Augmentation of in-tube evaporation and condensation with micro-fin tubes using refrigerants R-113 and R-22

Jatin C. Khanpara

Iowa State University
INFORMATION TO USERS

This reproduction was made from a copy of a manuscript sent to us for publication and microfilming. While the most advanced technology has been used to photograph and reproduce this manuscript, the quality of the reproduction is heavily dependent upon the quality of the material submitted. Pages in any manuscript may have indistinct print. In all cases the best available copy has been filmed.

The following explanation of techniques is provided to help clarify notations which may appear on this reproduction.

1. Manuscripts may not always be complete. When it is not possible to obtain missing pages, a note appears to indicate this.

2. When copyrighted materials are removed from the manuscript, a note appears to indicate this.

3. Oversize materials (maps, drawings, and charts) are photographed by sectioning the original, beginning at the upper left hand corner and continuing from left to right in equal sections with small overlaps. Each oversize page is also filmed as one exposure and is available, for an additional charge, as a standard 35mm slide or in black and white paper format.*

4. Most photographs reproduce acceptably on positive microfilm or microfiche but lack clarity on xerographic copies made from the microfilm. For an additional charge, all photographs are available in black and white standard 35mm slide format.*

*For more information about black and white slides or enlarged paper reproductions, please contact the Dissertations Customer Services Department.

UMI Dissertation Information Service
University Microfilms International
A Bell & Howell Information Company
300 N. Zeeb Road, Ann Arbor, Michigan 48106
Khanpara, Jatin C.

AUGMENTATION OF IN-TUBE EVAPORATION AND CONDENSATION WITH MICRO-FIN TUBES USING REFRIGERANTS R-113 AND R-22

Iowa State University

University Microfilms International 300 N. Zeeb Road, Ann Arbor, MI 48106
PLEASE NOTE:

In all cases this material has been filmed in the best possible way from the available copy. Problems encountered with this document have been identified here with a check mark ✓.

1. Glossy photographs or pages _____
2. Colored illustrations, paper or print _____
3. Photographs with dark background ✓
4. Illustrations are poor copy _____
5. Pages with black marks, not original copy _____
6. Print shows through as there is text on both sides of page _____
7. Indistinct, broken or small print on several pages ✓
8. Print exceeds margin requirements _____
9. Tightly bound copy with print lost in spine _____
10. Computer printout pages with indistinct print _____
11. Page(s) _________ lacking when material received, and not available from school or author.
12. Page(s) _________ seem to be missing in numbering only as text follows.
13. Two pages numbered _______. Text follows.
14. Curling and wrinkled pages _____
15. Dissertation contains pages with print at a slant, filmed as received _______
16. Other __________________________________________________________
    __________________________________________________________
    __________________________________________________________

University
Microfilms
International
Augmentation of in-tube evaporation and condensation with micro-fin tubes using refrigerants R-113 and R-22

by

Jatin C. Khanpara

A Dissertation Submitted to the Graduate Faculty in Partial Fulfillment of the Requirements for the Degree of DOCTOR OF PHILOSOPHY

Major: Mechanical Engineering

Approved:

Signature was redacted for privacy.

In Charge of Major Work

Signature was redacted for privacy.

For the Major Department

Signature was redacted for privacy.

For the Graduate College

Iowa State University
Ames, Iowa

1986
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>DEDICATION</td>
<td>xviii</td>
</tr>
<tr>
<td>NOMENCLATURE</td>
<td>xix</td>
</tr>
<tr>
<td>CHAPTER I. INTRODUCTION</td>
<td>1</td>
</tr>
<tr>
<td>The Problem</td>
<td>1</td>
</tr>
<tr>
<td>Objectives of Study</td>
<td>4</td>
</tr>
<tr>
<td>CHAPTER II. GENERAL REVIEW OF IN-TUBE TWO-PHASE HEAT TRANSFER</td>
<td>8</td>
</tr>
<tr>
<td>AND PRESSURE DROP FOR SMOOTH AND AUGMENTED TUBES</td>
<td></td>
</tr>
<tr>
<td>Smooth Tube Heat Transfer</td>
<td>8</td>
</tr>
<tr>
<td>Single-phase</td>
<td>8</td>
</tr>
<tr>
<td>Evaporation</td>
<td>9</td>
</tr>
<tr>
<td>Correlations for evaporation heat transfer</td>
<td>21</td>
</tr>
<tr>
<td>Condensation</td>
<td>30</td>
</tr>
<tr>
<td>Correlations for condensation heat transfer</td>
<td>33</td>
</tr>
<tr>
<td>Pressure Drop Studies</td>
<td>36</td>
</tr>
<tr>
<td>Introduction</td>
<td>36</td>
</tr>
<tr>
<td>Background</td>
<td>37</td>
</tr>
<tr>
<td>Correlations for smooth tube pressure drop</td>
<td>38</td>
</tr>
<tr>
<td>Augmented Heat Transfer</td>
<td>40</td>
</tr>
<tr>
<td>Single-phase</td>
<td>43</td>
</tr>
<tr>
<td>Augmented Evaporation</td>
<td>44</td>
</tr>
<tr>
<td>Introduction</td>
<td>44</td>
</tr>
<tr>
