3279} \\ &= 18.0911 (.36)^{0.3279} \\ &= 12.9413 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F} \end{aligned}$$

The overall heat transfer coefficient can be calculated:

$$U = \left[\frac{A_s}{h_s(A_{st} + \eta_f A_{sf})} + \frac{A_s}{h_p A_i} + \frac{A_s t_t}{A_{io} k_f} \right]^{-1}$$

where $\eta_f = -0.419(x) + 1.125$, where $x = h_s^{1/2}(0.1513)$

$$\eta_f = 0.90$$

$$\begin{aligned} U &= \left[\frac{706}{12.94(61.87 + 0.90[643.9])} + 0.0116 \right]^{-1} \\ U &= 10.34 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F} \end{aligned}$$

The NTU rating is:

$$N = \frac{UA_s}{\dot{m}_a C_a} = \frac{U}{Re_o} \frac{A_s D_h}{C_a \mu A_c}$$

where $Re_o = \frac{\dot{m}_a D_h}{A_c \mu}$ and is different from the wet condition Re and is found from an empirical relation

$$\begin{aligned} Re_o &= 2485.4291 \Delta P^{0.59566} \\ &= 2485.4291(0.36)^{0.59566} \\ Re_o &= 1,352 \end{aligned}$$

$$\begin{aligned} N &= \frac{10.34}{1352} (154.7) \frac{[Btu/hr-ft^2-^{\circ}F][ft^2]}{[lb/hr][Btu/lbm F]} \\ N &= 1.18 \end{aligned}$$

The surface effectiveness is:

$$\begin{aligned} \phi &= 1 - e^{-N} \\ \phi &= 1 - e^{-1.18} = 0.69 \end{aligned}$$

The dry heat transfer is:

$$\begin{aligned} Q_d &= \phi \frac{\mu A}{D} Re_o C_a (\Delta T) \\ Q_d &= \phi Re_o (T_{p1} - T_1) 4.562 \\ Q_d &= 0.69(1,352)(11.97 - 89.9)(4.562) \\ Q_d &= 126,830 \text{ Btu/hr} \end{aligned}$$

Therefore, the enhancement ratio is:

$$\begin{aligned} \frac{Q_w}{Q_d} &= \frac{496827}{126830} \\ &= 3.92 \end{aligned}$$

APPENDIX D

UNCERTAINTY ANALYSIS

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UNCERTAINTY ANALYSIS

All of the following uncertainty calculations follow the method outlined in Section 3.5.

Dry Heat Transfer:

The main parameter of interest in the dry heat transfer measurements is the overall heat transfer coefficient, U_o . U_o is calculated by the following equation:

$$U_o = \frac{Q_{rej}}{A_s \Delta T_m}$$

From this equation the uncertainty of U_o can be written

$$\frac{\delta U_o}{U_o} = \left[\left(\frac{\delta Q_{rej}}{Q_{rej}} \right)^2 + \left(\frac{\delta A_s}{A_s} \right)^2 + \left(\frac{\delta (\Delta T \ell_m)}{\Delta T \ell_m} \right)^2 \right]^{1/2}$$

The uncertainty in A_s is estimated as:

$$\frac{\delta A_s}{A_s} = \frac{10}{705.78} = 0.0142$$

The formula for Q_{rej} is:

$$Q_{rej} = Q_{in} + Q_{Peripheral} + \Delta Q$$

where ΔQ is included to account for the variation from equilibrium conditions. The uncertainty in Q_{rej} becomes:

$$\delta Q_{rej} = (\delta Q_{in}^2 + \delta Q_{Peripheral}^2 + \delta \Delta Q^2)^{1/2}$$

$$Q_{in} = m_h C_p \Delta T_h$$

$$\frac{\delta Q_{in}}{Q_{in}} = \left[\left(\frac{\delta m_h}{m_h} \right)^2 + \left(\frac{\delta C_p}{C_p} \right)^2 + \left(\frac{\delta \Delta T_h}{\Delta T_h} \right)^2 \right]^{1/2}$$

All the calculations done for dry heat transfer will use Test 1 (see Appendix C).