<td>Internally finned tubes</td>
<td>44</td>
</tr>
<tr>
<td>Micro-finned tubes</td>
<td>48</td>
</tr>
<tr>
<td>Augmented Condensation</td>
<td>49</td>
</tr>
<tr>
<td>Introduction</td>
<td>49</td>
</tr>
<tr>
<td>Internally finned tubes</td>
<td>54</td>
</tr>
<tr>
<td>Micro-finned tubes</td>
<td>54</td>
</tr>
<tr>
<td>CHAPTER III. AUGMENTATION OF R-113 IN-TUBE EVAPORATION AND</td>
<td>56</td>
</tr>
<tr>
<td>CONDENSATION WITH MICRO-FIN TUBES (WATER HEATED/COOLED SHORT TEST SECTION)</td>
<td></td>
</tr>
<tr>
<td>Introduction</td>
<td>56</td>
</tr>
<tr>
<td>Experimental Facility</td>
<td>57</td>
</tr>
<tr>
<td>General</td>
<td>57</td>
</tr>
<tr>
<td>Refrigerant loop</td>
<td>57</td>
</tr>
<tr>
<td>Section</td>
<td>Page</td>
</tr>
<tr>
<td>------------------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>Test section</td>
<td>62</td>
</tr>
<tr>
<td>Water loop</td>
<td>64</td>
</tr>
<tr>
<td>Experimental Procedure</td>
<td>65</td>
</tr>
<tr>
<td>Single-phase</td>
<td>65</td>
</tr>
<tr>
<td>Evaporation/condensation</td>
<td>66</td>
</tr>
<tr>
<td>Data Reduction</td>
<td>66</td>
</tr>
<tr>
<td>Single-phase heat transfer</td>
<td>67</td>
</tr>
<tr>
<td>Two-phase heat transfer</td>
<td>68</td>
</tr>
<tr>
<td>Results for Smooth Tube</td>
<td>70</td>
</tr>
<tr>
<td>Single-phase</td>
<td>70</td>
</tr>
<tr>
<td>Evaporation</td>
<td>70</td>
</tr>
<tr>
<td>Condensation</td>
<td>70</td>
</tr>
<tr>
<td>Results for Micro-fin Tubes in Single-Phase Flow</td>
<td>76</td>
</tr>
<tr>
<td>Results for Micro-fin Tubes in Evaporation</td>
<td>82</td>
</tr>
<tr>
<td>General</td>
<td>85</td>
</tr>
<tr>
<td>Analysis of results</td>
<td>92</td>
</tr>
<tr>
<td>Effect of geometrical parameters</td>
<td>98</td>
</tr>
<tr>
<td>Results for Micro-fin Tubes in Condensation</td>
<td>104</td>
</tr>
<tr>
<td>General</td>
<td>104</td>
</tr>
<tr>
<td>Analysis of results</td>
<td>108</td>
</tr>
<tr>
<td>Factors responsible for condensation enhancement</td>
<td>115</td>
</tr>
<tr>
<td>Effect of geometrical parameters</td>
<td>120</td>
</tr>
<tr>
<td>Conclusions</td>
<td>124</td>
</tr>
<tr>
<td>Single-phase study</td>
<td>124</td>
</tr>
<tr>
<td>Evaporation study</td>
<td>124</td>
</tr>
<tr>
<td>Condensation study</td>
<td>125</td>
</tr>
<tr>
<td>CHAPTER IV. AUGMENTATION OF R-113 IN-TUBE EVAPORATION WITH A MICRO-FIN TUBE (ELECTRICALLY HEATED LONG TEST SECTION)</td>
<td>127</td>
</tr>
<tr>
<td>Introduction</td>
<td>127</td>
</tr>
<tr>
<td>Experimental Facility</td>
<td>127</td>
</tr>
<tr>
<td>General</td>
<td>127</td>
</tr>
<tr>
<td>Refrigerant loop</td>
<td>128</td>
</tr>
<tr>
<td>Test section</td>
<td>131</td>
</tr>
<tr>
<td>Guard heater</td>
<td>133</td>
</tr>
<tr>
<td>Experimental Procedure</td>
<td>138</td>
</tr>
<tr>
<td>Single-phase</td>
<td>138</td>
</tr>
<tr>
<td>Evaporation</td>
<td>140</td>
</tr>
<tr>
<td>Section</td>
<td>Page</td>
</tr>
<tr>
<td>------------------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>Data Reduction</td>
<td></td>
</tr>
<tr>
<td>Single-phase heat transfer</td>
<td>142</td>
</tr>
<tr>
<td>Local evaporation heat transfer</td>
<td>144</td>
</tr>
<tr>
<td>Results for Smooth Tube</td>
<td></td>
</tr>
<tr>
<td>Heat transfer</td>
<td>146</td>
</tr>
<tr>
<td>Pressure drop</td>
<td>147</td>
</tr>
<tr>
<td>Results for Micro-fin Tube</td>
<td></td>
</tr>
<tr>
<td>Heat transfer</td>
<td>164</td>
</tr>
<tr>
<td>Pressure drop</td>
<td>164</td>
</tr>
<tr>
<td>Comparison Between Short and Long Test Sections</td>
<td></td>
</tr>
<tr>
<td>Smooth tube</td>
<td>171</td>
</tr>
<tr>
<td>Micro-finned tube</td>
<td>180</td>
</tr>
<tr>
<td>Conclusions</td>
<td></td>
</tr>
<tr>
<td>Single-phase study</td>
<td>187</td>
</tr>
<tr>
<td>Evaporation study</td>
<td>187</td>
</tr>
<tr>
<td>CHAPTER V. AUGMENTATION OF R-22 IN-TUBE EVAPORATION WITH A MICRO-FIN TUBE (ELECTRICALLY HEATED LONG TEST SECTION)</td>
<td>189</td>
</tr>
<tr>
<td>Introduction</td>
<td></td>
</tr>
<tr>
<td>Experimental Facility</td>
<td>189</td>
</tr>
<tr>
<td>Refrigerant loop</td>
<td>189</td>
</tr>
<tr>
<td>Test section</td>
<td>190</td>
</tr>
<tr>
<td>Condenser loop</td>
<td>194</td>
</tr>
<tr>
<td>Experimental Procedure</td>
<td>198</td>
</tr>
<tr>
<td>Single-phase</td>
<td>198</td>
</tr>
<tr>
<td>Evaporation</td>
<td>198</td>
</tr>
<tr>
<td>Results for Smooth Tube</td>
<td></td>
</tr>
<tr>
<td>Heat transfer</td>
<td>200</td>
</tr>
<tr>
<td>Pressure drop</td>
<td>200</td>
</tr>
<tr>
<td>Results for Micro-finned Tube</td>
<td></td>
</tr>
<tr>
<td>Heat transfer</td>
<td>209</td>
</tr>
<tr>
<td>Conclusions</td>
<td>223</td>
</tr>
<tr>
<td>CHAPTER VI. AUGMENTATION OF R-22 IN-TUBE EVAPORATION AND CONDENSATION WITH A MICRO-FIN TUBE (WATER HEATED/COOLED LONG TEST SECTION)</td>
<td>225</td>
</tr>
<tr>
<td>Introduction</td>
<td>225</td>
</tr>
</tbody>
</table>
Pre-heater/Pre-evaporator Tube 299
After-condenser/Condenser 299
Annulus Side Heat Exchanger 300
Recirculating Pump for Water Loop 300
Accumulator 300
Power Control Unit for Boiler 301
Instrumentation 301
Data acquisition system 301
Temperature 302
Pressure 303
Flow measurements 304

APPENDIX C. EXPERIMENTAL PROCEDURE 307

Initial Testing of R-113 Test Rig 307
Removal of Noncondensable Gases 308
Flow Stability 309
Consistency and Repeatability 310

APPENDIX D. SAMPLE CALCULATIONS AND ERROR ANALYSIS FOR TEST APPARATUS USING R-113 323

Test Apparatus for Average Heat Transfer Coefficients 323
Sample calculation 323
Propagation of error 328

Test Apparatus for Local Heat Transfer Coefficients 335
Sample calculation 335
Propagation of error 341

APPENDIX E. DETAILS OF EXPERIMENTAL LOOP (R-22) COMPONENTS 345

Pump 345
After-condenser/Condenser 345
Boiler 346
<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-12 Refrigeration Unit</td>
<td>346</td>
</tr>
<tr>
<td>Superheater</td>
<td>348</td>
</tr>
<tr>
<td>Recirculating Pump</td>
<td>348</td>
</tr>
<tr>
<td>Accumulator</td>
<td>349</td>
</tr>
<tr>
<td>Instrumentation</td>
<td>349</td>
</tr>
<tr>
<td>Data acquisition system</td>
<td>349</td>
</tr>
<tr>
<td>Temperature measurements</td>
<td>350</td>
</tr>
<tr>
<td>Pressure measurements</td>
<td>350</td>
</tr>
<tr>
<td>Flow measurements</td>
<td>351</td>
</tr>
<tr>
<td>APPENDIX F. DATA REDUCTION COMPUTER PROGRAM LISTINGS</td>
<td>353</td>
</tr>
<tr>
<td>Water Heated/Cooled Short Test Section (R-113 as a refrigerant)</td>
<td>354</td>
</tr>
<tr>
<td>Single-phase</td>
<td>354</td>
</tr>
<tr>
<td>Evaporation</td>
<td>361</td>
</tr>
<tr>
<td>Condensation</td>
<td>368</td>
</tr>
<tr>
<td>Electrically Heated Long Test Section (R-113 as a refrigerant)</td>
<td>369</td>
</tr>
<tr>
<td>Single-phase</td>
<td>369</td>
</tr>
<tr>
<td>Evaporation</td>
<td>375</td>
</tr>
<tr>
<td>Electrically Heated Long Test Section (R-22 as a refrigerant)</td>
<td>382</td>
</tr>
<tr>
<td>Water Heated/Cooled Long Test Section (R-22 as a refrigerant)</td>
<td>383</td>
</tr>
<tr>
<td>Single-phase</td>
<td>383</td>
</tr>
<tr>
<td>Evaporation</td>
<td>389</td>
</tr>
<tr>
<td>Condensation</td>
<td>396</td>
</tr>
</tbody>
</table>
# LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Figure 1.1.</td>
<td>A typical tube cross section, side view, and fin profile</td>
<td>2</td>
</tr>
<tr>
<td>Figure 3.1.</td>
<td>Schematic diagram of flow loop</td>
<td>59</td>
</tr>
<tr>
<td>Figure 3.2.</td>
<td>A photographic view of the test apparatus</td>
<td>60</td>
</tr>
<tr>
<td>Figure 3.3.</td>
<td>Details of the test section</td>
<td>63</td>
</tr>
<tr>
<td>Figure 3.4.</td>
<td>Comparison of experimental Nusselt numbers against predictions of the Dittus-Boelter/McAdams and Petukhov-Popov equations</td>
<td>71</td>
</tr>
<tr>
<td>Figure 3.5.</td>
<td>Evaporation heat transfer coefficients for smooth tube and augmented Tube 5</td>
<td>73</td>
</tr>
<tr>
<td>Figure 3.6.</td>
<td>Comparison of evaporation heat transfer coefficients with predictions of Kandlikar (1983)</td>
<td>74</td>
</tr>
<tr>
<td>Figure 3.7.</td>
<td>Comparison of smooth tube pressure drop data with predictions of the Lockhart-Martinelli (1949) correlation</td>
<td>75</td>
</tr>
<tr>
<td>Figure 3.8.</td>
<td>Condensation coefficients for smooth tube and augmented Tube 7</td>
<td>77</td>
</tr>
<tr>
<td>Figure 3.9.</td>
<td>Comparison of condensation heat transfer coefficient data with predictions of Shah (1979), Traviss (1972), and Cavallini et al. (1974)</td>
<td>78</td>
</tr>
<tr>
<td>Figure 3.10.</td>
<td>A comparison of pressure drop for smooth tube with the Lockhart-Martinelli (1949) correlation</td>
<td>79</td>
</tr>
<tr>
<td>Figure 3.11.</td>
<td>Fin profiles for enhanced tubes</td>
<td>83</td>
</tr>
<tr>
<td>Figure 3.12.</td>
<td>Single-phase enhanced heat transfer results</td>
<td>84</td>
</tr>
<tr>
<td>Figure 3.13.</td>
<td>Evaporation heat transfer coefficients at low mass velocity</td>
<td>86</td>
</tr>
<tr>
<td>Figure 3.14.</td>
<td>Evaporation heat transfer coefficients at medium mass velocity</td>
<td>87</td>
</tr>
<tr>
<td>Figure 3.15.</td>
<td>Evaporation heat transfer coefficients at high mass velocity</td>
<td>88</td>
</tr>
<tr>
<td>Figure</td>
<td>Description</td>
<td>Page</td>
</tr>
<tr>
<td>--------</td>
<td>------------------------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>3.16.</td>
<td>Evaporation pressure drops at low mass velocity</td>
<td>89</td>
</tr>
<tr>
<td>3.17.</td>
<td>Evaporation pressure drops at medium mass velocity</td>
<td>90</td>
</tr>
<tr>
<td>3.18.</td>
<td>Evaporation pressure drops at high mass velocity</td>
<td>91</td>
</tr>
<tr>
<td>3.19.</td>
<td>Effect of surface area on evaporation enhancement factors at low mass velocity</td>
<td>99</td>
</tr>
<tr>
<td>3.20.</td>
<td>Effect of surface area on evaporation enhancement factors at medium mass velocity</td>
<td>100</td>
</tr>
<tr>
<td>3.21.</td>
<td>Effect of surface area on evaporation enhancement factors at high mass velocity</td>
<td>101</td>
</tr>
<tr>
<td>3.22.</td>
<td>Condensation heat transfer coefficients at low mass velocity</td>
<td>105</td>
</tr>
<tr>
<td>3.23.</td>
<td>Condensation heat transfer coefficients at medium mass velocity</td>
<td>106</td>
</tr>
<tr>
<td>3.24.</td>
<td>Condensation heat transfer coefficients at high mass velocity</td>
<td>107</td>
</tr>
<tr>
<td>3.25.</td>
<td>Condensation pressure drops at low mass velocity</td>
<td>112</td>
</tr>
<tr>
<td>3.26.</td>
<td>Condensation pressure drops at medium mass velocity</td>
<td>113</td>
</tr>
<tr>
<td>3.27.</td>
<td>Condensation pressure drops at high mass velocity</td>
<td>114</td>
</tr>
<tr>
<td>3.28.</td>
<td>Effect of surface area on condensation enhancement factors at low mass velocity</td>
<td>117</td>
</tr>
<tr>
<td>3.29.</td>
<td>Effect of surface area on condensation enhancement factors at medium mass velocity</td>
<td>118</td>
</tr>
<tr>
<td>3.30.</td>
<td>Effect of surface area on condensation enhancement factors at high mass velocity</td>
<td>119</td>
</tr>
<tr>
<td>4.1.</td>
<td>Schematic diagram of flow loop for testing enhanced tubing in evaporation</td>
<td>129</td>
</tr>
<tr>
<td>4.2.</td>
<td>A photographic view of the test loop used for determining the local evaporation heat transfer coefficients</td>
<td>130</td>
</tr>
<tr>
<td>4.3.</td>
<td>Location of thermocouples and pressure taps along the test section</td>
<td>134</td>
</tr>
<tr>
<td>Figure 4.4.</td>
<td>Guard heater cross section</td>
<td>137</td>
</tr>
<tr>
<td>Figure 4.5.</td>
<td>Effect of the guard heater on the wall temperature profile at medium mass velocity</td>
<td>139</td>
</tr>
<tr>
<td>Figure 4.6.</td>
<td>Wall temperature profile for single-phase heat transfer test using the long test section</td>
<td>148</td>
</tr>
<tr>
<td>Figure 4.7.</td>
<td>Comparison of experimental single-phase heat transfer coefficient with predictions of Dittus-Boelter/McAdams correlation</td>
<td>149</td>
</tr>
<tr>
<td>Figure 4.8.</td>
<td>Comparison of experimental single-phase Nusselt numbers with predictions of Petukhov-Popov correlation</td>
<td>150</td>
</tr>
<tr>
<td>Figure 4.9.</td>
<td>Wall temperature profile for evaporation heat transfer at medium mass flow rate using the long test section</td>
<td>151</td>
</tr>
<tr>
<td>Figure 4.10.</td>
<td>Local evaporation heat transfer coefficients for the smooth tube at low, medium, and high mass velocities</td>
<td>153</td>
</tr>
<tr>
<td>Figure 4.11.</td>
<td>Comparison between experimental and predicted evaporation heat transfer coefficients at low mass velocity</td>
<td>154</td>
</tr>
<tr>
<td>Figure 4.12.</td>
<td>Comparison between experimental and predicted evaporation heat transfer coefficients at medium mass velocity</td>
<td>155</td>
</tr>
<tr>
<td>Figure 4.13.</td>
<td>Comparison between experimental and predicted evaporation heat transfer coefficients at high mass velocity</td>
<td>156</td>
</tr>
<tr>
<td>Figure 4.14.</td>
<td>Effect of heat flux on evaporation heat transfer coefficient for the long, smooth tube at low mass velocity</td>
<td>157</td>
</tr>
<tr>
<td>Figure 4.15.</td>
<td>Effect of heat flux on evaporation heat transfer coefficient for the long, smooth tube at medium mass velocity</td>
<td>158</td>
</tr>
<tr>
<td>Figure 4.16.</td>
<td>Effect of heat flux on evaporation heat transfer coefficient for the long, smooth tube at high mass velocity</td>
<td>159</td>
</tr>
<tr>
<td>Figure 4.17.</td>
<td>Wall temperature profile for low mass flow rate at the dryout condition</td>
<td>161</td>
</tr>
</tbody>
</table>
Figure 4.18. Comparison of smooth tube evaporation pressure drop with the predictions of Lockhart-Martinelli (1949) 163

Figure 4.19. Single-phase heat transfer coefficients for micro-fin Tube 10 for the long test section 165

Figure 4.20. Evaporation heat transfer coefficients for micro-fin Tube 10 for the long test section 166

Figure 4.21. Evaporation pressure drop for micro-fin Tube 10 for the long test section 172

Figure 4.22. Comparison of single-phase heat transfer coefficients for the short and long smooth tubes 173

Figure 4.23. Comparison of evaporation heat transfer coefficients for the short and long smooth tubes at low mass velocity 174

Figure 4.24. Comparison of evaporation heat transfer coefficients for the short and long smooth tubes at medium mass velocity 175

Figure 4.25. Comparison of evaporation heat transfer coefficients for the short and long smooth tubes at high mass velocity 176