$$m_h = [\text{gpm}] [\text{lb/ft}^3] [\text{ft}^3/\text{gal}] [\text{min/hr}]$$

$$\frac{\delta m_h}{m_h} = \left[\left(\frac{\delta \text{gpm}}{\text{gpm}} \right)^2 + \left(\frac{\delta \rho}{\rho} \right)^2 \right]^{1/2} = \left[\left(\frac{0.5}{25} \right)^2 + \left(\frac{0.05}{61.84} \right)^2 \right]^{1/2}$$

$$\frac{\delta m_h}{m_h} = 0.02$$

$$\frac{\delta C_p}{C_p} = \frac{0.005}{1.0} = 0.005$$

$$\frac{\delta \Delta T_h}{\Delta T_h} = \frac{0.5}{5.22} = 0.0958$$

Therefore:

$$\frac{\delta Q_{in}}{Q_{in}} = \left[(0.020)^2 + (0.005)^2 + (0.0958)^2 \right]^{1/2}$$

$$\frac{\delta Q_{in}}{Q_{in}} = 0.098$$

Since $Q_{in} = 64500 \text{ Btu/hr}$

$$\delta Q_{in} = 6321 \text{ Btu/hr}$$

$\delta Q_{Peripheral}$ is estimated at 500 Btu/hr.

$\delta \Delta Q$ is an estimate of the observed variation from equilibrium in the system. Typically the temperature of the water in the main loop varied less than 0.1°C in 30 minutes. Therefore,

$$\begin{aligned}\delta \Delta Q &= \frac{0.1(1.8)}{30/60} \times 4670 \times 1.0 \\ &[\text{°F/hr}] [\text{lb}] [\text{Btu/lb-°F}] \\ &= 1,681 \text{ Btu/hr}\end{aligned}$$

The total uncertainty in Q_{rej} becomes:

$$\begin{aligned}\delta Q_{rej} &= [(6321)^2 + (500)^2 + (1,681)^2]^{1/2} \\ &= 6,560 \frac{\text{Btu}}{\text{hr}}\end{aligned}$$

and
$$\frac{\delta Q_{rej}}{Q_{rej}} = \frac{6,560}{84,300} = 0.078$$

The equation for ΔT_{lm} is:

$$\Delta T_{lm} = \frac{(T_{p1} - T_{\infty 1}) - (T_{p2} - T_{\infty 2})}{\ln \frac{T_{p1} - T_{\infty 1}}{T_{p2} - T_{\infty 2}}}$$

Assign:

$$u = (T_{p1} - t_{\infty 1}) - (T_{p2} - t_{\infty 2}) = \Delta T_1 - \Delta T_2$$

$$v = n \frac{(T_{p1} - t_{\infty 1})}{(T_{p2} - t_{\infty 2})} = \ln \frac{\Delta T_1}{\Delta T_2}$$

$$\delta \Delta T_{lm} = \left[\frac{\partial \Delta T_{lm}}{\partial \Delta T_1}^2 \delta \Delta T_1^2 + \frac{\partial \Delta T_{lm}}{\partial \Delta T_2}^2 \delta \Delta T_2^2 \right]^{1/2}$$

$$\frac{\partial \Delta T_{lm}}{\partial \Delta T_1} = \frac{v \frac{\partial u}{\partial \Delta T_1} - u \frac{\partial v}{\partial \Delta T_1}}{v^2}$$

$$\frac{\partial u}{\partial \Delta T_1} = 1 \quad \frac{\partial u}{\partial \Delta T_2} = -1$$

$$\frac{\partial v}{\partial \Delta T_1} = \frac{1}{\Delta T_1} \quad \frac{\partial v}{\partial \Delta T_2} = \frac{-1}{\Delta T_2}$$

Therefore:

$$\frac{\partial \Delta T_{lm}}{\partial \Delta T_1} = \frac{\ln \frac{\Delta T_1}{\Delta T_2} - \frac{(\Delta T_1 - \Delta T_2)}{\Delta T_1}}{\ln \left(\frac{\Delta T_1}{\Delta T_2} \right)^2} = \frac{1 - \frac{\Delta T_{lm}}{\Delta T_1}}{\ln \frac{\Delta T_1}{\Delta T_2}}$$

Likewise:

$$\frac{\partial \Delta T_{lm}}{\partial \Delta T_2} = \frac{\frac{\Delta T_{lm}}{\Delta T_2} - 1}{\ln \frac{\Delta T_1}{\Delta T_2}}$$

Finally

$$\frac{\delta \Delta T_{lm}}{\Delta T_m} = \left[\left(\frac{1 - \Delta T_{lm} / \Delta T_1}{\Delta T_1 - \Delta T_2} \right)^2 \delta \Delta T_1^2 + \left(\frac{\Delta T_{lm} / \Delta T_2 - 1}{\Delta T_1 - \Delta T_2} \right)^2 \delta \Delta T_2^2 \right]^{1/2}$$