Figure 4.26. Comparison of evaporation pressure drops for the short and long smooth tubes 179

Figure 4.27. Comparison of single-phase heat transfer results for the short and long micro-fin tubes (Tube 10) 181

Figure 4.28. Comparison of evaporation heat transfer coefficients for the short and long micro-fin tubes (Tube 10) at low mass velocity 182

Figure 4.29. Comparison of evaporation heat transfer coefficients for the short and long micro-fin tubes (Tube 10) at medium mass flow velocity 183

Figure 4.30. Comparison of evaporation heat transfer coefficients for the short and long micro-fin tubes (Tube 10) at high mass velocity 184

Figure 4.31. Comparison of evaporation pressure drops for the short and long micro-fin tubes (Tube 10) 186

Figure 5.1. Refrigerant R-22 test loop for evaporation heat transfer 191
| Figure 5.2. | A photographic view of test apparatus | Page 192 |
| Figure 5.3. | Location of thermocouples and pressure taps on the test tube | Page 196 |
| Figure 5.4. | Wall temperature profile for single-phase heat transfer in a smooth tube | Page 201 |
| Figure 5.5. | Comparison of experimental single-phase heat transfer coefficient data with Dittus-Boelter/McAdams correlation for the smooth tube | Page 202 |
| Figure 5.6. | Comparison of experimental single-phase Nusselt numbers with Petukhov-Popov correlation for the smooth tube | Page 203 |
| Figure 5.7. | Wall temperature profile for evaporation heat transfer at low mass flow rate for the smooth tube | Page 204 |
| Figure 5.8. | Evaporation heat transfer coefficients at low, medium, and high mass velocities for the smooth tube | Page 205 |
| Figure 5.9. | Effect of heat flux on local evaporation heat transfer coefficients for the smooth tube at low mass velocity | Page 206 |
| Figure 5.10. | Effect of heat flux on local evaporation heat transfer coefficients for the smooth tube at medium mass velocity | Page 207 |
| Figure 5.11. | Effect of heat flux on evaporation heat transfer coefficients for the smooth tube at high mass velocity | Page 208 |
| Figure 5.12. | Comparison of evaporation heat transfer data with predictions of Kandlikar (1983) | Page 210 |
| Figure 5.13. | Comparison of evaporation heat transfer data with predictions of Shah (1982) | Page 211 |
| Figure 5.14. | Comparison of experimental data with predictions of Pujol and Stenning (1969) | Page 212 |
| Figure 5.15. | Comparison of evaporation pressure drop with predictions of Lockhart-Martinelli (1949) equation | Page 213 |
| Figure 5.16. | Single-phase heat transfer coefficients for micro-fin Tube 10 | Page 215 |
| Figure 5.17 | Evaporation heat transfer coefficients for micro-fin Tube 10 at low, medium, and high mass velocities | 216 |
| Figure 5.18 | Comparison of enhancement factors for evaporation heat transfer using refrigerants R-113 and R-22 | 221 |
| Figure 5.19 | Pressure-density dependence of R-113 and R-22 | 222 |
| Figure 6.1  | Annulus design drawing | 226 |
| Figure 6.2  | Graph indicating Wilson plot technique | 229 |
| Figure 6.3  | A schematic of the R-22 test loop | 231 |
| Figure 6.4  | A photographic view of the test apparatus | 232 |
| Figure 6.5  | Details of the test section | 235 |
| Figure 6.6  | Spacer details | 236 |
| Figure 6.7  | A schematic of test apparatus for annulus calibration tests | 239 |
| Figure 6.8  | Calibration of annulus at low mass flow rate | 244 |
| Figure 6.9  | Calibration of annulus at medium mass flow rate | 245 |
| Figure 6.10 | Calibration of annulus at high mass flow rate | 246 |
| Figure 6.11 | Comparison of experimental annulus side heat transfer coefficient data against predictions of McAdams (1942) and Kays and Leung (1963) | 248 |
| Figure 6.12 | Comparison of single-phase heat transfer coefficients data with predictions of Dittus-Boelter/McAdams correlation | 260 |
| Figure 6.13 | Comparison of single-phase Nusselt numbers with predictions of Petukhov-Popov correlation | 261 |
| Figure 6.14 | Average evaporation heat transfer coefficients for the smooth tube | 262 |
| Figure 6.15 | Comparison of average condensation heat transfer coefficients with predictions of Kandlikar (1983), Shah (1982), and Pujol and Stenning (1969) | 263 |
| Figure 6.16. | Average condensation heat transfer coefficients for the smooth tube | Page 265 |
| Figure 6.17. | Comparison of average condensation heat transfer coefficients with predictions of Shah (1979), Traviss et al. (1972), and Cavallini et al. (1974) | Page 266 |
| Figure 6.18. | Comparison of average evaporation and condensation pressure drop with predictions of Lockhart-Martinelli (1949) equation | Page 267 |
| Figure 6.19. | Single-phase heat transfer coefficients for micro-fin Tube 10 | Page 268 |
| Figure 6.20. | Average evaporation heat transfer coefficients for micro-fin Tube 10 | Page 270 |
| Figure 6.21. | Average condensation heat transfer coefficients for micro-fin Tube 10 | Page 271 |
| Figure A.1. | Effect of mass velocity on local evaporation heat transfer at constant (low) system pressure and constant heat flux | Page 291 |
| Figure A.2. | Effect of mass velocity on local evaporation heat transfer at constant (high) system pressure and constant heat flux | Page 292 |
| Figure A.3. | Effect of heat flux on local evaporation heat transfer at a constant (low) mass velocity and constant average system pressure | Page 293 |
| Figure A.4. | Effect of heat flux on local evaporation heat transfer at a constant (high) mass velocity and constant average system pressure | Page 294 |
| Figure A.5. | Effect of system pressure on evaporation heat transfer at constant (low) mass velocity and constant heat flux | Page 295 |
| Figure A.6. | Effect of average system pressure on local evaporation heat transfer at constant (high) mass velocity and constant heat flux | Page 296 |
| Figure C.1. | Comparison of single-phase heat transfer coefficients with Dittus-Boelter/McAdams correlation | Page 311 |
| Figure C.2. | Wilson plot for calculating refrigerant side heat transfer coefficient | Page 312 |
| Figure C.3. | Repeatability test for evaporation heat transfer coefficients with the smooth tube | 313 |
| Figure C.4. | Repeatability test for condensation heat transfer coefficients with the smooth tube | 314 |
| Figure C.5. | Repeatability test for evaporation pressure drop with the smooth tube | 315 |
| Figure C.6. | Repeatability test for condensation pressure drop with the smooth tube | 316 |
| Figure C.7. | Repeatability test for single-phase heat transfer with micro-fin Tube 9 | 317 |
| Figure C.8. | Repeatability test for condensation heat transfer coefficients with micro-fin Tube 9 | 318 |
| Figure C.9. | Repeatability test for evaporation heat transfer coefficients with micro-fin Tube 9 | 319 |
| Figure C.10. | Repeatability test for evaporation pressure drop with micro-fin Tube 9 | 320 |
| Figure C.11. | Repeatability test for condensation pressure drop using micro-fin Tube 9 | 321 |
# LIST OF TABLES

| Table 2.1. | Summary of evaporation heat transfer studies of smooth tubes with R-113 | Page |
| Table 2.2. | Summary of evaporation heat transfer studies of smooth tubes with R-22 | 14 |
| Table 2.3. | Summary of evaporation heat transfer studies of smooth tubes with R-11 and R-12 | 18 |
| Table 2.4. | Summary of correlations for predicting smooth tube evaporation heat transfer coefficients | 22 |
| Table 2.5. | Summary of condensation heat transfer studies of smooth tube with refrigerants | 31 |
| Table 2.6. | Summary correlation for predicting smooth tube condensation heat transfer coefficients | 34 |
| Table 2.7. | Values of exponents m,n and constants $C'_1$, $C'_g$ for the Lockhart-Martinelli parameter in various flow types | 41 |
| Table 2.8. | Summary of evaporation studies with internally finned tubes | 45 |
| Table 2.9. | Summary of geometrical dimensions characteristic of micro-fin tubes for evaporation | 47 |
| Table 2.10. | Summary of condensation heat transfer studies with internally finned tubes | 50 |
| Table 2.11. | Summary of the geometrical characteristics of micro-fin tubes used for condensation | 53 |
| Table 3.1. | Operating parameter range for water-jacketed evaporation/condensation test facility (R-113 as a refrigerant) | 58 |
| Table 3.2. | Selected geometrical parameters of the tubes | 80 |
| Table 3.3. | Evaporation heat transfer (pressure drop) enhancement factors for low mass velocity | 93 |
| Table 3.4. | Evaporation heat transfer (pressure drop) enhancement factors for medium mass velocity | 94 |
| Table 3.5. | Evaporation heat transfer (pressure drop) enhancement factors for high mass velocity | 95 |
Table 3.6. Tube performance ranking for evaporation heat transfer and pressure drop 97
Table 3.7. Condensation heat transfer (pressure drop) enhancement factors at low mass velocity 109
Table 3.8. Condensation heat transfer (pressure drop) enhancement factors at medium mass velocity 110
Table 3.9. Condensation heat transfer (pressure drop) enhancement factors at high mass velocity 111
Table 3.10. Tube performance ranking for condensation heat transfer and pressure drop 116
Table 4.1. Operating parameter range for electrically heated evaporation test facility (R-113 as a refrigerant) 132
Table 4.2. Dryout heat flux for different mass velocities (smooth tubes) 162
Table 4.3. Local heat transfer enhancement factors for micro-fin Tube 10 at low mass velocity 168
Table 4.4. Local heat transfer enhancement factors for micro-fin Tube 10 at medium mass velocity 169
Table 4.5. Local heat transfer enhancement factors for micro-fin Tube 10 at high mass velocity 170
Table 4.6. Evaporation heat transfer enhancement factors for micro-fin Tube 10 using the short test section 185
Table 5.1. Operating parameter range for electrically heated evaporation test facility (R-22 as a refrigerant) 193
Table 5.2. Local heat transfer enhancement factors for micro-fin Tube 10 at low mass velocity 218
Table 5.3. Local heat transfer enhancement factors for micro-fin Tube 10 at medium mass velocity 219
Table 5.4. Local heat transfer enhancement factors for micro-fin Tube 10 at high mass velocity 220
Table 6.1. Operating parameter range for water-jacketed evaporation/condensation test facility (R-22 as a refrigerant) 230
Table 6.2. Annulus calibration test results 247
DEDICATION

This dissertation is dedicated to my parents, Champaklal and Jasvanti Khanpara
**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Expression</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>cross-sectional flow area</td>
<td></td>
</tr>
<tr>
<td>Bo</td>
<td>Boiling number (nondimensional), ( q / G )</td>
<td></td>
</tr>
<tr>
<td>b</td>
<td>fin width</td>
<td></td>
</tr>
<tr>
<td>C'</td>
<td>specific heat</td>
<td></td>
</tr>
<tr>
<td>C_1</td>
<td>constant in Eq. (2.44)</td>
<td></td>
</tr>
<tr>
<td>C_g</td>
<td>constant in Eq. (2.44)</td>
<td></td>
</tr>
<tr>
<td>Co</td>
<td>Convection number (nondimensional), ( \left( \frac{1 - X}{X} \right)^{0.8} \left( \frac{\rho_g}{\rho_1} \right)^{0.5} )</td>
<td></td>
</tr>
<tr>
<td>D</td>
<td>tube diameter</td>
<td></td>
</tr>
<tr>
<td>D1-D6</td>
<td>constants in Kandlikar's (1983) equation</td>
<td></td>
</tr>
<tr>
<td>dP</td>
<td>change in pressure</td>
<td></td>
</tr>
<tr>
<td>dZ</td>
<td>change in length</td>
<td></td>
</tr>
<tr>
<td>e</td>
<td>fin height</td>
<td></td>
</tr>
<tr>
<td>F</td>
<td>two-phase Reynolds number function, a constant in Eqs. (2.15) and (2.16)</td>
<td></td>
</tr>
<tr>
<td>F_{fl}</td>
<td>a constant in Kandlikar (1983) equation</td>
<td></td>
</tr>
<tr>
<td>Fl</td>
<td>Flux number (nondimensional), ( G i_{fg} / q )</td>
<td></td>
</tr>
<tr>
<td>F_1, F_2</td>
<td>constants in Traviss et al. (1972) equation</td>
<td></td>
</tr>
<tr>
<td>Fr</td>
<td>Froude number (nondimensional), ( G^2 / \rho_1 g D_1 )</td>
<td></td>
</tr>
<tr>
<td>f</td>
<td>Fanning friction factor (nondimensional), Eq. (2.5)</td>
<td></td>
</tr>
<tr>
<td>G</td>
<td>mass velocity</td>
<td></td>
</tr>
<tr>
<td>g</td>
<td>acceleration due to gravity</td>
<td></td>
</tr>
<tr>
<td>g</td>
<td>gravitational constant</td>
<td></td>
</tr>
<tr>
<td>h</td>
<td>heat transfer coefficient</td>
<td></td>
</tr>
<tr>
<td>i_{fg}</td>
<td>enthalpy of vaporization</td>
<td></td>
</tr>
<tr>
<td>J</td>
<td>mechanical equivalent of heat</td>
<td></td>
</tr>
<tr>
<td>K</td>
<td>thermal conductivity</td>
<td></td>
</tr>
</tbody>
</table>
$K_f$ constant, $J \Delta i_{fg} g_c / L g$
$L$ heated length of the test section
$LMTD$ log mean temperature difference
$l$ tube wall thickness
$m$ mass flow rate
$N$ number of sections, a constant in Eqs. (2.19) and (2.25)
$Nu$ Nusselt number (nondimensional), $h D_i / K$
$n$ number of fins
$P$ pressure
$\tilde{P}$ reduced pressure (nondimensional), $P/P_{cr}$
$Pr$ Prandtl number (nondimensional), $C_p \mu / K$
$Pe$ perimeter
$p$ fin pitch
$Q$ heat transfer rate
$q$ heat flux
$Re$ Reynolds number (nondimensional), $\rho VD / \mu$
$S$ bubble suppression factor
$T$ temperature
$U$ overall heat transfer coefficient
$u$ velocity
$V$ voltage, velocity
$v$ specific volume
$W$ width
$X$ quality
$X_{tt}$ Martinelli parameter (liquid phase turbulent/vapor phase turbulent)
Subscripts

- av: average
- aug: augmented
- b: bubble
- bs: bubble suppression
- boiler: boiler
- cb: convective boiling
- conv: convection
- cr: critical
- c/s: cross section
- D: based on diameter
- D.B.: Dittus-Boelter
- e: equivalent
- F: test fluid
- f: friction
- g: vapor
- h: based on hydraulic diameter ($= 4 A_{c/s}/Pe$)
- in: inlet
- i: inside
- L: local
- LO: assuming total flow to be liquid
- l: liquid phase
- latent: latent
- location: location on the test section
- m: momentum
- mac: macro-convective
<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Full Form</th>
</tr>
</thead>
<tbody>
<tr>
<td>mic</td>
<td>micro-convective</td>
</tr>
<tr>
<td>nb</td>
<td>nucleate boiling</td>
</tr>
<tr>
<td>nucl</td>
<td>nucleate</td>
</tr>
<tr>
<td>o</td>
<td>outside</td>
</tr>
<tr>
<td>out</td>
<td>outlet</td>
</tr>
<tr>
<td>pre</td>
<td>preheater</td>
</tr>
<tr>
<td>s</td>
<td>shunt</td>
</tr>
<tr>
<td>sat</td>
<td>saturated</td>
</tr>
<tr>
<td>sect</td>
<td>sectional</td>
</tr>
<tr>
<td>sens</td>
<td>sensible</td>
</tr>
<tr>
<td>sh</td>
<td>shell</td>
</tr>
<tr>
<td>smooth</td>
<td>smooth tube</td>
</tr>
<tr>
<td>sup</td>
<td>superheater</td>
</tr>
<tr>
<td>sur</td>
<td>surface</td>
</tr>
<tr>
<td>T</td>
<td>terminal</td>
</tr>
<tr>
<td>TP</td>
<td>two-phase</td>
</tr>
<tr>
<td>t</td>
<td>tip (peak) of fin geometry</td>
</tr>
<tr>
<td>tt</td>
<td>liquid phase turbulent/vapor phase turbulent</td>
</tr>
<tr>
<td>tv</td>
<td>liquid phase turbulent/vapor phase viscous</td>
</tr>
<tr>
<td>test</td>
<td>test tube</td>
</tr>
<tr>
<td>tube</td>
<td>tube</td>
</tr>
<tr>
<td>vt</td>
<td>liquid phase viscous/vapor phase turbulent</td>
</tr>
<tr>
<td>vv</td>
<td>liquid phase viscous/vapor phase turbulent</td>
</tr>
<tr>
<td>v</td>
<td>valley of fin geometry</td>
</tr>
<tr>
<td>w</td>
<td>wall</td>
</tr>
<tr>
<td>W</td>
<td>water</td>
</tr>
</tbody>
</table>
Greek symbols