For Test #1

$$T_{p1} = 120.2^\circ\text{F}$$

$$\Delta T_1 = 19.5^\circ\text{F}$$

$$t_{\infty 1} = 100.7^\circ\text{F}$$

$$T_{p2} = 119.6^\circ\text{F}$$

$$\Delta T_2 = 6.6^\circ\text{F}$$

$$T_{\infty 2} = 113.0^\circ\text{F}$$

$$\begin{aligned} \frac{\delta \Delta T_{lm}}{\Delta T_{lm}} &= \left[\left(\frac{1 - 11.9/19.5}{19.5 - 6.6} \right)^2 \delta \Delta T_1^2 + \left(\frac{11.9/6.6 - 1}{19.5 - 6.6} \right)^2 \delta \Delta T_2^2 \right]^{1/2} \\ &= \left[0.0009 \delta \Delta T_1^2 + 0.0039 \delta \Delta T_2^2 \right]^{1/2} \end{aligned}$$

Estimate $\delta \Delta T_1 = 1.0^\circ\text{F}$

$$\delta \Delta T_2 = 1.0^\circ\text{F}$$

$$\begin{aligned} \frac{\delta \Delta T_{lm}}{\Delta T_{lm}} &= \left[0.0009 (1.0)^2 + 0.0039 (1.0)^2 \right]^{1/2} \\ &= 0.069 \end{aligned}$$

The uncertainty in U_o can now be calculated:

$$\begin{aligned} \frac{\delta U_o}{U_o} &= \left[(0.078)^2 + (0.0142)^2 + (0.069)^2 \right]^{1/2} \\ &= 0.10 \end{aligned}$$

Wet heat transfer:

Two major parameters are of interest in wet heat transfer, namely h_d^* and U_o^* . Uncertainty analysis will be done for U_{o2} since it was used to demonstrate the variations in U_o^* in Section 4.

$$U_{o2}^* = \frac{m_a C_a}{A_s} \ln \left[1 - \frac{Q_{rej}}{m_a (i_{\infty 1} - i'_{p1})} \right]$$

$$\text{Define } x = \frac{m_a C_a}{A_s} \quad y = \frac{Q_{rej}}{m_a (i_{\infty 1} - i'_{p1})}$$

$$U_{o2}^* = x \ln [1-y]$$

$$\delta U_{o2}^* = \left[\left(\frac{\partial U_{o2}^*}{\partial x} \right)^2 \delta x^2 + \left(\frac{\partial U_{o2}^*}{\partial y} \right)^2 \delta y^2 \right]^{1/2}$$

$$\frac{\partial U_{o2}^*}{\partial x} = \ln [1-y] \quad \frac{\partial U_{o2}^*}{\partial y} = - \frac{x}{1-y}$$

$$\delta U_{o2}^* = \left[\left[(\ln [1-y])^2 \delta x^2 + \left(\frac{x}{1-y} \right)^2 \delta y^2 \right] \right]^{1/2}$$

$$\frac{\delta U_{o2}^*}{U_{o2}^*} = \left[\left(\frac{\delta x}{x} \right)^2 + \left(\frac{\delta y}{(1-y) \ln 1-y} \right)^2 \right]^{1/2}$$

$$\frac{\delta x}{x} = \left[\left(\frac{\delta m_a}{m_a} \right)^2 + \left(\frac{\delta C_a}{C_a} \right)^2 + \left(\frac{\delta A_s}{A_s} \right)^2 \right]^{1/2}$$

All calculations are done for test 2-1

$$\frac{\delta A_s}{A_s} = 0.0142$$

$$\frac{\delta C_a}{C_a} = \frac{0.001}{0.253} = 0.004$$

$$\frac{\delta \dot{m}_a}{\dot{m}_a} = 0.05 \quad (\text{estimated})$$

The value of $\delta \dot{m}_a / \dot{m}_a$ will vary with the magnitude of manometer reading. All things considered, 0.05 seemed to be a reasonable value.