\( \alpha \)  
- helix angle

\( \beta \)  
- spiral angle

\( \gamma \)  
- lead angle

\( \Delta \)  
- change in value

\( \delta \)  
- fin height

\( \mu \)  
- viscosity

\( \rho \)  
- density

\( \sigma \)  
- surface tension

\( \phi \)  
- two-phase friction multiplier, Eqs. (2.36) and (2.38)

\( \chi \)  
- Lockhart-Martinelli parameter, Eq. (2.39)

\( \psi \)  
- vapor Reynolds number/liquid Reynolds number \((Re_v/Re_l)\), 
  two-phase multiplier in Eq. (2.18)
CHAPTER I. INTRODUCTION

The Problem

Augmentation techniques can reduce heat exchanger size, decrease pumping power, or reduce approach temperature difference. Therefore, augmentation or enhancement of heat transfer has gained significant importance in recent years due to increasing energy and material costs. A recent study by Bergles et al. (1983) reported an exponential growth in augmentation heat transfer literature to a level of 3,045 technical publications, not including patents and manufacturer's literature.

Various techniques have been investigated over a period of years for augmenting two-phase heat transfer as applied to refrigeration and air-conditioning systems (Bergles et al., 1981). One of the most promising techniques for condensers and evaporators is the use of internally finned tubes. The selection of a specific fin configuration is usually a compromise between increasing the heat transfer, which is desirable, and increasing the pressure drop, which undesirably increases the pumping power. In recent years, tubes variously referred to as "spiral grooved," "multi-grooved," "rippled finned," and "inner-grooved" have received considerable attention due to their excellent thermal-hydraulic performance. The characteristic that distinguishes these "micro-fin" tubes from other form of internally finned tubes, e.g., those tested by Kubanek and Miletti (1979), is that the fins are smaller (typical heights of less than 0.007 in. or 0.18 mm) and more numerous (typically greater than 60 fins). Cross-sectional views of a typical micro-fin tube are shown in Fig. 1.1.
Figure 1.1. A typical tube cross section, side view, and fin profile
Heat exchanger units in commercial refrigeration and air-conditioning systems operate with annular flow conditions over a significant part of the overall tube length. Also, the liquid annular film attached to the tube wall is the major thermal resistance effecting heat transfer (Rohsenow et al., 1983). Therefore, heat transfer in micro-fin tubes is increased not only because of larger surface areas but also because the liquid film is disturbed. In general, heat transfer enhancements of approximately 50 to 100% (i.e., enhancement factors of 1.5 to 2.0) relative to the smooth tube have been recorded using these micro-fin tubes. The pressure drop increases for these same tubes are approximately 40-80% referenced to a smooth tube (Tojo et al., 1984).

Past research directed towards improving the thermal-hydraulic performance of micro-fin tubes is limited. Specifically, earlier studies consisted mainly of measuring the heat transfer and pressure drop of individual tubes rather than performing experiments on a large group of tubes which would have made it possible to study the effects of fin shape. In addition, a comparison of enhancement factors for two different refrigerants (e.g., R-113 and R-22) using the same tube geometry has not been reported in literature. Finally, there appear to be no experimental data available on single-phase heat transfer using these micro-fin tubes. Single-phase heat transfer data are important since subcooled liquid and superheated vapor generally exist in evaporators and condensers.
Objectives of Study

The objectives of the present investigation are

1. To screen the micro-fin tubes having different fin geometries using water heated/cooled short test section (41 in. or 1.0 m long) and refrigerant R-113 as a test fluid. The geometrical parameters of interest are: peak shape, valley shape, fin height, number of fins, and spiral angle.

2. To verify the short length test section apparatus using local evaporation with a 12.5 ft (3.81 m) long electrically heated test section with R-113 as a test fluid.

3. To determine the local evaporation heat transfer coefficients using a 12.1 ft (3.68 m) long electrically heated test section with R-22 as a test fluid. Also, to compare the local evaporation enhancement factors using two different refrigerants, i.e., R-113 and R-22. Additionally, to compare these local evaporation heat transfer coefficients against the average values attained using a water heated 12 ft (3.65 m) long test section with R-22 as a test fluid.

4. To simulate commercial refrigeration and air-conditioning evaporators/condensers using a water heated/cooled 12 ft (3.65 m) long test section.

5. To attain single-phase heat transfer coefficients for enhanced tubes; subcooled liquid/superheated vapor
conditions generally exist in actual refrigeration and air-conditioning systems.

In order to achieve the above goals, an experimental research program was initiated in the Heat Transfer Laboratory of the Department of Mechanical Engineering and Engineering Research Institute at Iowa State University. The program was divided into two main phases, each of which involved the design, construction, and operation of two different test apparatus.

In the first phase, an experimental facility was designed and built to obtain average thermal-hydraulic data for R-113 flowing inside short length (i.e., 41 in. or 1.0 m) tubes. In addition to a smooth tube, nine enhanced tubes with varied fin geometries were tested for single-phase, evaporation, and condensing flows. A statistical analysis was then used to analyze the effects of various fin geometrical parameters. Local evaporation enhancement factors for a 12.5 ft (3.81 m) long test section were also obtained by modifying the R-113 experimental facility to accommodate direct electrical heating of the test tube.

In the second phase, a more elaborate test facility was designed and built with R-22 as the test fluid. The test facility was operated at relatively higher system pressures (100-350 psig or 0.68-2.41 MPa), which was required because of the thermophysical properties of R-22. An electrically heated 12.1 ft (3.65 m) long test section was used to obtain local evaporation enhancement factors. In addition, the R-22 test apparatus was modified to perform evaporation and condensation tests using water in an outer annulus as the heat source/sink. These
experiments were performed at conditions comparable to actual commercial refrigeration and air-conditioning systems.

A brief review of literature for smooth and internally finned tubes is described in Chapter II, with more emphasis on refrigerants and annular flows. Chapter III describes the test apparatus and the results of the screening tests performed on 41 in. (1.0 m) long test sections with R-113 as a test fluid. The experimental facility and the local evaporation test results using R-113 as a test fluid are discussed in Chapter IV. A systematic comparison of the evaporation tests using short and long length test section is also presented.

Subsequently, the test apparatus for determining the local evaporation heat transfer coefficients using R-22 as a test fluid is described in Chapter V. A systematic comparison of the local evaporation heat transfer coefficients using two different refrigerants (i.e., R-113 and R-22) is also discussed. Chapter VI describes the test flow loop built to conduct two-phase heat transfer and pressure drop tests using 12 ft (3.61 m) long water heated/cooled test sections with R-22 as a test fluid. A comparison of the local and average evaporation heat transfer coefficients is also reported. Additionally, increases in the condensation heat transfer coefficients using two different refrigerants (i.e., R-22 and R-113) is qualitatively discussed.