$$\begin{aligned} \frac{\delta x}{x} &= \left[(0.0142)^2 + (0.004)^2 + (0.05)^2 \right]^{1/2} \\ &= 0.052 \end{aligned}$$

$$\frac{\delta y}{y} = \left[\left(\frac{\delta Q_{rej}}{Q_{rej}} \right)^2 + \left(\frac{\delta \dot{m}_a}{\dot{m}_a} \right)^2 + \left(\frac{\delta i_{\infty 1} - i_{p1}}{i_{\infty 1} - i_{p1}} \right)^2 \right]^{1/2}$$

$$Q_{rej} = \left[\delta Q_{in}^2 + \delta Q_{per}^2 + \Delta Q^2 \right]^{1/2}$$

$$\frac{\delta \dot{Q}_{in}}{Q_{in}} = \left[\left(\frac{\delta \dot{m}_h}{\dot{m}_h} \right)^2 + \left(\frac{\delta C_p}{C_p} \right)^2 + \left(\frac{\delta \Delta T_h}{\Delta T_h} \right)^2 \right]^{1/2}$$

$$\dot{m}_h = (\text{gpm})(\text{lb/ft}^3)(\text{ft}^3/\text{gal})(\text{min/hr})$$

$$\dot{m}_h = (25)(61.84)(0.1337)(60) = 12402$$

$$\frac{\delta \dot{m}_h}{\dot{m}_h} = \left[\left(\frac{\delta \text{gpm}}{\text{gpm}} \right)^2 + \left(\frac{\delta \rho}{\rho} \right)^2 \right]^{1/2} = \left[\left(\frac{0.5}{25} \right)^2 + \left(\frac{0.05}{61.84} \right)^2 \right]^{1/2} = 0.02$$

$$\frac{\delta C_p}{C_p} = \frac{0.005}{1} = 0.005$$

$$\frac{\delta \Delta T_h}{\Delta T_h} = \frac{0.5}{38.97} = 0.013$$

$$\frac{\delta Q_{in}}{Q_{in}} = \left[(0.02)^2 + (0.005)^2 + (0.013)^2 \right]^{1/2} = 0.0244$$

$$Q_{in} = \dot{m}_h C_p \Delta T_h$$

$$\dot{m}_h = 12402 \text{ lb/hr}$$

$$\Delta T_h = 38.91 \quad C_p = 1.0$$

$$Q_{in} = (12402)(38.91)(1) = 482562$$

$$\delta Q_{in} = 11775$$

$$\delta Q_{perph} = 500 \quad (\text{estimated})$$

$$\delta \Delta Q = 1681 \quad (\text{see page D.2})$$

$$\begin{aligned} \delta Q_{rej} &= \left[(11775)^2 + (500)^2 + (1681)^2 \right]^{1/2} \\ &= 11,905 \end{aligned}$$

$$\frac{\delta Q_{rej}}{Q_{rej}} = \frac{11,905}{500,016} = 0.024$$

$$\delta(i_{\infty 1} - i'_{p1}) = \left[(\delta i_{\infty 1})^2 + (\delta i_{p1})^2 \right]^{1/2}$$

$$\delta i_{\infty 1} = \delta i'_{p1} = 2.0 \text{ [from psychrometric chart]}$$

$$\delta(i_{\infty 1} - i'_{p1}) = 2.83$$

From Appendix C

$$i_{\infty 1} = 37.3 \text{ Btu/lb}$$

$$i'_{p1} = 120.42 \text{ Btu/lb}$$

Therefore:

$$\frac{\delta y}{y} = \left[(0.024)^2 + (0.05)^2 + \left(\frac{2.83}{120.42-37.3} \right)^2 \right]^{1/2}$$

$$= 0.065$$

$$y = \frac{Q_{rej}}{m_a(i_{\infty 1} - i_{p i})} = \frac{500016}{12794(120.42-37.3)}$$

$$= 0.4702$$

$$\delta y = 0.0306$$

Finally:

$$\frac{\delta U_{02}^*}{U_{02}^*} = \left[(0.052)^2 + \left(\frac{0.0306}{(1-0.4702) n(1-0.4702)} \right)^2 \right]^{1/2}$$

$$\frac{\delta U_{02}^*}{U_{02}^*} = 0.105$$

The uncertainty in the deluge coefficient h^* can be calculated as follows:

$$h_d^* = \left[\frac{1}{\rho U_o^*} - \frac{1}{\xi h_s a_s^*} - \frac{1}{U_{pt}} \right]^{-1}$$

$$= [x - y - z]^{-1}$$

$$\delta h_d^* = \left[\left(\frac{\partial h_d^*}{\partial x} \right)^2 \delta x^2 + \left(\frac{\partial h_d^*}{\partial y} \right)^2 \delta y^2 + \left(\frac{\partial h_d^*}{\partial z} \right)^2 \delta z^2 \right]^{1/2}$$

$$\frac{\partial h_d^*}{\partial x} = \frac{-\partial h_d^*}{\partial y} = \frac{-\partial h_d^*}{\partial z} = \frac{-1}{(x - y - z)^2}$$

$$\frac{\delta h_d^*}{h_d^*} = \left[\frac{\delta x^2 + \delta y^2 + \delta z^2}{(x - y - z)^2} \right]^{1/2}$$

$$\delta x^2 = \frac{1}{\xi^2 U_0^{*2}} \left[\frac{\delta \xi^2}{\xi^2} + \frac{\delta U_0^{*2}}{U_0^{*2}} \right]$$

$$\xi = \frac{i_{p1}' - i_r'}{c_a (T_{p1} - T_r)}$$

$$\frac{\delta \xi}{\xi} = 0.05 \quad [\text{estimate from psychrometric tables}]$$

From previous calculations:

$$\frac{\delta U_0^*}{U_0^*} = 0.105 \quad U_{02}^* = 2.87$$

$$\delta x^2 = \frac{1}{(10.08)^2} \frac{1}{(2.87)^2} (0.05)^2 + (0.105)^2$$

$$\delta x^2 = 1.61 \times 10^{-5}$$

$$\delta y^2 = \frac{1}{\xi^2 h_s^2 a_s^{*2}} \left(\frac{\delta \xi^2}{\xi^2} + \frac{\delta h_s^2}{h_s^2} + \frac{\delta a_s^{*2}}{a_s^{*2}} \right)$$

Assume: $\frac{\delta h_s}{h_s} = 0.10$

$$\frac{\delta a_s^*}{a_s^*} = 0.10$$

From previous calculations:

$$h_s = 12.94 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F} \quad a_s^* = 0.635$$

$$\delta y^2 = \frac{1}{(10.08)^2 (12.94)^2 (0.635)^2} \left[(0.05)^2 + (0.010)^2 + (0.10)^2 \right]$$

$$= 3.28 \times 10^{-6}$$

$$\delta z^2 = \frac{\delta U_{pt}^2}{U_{pt}^4}$$

From previous calculations:

$$\frac{\delta U_{pt}}{U_{pt}} = 0.10 \quad U_{pt} = 85.6 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

$$\delta z^2 = \frac{(0.10)^2}{(85.6)^2} = 1.36(10^{-6})$$

$$x = \frac{1}{U_o^*} = \frac{1}{10.08(2.87)} = 0.035$$

$$y = \frac{1}{h_s a_s^*} = \frac{1}{10.08(12.94)(0.635)} = 0.012$$

$$z = \frac{1}{U_{pt}} = \frac{1}{85.62} = 0.0117$$

Finally:

$$\frac{\delta h_d^*}{h_d^*} = \left[\frac{1.61 \times 10^{-5} + 3.28 \times 10^{-6} + 1.4 \times 10^{-6}}{(0.035 - 0.012 - 0.0117)^2} \right]^{1/2}$$

$$\frac{\delta h_d^*}{h_d^*} = 0.40$$

APPENDIX E

DEVELOPMENT OF HEAT EXCHANGER PERFORMANCE
AS A FUNCTION OF FAN POWER

APPENDIX E

DEVELOPMENT OF HEAT EXCHANGER PERFORMANCE AS A FUNCTION OF FAN POWER

The equations for the rate of heat transfer and fan power are

$$Q = \phi G_o A_f C_a \Delta T_1 \quad (E.1)$$

$$P = \frac{\sigma A_f v^2 G_o^3}{2 g} K + \frac{f A_s}{\sigma A_f} \quad (E.2)$$

An expression for G_o as a function of fan power is obtained by using the following assumptions

$$K \ll \frac{f A_s}{\sigma A_f} \quad (E.3)$$

$$f = f_r \left(\frac{Re}{1000} \right) \quad (E.4)$$

$$Re = \frac{D_h G_o}{\mu} \quad (E.5)$$

$$A_s = s A_f L \quad (E.6)$$

The expression for G_o is then

$$G_o = \left[P \left(\frac{500 g_c \mu}{\sigma L A_f v^2 f_r} \right) \left(\frac{D_h}{1000 \mu} \right)^{(1+m)} \right]^{\left(\frac{1}{3-m} \right)} \quad (E.7)$$