Finally, general conclusions of the present investigation with recommendations are reported in Chapter VII. The details of various components used in four different test rigs and the sample calculations together with the propagation of error are discussed in the Appendices. The experimental data obtained using these four different test rigs are
recorded in the Heat Transfer Laboratory of the Department of Mechanical Engineering at Iowa State University, Ames, Iowa.
CHAPTER II. GENERAL REVIEW OF IN-TUBE TWO-PHASE HEAT TRANSFER
AND PRESSURE DROP FOR SMOOTH AND AUGMENTED TUBES

Smooth Tube Heat Transfer

Single-phase

Extensive experimental and theoretical work has been conducted by researchers on forced convection heat transfer and pressure drop in smooth, circular ducts with turbulent flow. Transition to turbulent flow in ducts starts at a Reynolds number of approximately 2300 and then reaches a fully turbulent condition at $5 \times 10^4 < Re < 10^5$ (Shah and Johnson, 1981). This Reynolds number range may vary depending on the degree of turbulence of the incoming flow and the shape of the inlet section. Shah and Johnson (1981) carried out an extensive review of literature on in-tube, forced convection heat transfer. In this section, some of the correlations relevant to the present study for single-phase heat transfer are briefly discussed.

The classical Dittus-Boelter equation (Dittus and Boelter, 1930) can be stated as

$$Nu = 0.0265 \, Re^{0.8} \, Pr^{0.3} \quad \text{(for cooling)} \quad (2.1)$$

and,

$$Nu = 0.0243 \, Re^{0.8} \, Pr^{0.4} \quad \text{(for heating)} \quad (2.2)$$

McAdams (1933) slightly modified this equation as follows:
Nu = 0.023 Re^{0.8} Pr^n \quad (2.3)

where n = 0.3 for cooling and 0.4 for heating. Equation (2.3) is referred to here as the Dittus-Boelter/McAdams correlation. Using a semi-empirical correlation developed by Prandtl (1944),

\[
Nu = \frac{(f/8)(Re \Pr)}{1 + 8.7(f/8)^{0.5}(Pr - 1)} \quad (2.4)
\]

Petukhov (1970) suggested the following equation:

\[
Nu = \frac{(f/8)(Re \Pr)}{1 + 12.7(f/8)^{0.5}(Pr^{2/3} - 1)} \quad (2.5)
\]

where

\[f = (1.82 \log_{10}Re - 1.64)^{-2.0}\]

This equation is based on fully developed turbulent flow. Shah (1979) concluded that the Dittus-Boelter/McAdams correlation overpredicts experimental data by approximately 20% and strongly recommends the Petukhov-Popov correlation, Eq. (2.5). However, in the present study, both Eqs. (2.3) and (2.5) are used to check single-phase heat transfer coefficients.

Evaporation

Introduction Both boiling and thin film evaporation or convective vaporization occur in refrigerant evaporators. The latter
region is characterized by annular flow and it occupies the largest region in refrigerant evaporators. Therefore, more stress is given to evaporation with annular flow in the present discussion. In the annular flow region, a thin liquid film is normally found on the heating surface while vapor and liquid droplets occupy the center of the tube. Heat is transferred by conduction and convection through this thin liquid film and vaporization occurs at the liquid/vapor interface. Nucleate boiling within the film is usually suppressed. Relative to single-phase flow, higher heat transfer coefficients are attained during this forced convective evaporation process.

Over a period of years, numerous correlations have been developed to predict heat transfer coefficients for a variety of fluids. Therefore, considering this large quantity of literature, studies related only to refrigerants R-113, R-22, R-11, and R-12 are presented in this section. In addition, special emphasis has been placed on understanding the effects of parameters such as mass velocity, quality, and heat flux on evaporation heat transfer. The selection of the test fluid and the effects of parameters was based on the objectives of the present research work. The detailed survey of the literature presented in this section was also very valuable in designing the test apparatus.

R-113 The details of various experimental studies conducted using R-113 as a test fluid are reviewed in this section. The ranges of the experimental parameters and the tube dimensions are listed in Table 2.1. Highlights of some of the experimental research work relevant to the present study are reported in the following section.
<table>
<thead>
<tr>
<th>Author</th>
<th>Type of System</th>
<th>Method of Heating</th>
<th>Orientation</th>
<th>Tube Geometry</th>
<th>Saturation Pressure psia (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Blatt and Adt (1964)</td>
<td>Compressor</td>
<td>Water</td>
<td>Horizontal</td>
<td>0.25 in. (6.35 mm) I.D., 0.18 in. (4.57 mm) long</td>
<td>63.8-101.5 (0.44-0.70)</td>
</tr>
<tr>
<td>Gouse and Comou (1965)</td>
<td>Pump</td>
<td>Electrical</td>
<td>Horizontal</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>Gouse and Dickson (1966)</td>
<td>Pump</td>
<td>Electrical</td>
<td>Horizontal</td>
<td>0.443 in. (11.34 mm) I.D., 20 ft (6.14 m) long</td>
<td>---</td>
</tr>
<tr>
<td>Pujol and Stenning (1969)</td>
<td>Pump</td>
<td>Electrical</td>
<td>Vertical</td>
<td>---</td>
<td>17-64.2 (0.12-0.44)</td>
</tr>
<tr>
<td>Reidle and Purcupile (1973)</td>
<td>Pump</td>
<td>Electrical</td>
<td>Horizontal</td>
<td>0.26-0.744 in. (6.65-19.04 mm) I.D., 12 ft (3.69 m) long</td>
<td>107.1 (0.74)</td>
</tr>
<tr>
<td>Mass Velocity (lbm/(hr*ft^2)) x 10^{-5}</td>
<td>Heat Flux (Btu/(hr*ft^2)) x 10^{-3}</td>
<td>G</td>
<td>Inlet, Outlet Conditions</td>
<td>q</td>
<td>Data Correlated</td>
</tr>
<tr>
<td>----------------------------------------</td>
<td>-------------------------------------</td>
<td>---</td>
<td>--------------------------</td>
<td>---</td>
<td>------------------</td>
</tr>
<tr>
<td>43-54 \times 10^{-3} \times 10^{-5}</td>
<td>\text{Low}</td>
<td>2-15</td>
<td>Yes</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(5.83-7.32)</td>
<td>(6.31-47.33)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>\text{inlet-quality, super-heated outlet}</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.73-9.09 \times 10^{-3} \times 10^{-5}</td>
<td>\text{Subcooled}</td>
<td>1.8-8.0</td>
<td>No</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(0.41-1.23)</td>
<td>(5.68-25.24)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>X_{\text{out}} = 40</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.59-2.53 \times 10^{-3} \times 10^{-5}</td>
<td>\text{Subcooled}</td>
<td>4.1-8.4</td>
<td>Yes</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(0.22-0.34)</td>
<td>(12.94-26.50)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>X_{\text{out}} = 74</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.47-13.96 \times 10^{-3} \times 10^{-5}</td>
<td>X_{\text{in}} = 0</td>
<td>1.8-17.76</td>
<td>Yes</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(0.20-1.89)</td>
<td>(5.68-56.03)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>X_{\text{out}} = 70</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>9.85 \times 10^{-3} \times 10^{-5}</td>
<td>\text{Subcooled}</td>
<td>3.8</td>
<td>No</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(1.33)</td>
<td>(11.99)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Gouse and Coumou (1965) undertook an experimental investigation to understand the mechanism of boiling. They observed flow patterns in a smooth tube and found that annular flow conditions prevailed for most of their experiments. They noticed that in subcooled boiling, bubbles do not collapse as they detach from the heated wall, but rather continue to grow while moving downstream in the tube. In the nucleate boiling region, the liquid was slightly superheated with respect to the saturation temperature. However, the amount of superheat decreased towards the end of the heated section.