The following substitutions and assumptions may then be used to develop an expression for $Q/\Delta T_i$ as a function of fan power. For the limiting case where

$$\left(\frac{T_{p2} - T_{p1}}{T_2 - T_1} \right) \approx 0 \quad (E.8)$$

(as in a condenser, or as in the WATA tests where m_p is very high), the effectiveness may be given by

$$\phi = 1 - e^{-N} \quad (E.9)$$

where N is defined by

$$N = \frac{h_o A_s}{m_a C_a} = \frac{4 h_o L}{G_o D_h C_a} \quad (E.10)$$

The Colburn factor is defined by

$$j_o = \frac{h_o}{C_a G_o} Pr^{2/3} \quad (E.11)$$

where it is assumed that j_o is of the form

$$j_o = j_r \left(\frac{Re}{1000} \right)^{-n} \quad (E.12)$$

Then h_o can be given by

$$h_o = C_a Pr^{-2/3} j_r \left(\frac{1000 \mu}{D_h} \right)^n G_o^{(1-n)} \quad (E.13)$$

Using Equation (E.13) for h_o , Equation (E.9) and (E.10) for ϕ , Equation (E.1) may be given as follows,

$$Q = A_f G_o C_a \Delta T_1 \left\{ 1 - \exp \left[- \left(\frac{4Lj_r}{\mu Pr^{2/3}} \right) \left(\frac{1000 \mu}{D_h} \right)^{(1+n)} G_o^{-n} \right] \right\} \quad (E.14)$$

Combining Equation (E.14) with Equation (E.7) for G_o then gives

$$\frac{Q}{C_a \Delta T_1 A_f} = C_1 P^{\left(\frac{1}{3-m}\right)} \left\{ 1 - \exp \left[-C_2 P^{\left(\frac{-n}{3-m}\right)} \right] \right\} \quad (E.15)$$

where the constants C_1 and C_2 are defined as follows:

$$C_1 = \sigma \left[\frac{1000 \mu g_c}{2\sigma L A_f v^2 f_r} \left(\frac{D_h}{1000 \mu} \right)^{(1+m)} \right]^{1/(3-m)} \quad (E.16)$$

$$C_2 = \left(\frac{C_1}{\sigma C_p} \right)^{-n} \left[\frac{4j_r Pr^{-2/3}}{1000 \mu} \left(\frac{1000 \mu}{D_h} \right)^{(1+n)} \right] \quad (E.17)$$

The expression for the effectiveness - Ntu relation for crossflow with both fluids unmixed involves an infinite series for which graphs are available. The expression for counterflow yields values of ϕ that are less than 10 low for values of $Ntu < 5$. This expression is:

$$\phi = \frac{1 - \exp \left[-Ntu \left(1 - \frac{\beta_w}{\beta_a} \right) \right]}{1 - \frac{\beta_w}{\beta_a} \exp \left[-Ntu \left(1 - \frac{\beta_w}{\beta_a} \right) \right]} \quad \text{for } \beta_w < \beta_a \quad (E.18)$$

The resulting expression for Q as a function of P is:

$$\frac{Q}{\Delta T_1 C_a A_f} = \frac{C_1 P^{1/(3-m)} \left\{ 1 - \exp \left[-C_2 P^{-n/(3-m)} \left(1 - C_3 P^{1/(3-m)} \right) \right] \right\}}{1 - C_3 P^{1/(3-m)} \exp \left\{ C_2 P^{-n/(3-m)} \left[1 - C_3 P^{1/(3-m)} \right] \right\}} \quad (E.19)$$

where C_3 is defined by

$$C_3 = \frac{C_1}{m_p C_p}$$

This is identical with the use of the log-mean temperature difference as the driving force for heat transfer. For crossflow heat exchangers, this, together with the crossflow correction factor F_T , can be used to characterize the type of heat exchangers of interest. Thus $Q/\Delta T_i C_2 A_f$ can be calculated by including F_T in the expression for C_2 ; i.e.:

$$C_2 = \left(\frac{C_1}{\phi} \right)^{-n} \left[\frac{4 F_T j_r L P r^{-2/3}}{1000 \mu} \left(\frac{1000 \mu}{D_h} \right)^{1+n} \right] \quad (E.21)$$

This requires an iterative procedure of: 1) assuming a value for F_T , 2) verifying this value following the calculation of Q and the resultant temperature rise of stream representing C_{min} , and 3) repeating the calculation as necessary.

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