This study was extended by Gouse and Dickson (1966). They suggested void fraction as an important parameter for subcooled and low quality boiling. However, for the annular flow boiling case the use of quality was recommended. In the final phase of their research (Dickson and Gouse, 1967), it was concluded that with increases in quality, the local heat transfer coefficient reaches a maximum value and then either levels off or decreases. The decrease of heat transfer coefficients at higher qualities was quite distinct from the dryout phenomenon that is caused by a complete evaporation of the liquid film at some critical value of quality. With electrically heated tubes, dryout is characterized by a sudden rise in tube wall temperature due to a significant decrease in the heat transfer coefficient. From their experimental data the local heat transfer coefficient was observed to be a function of both heat flux and mass velocity. The authors considered several correlations and concluded that Chen's (1966) correlation provided the best fit for predicting heat transfer coefficients.
A large difference in the values of the heat transfer coefficient for R-113 flowing vertically upwards or downwards was observed by Pujol and Stenning (1969). In addition, similar to the observations of Gouse and Dickson (1966), the heat transfer coefficient was noted to be a strong function of quality in the nucleation suppression region. In contrast, quality had an insignificant effect on heat transfer in the nucleate boiling region. Separate correlations were developed for each flow direction. The correlation developed for nucleate suppressed boiling in the vertical upward direction is

\[
\frac{h_{TP}}{h_1} = 4.0 \left( \frac{1}{X_{tt}} \right)^{0.37} \tag{2.6}
\]

where

\[
h_1 = 0.023 (Re_D)^{0.8} (Pr_1)^{0.4} (K_1/D)
\]

R-22 Details of experimental evaporation studies conducted using R-22 as a test fluid are reported in this section. The ranges of the experimental parameters and the tube dimensions are listed in Table 2.2.

Pierre (1956) correlated experimental data for average heat transfer coefficients for R-22 undergoing complete evaporation. This correlation is
<table>
<thead>
<tr>
<th>Author</th>
<th>Type of System</th>
<th>Method of Heating</th>
<th>Orientation</th>
<th>Tube Geometry</th>
<th>Saturation Temperature $T_{sat}$ °F (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pierre (1956)</td>
<td>Compressor</td>
<td>------</td>
<td>Horizontal</td>
<td>0.47-0.70 in. (12.0-17.92 mm) I.D., 14-34.5 ft (4.30-10.60 m) long</td>
<td>14-32 (-10.0)</td>
</tr>
<tr>
<td>Altman et al. (1960)</td>
<td>Pump</td>
<td>Water</td>
<td>Horizontal with U-bend</td>
<td>0.343 in. (8.8 mm) I.D., Two 4 ft (1.22 m) long test sections</td>
<td>40-75 (4.4-24)</td>
</tr>
<tr>
<td>Johnston and Chaddock (1964)</td>
<td>Natural circulation</td>
<td>Electrical</td>
<td>Horizontal</td>
<td>0.5 in. (12-70 mm) O.D., 12 ft (3.68 m) long</td>
<td>-55-26 (-48 -3)</td>
</tr>
<tr>
<td>Lavin and Young (1965)</td>
<td>Pump</td>
<td>Electrical</td>
<td>Horizontal</td>
<td>0.235 in. (6.01 mm) O.D.</td>
<td>75-100 (24-38)</td>
</tr>
</tbody>
</table>
### Mass Velocity

\[ \text{(lb m/ (hr ft²))} \times 10^{-5} \]

### Heat Flux

\[ \text{(Btu/hr ft²)} \times 10^{-3} \]

<table>
<thead>
<tr>
<th>Inlet Conditions</th>
<th>Outlet Conditions</th>
<th>Heat Flux ( q )</th>
<th>Correlated</th>
</tr>
</thead>
<tbody>
<tr>
<td>[ 15-100 ]</td>
<td>[ 0.368-9.5 ]</td>
<td>[ 0.368-9.5 ]</td>
<td>Yes</td>
</tr>
<tr>
<td>[ (0.03-0.27) ]</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>[ 1-60 ]</td>
<td>[ 2.0-20 ]</td>
<td>[ 2.0-20 ]</td>
<td>Yes</td>
</tr>
<tr>
<td>[ (0.03-0.26) ]</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>[ 43-72 ]</td>
<td>[ 0.5-6.82 ]</td>
<td>[ 0.5-6.82 ]</td>
<td>Yes</td>
</tr>
<tr>
<td>[ (0.02-0.05) ]</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>[ 17-100 ]</td>
<td>[ 0.4-120.0 ]</td>
<td>[ 0.4-120.0 ]</td>
<td>Yes</td>
</tr>
<tr>
<td>[ (0.54-3.2) ]</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Author</td>
<td>Type of System</td>
<td>Method of Heating</td>
<td>Orientation</td>
</tr>
<tr>
<td>-----------------------------</td>
<td>----------------</td>
<td>-------------------</td>
<td>-------------</td>
</tr>
<tr>
<td>Anderson, Rich, and Geary</td>
<td>Compressor</td>
<td>---</td>
<td>Horizontal</td>
</tr>
<tr>
<td>(1966)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Staub and Zuber (1966)</td>
<td>Pump</td>
<td>Electrical</td>
<td>Horizontal</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Kubanek and Miletti (1979)</td>
<td>Compressor</td>
<td>Water</td>
<td>Horizontal</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mass Velocity (lbm/(hr*ft(^2)))</td>
<td>Heat Flux (Btu/(hr*ft(^2)))</td>
<td></td>
<td></td>
</tr>
<tr>
<td>----------------------------------</td>
<td>--------------------------------</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(10^{-5})</td>
<td>(x10^{-3})</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inlet temperature ((kg/(m(^2)*s))</td>
<td>Inlet x (10^{-3}) )</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Outlet G Conditions</td>
<td>Outlet (kW/m(^2)) Data</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.23-2.5  (0.01-0.03)</td>
<td>3.0-7.2  (9.47-22.72)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.6-6.0   (0.01-0.08)</td>
<td>1.3-23.0  (4.10-72.57)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.5-20    (0.01-0.27)</td>
<td>36.52    (115.22)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
\[ \frac{h_{av}}{k_1} = 0.00082 \left[ \left( \frac{GD}{\mu_1} \right)^2 \left( \frac{\Delta x_{fg}^c}{L g} \right) \right]^{0.4} \]  

(2.7)

for \(10^9 < Re^2 K_f < 0.7 \times 10^{12}\)

Altman et al. (1960) developed the following correlation for local and average heat transfer coefficients:

\[ \frac{hD}{k_1} = 0.0225 \left( \frac{GD}{\mu_1} \right)^{3/4} \left( \frac{\Delta x_{fg}^c}{L} \right)^{3/8} \]  

(2.8)

for \(2.5 \times 10^{10} < Re^2 K_f < 1.5 \times 10^{12}\)

The experimental data attained during their investigation were approximately 15\% higher than the predictions of Pierre. Tests conducted by Johnston and Chaddock (1964) using both R-22 and R-12 were carried out in either mist or stratified flow conditions. The resulting two correlations for R-22 are

\[ Nu_1 = 0.000711 Re_1 K_f^{0.5} \]  

(2.9)

for \(10^9 < Re^2 K_f < 7 \times 10^{10}\)

and for R-12 (for the same range of \(Re^2 K_f\))

\[ Nu_1 = 0.000634 Re_1 K_f^{0.5} \]  

(2.10)
Altman et al. (1960) reported the heat transfer coefficients were independent of quality at low mass velocities. They concluded that the increased wetting due to the annular film and turbulence in the liquid film justified the dependence of heat transfer coefficient on quality in the annular flow region. Staub and Zuber (1966) and Johnston and Chaddock (1964) also concluded that there was a strong dependence of quality on evaporation heat transfer.

In the Kubanek and Miletti (1979) study, short (~2.68 ft) and long (~8.0 ft) length test sections were compared for average qualities of 0.2 and 0.7. Insignificant differences in the heat transfer coefficient were observed for similar heat fluxes and mass velocities. This observation was not discussed.

R-11 and R-12 Experimental studies from the literature for boiling heat transfer using R-11 and R-12 are listed in Table 2.3. This survey was valuable in designing and understanding various experimental aspects of the test loop. The effects of various parameters used in correlating experimental data are also discussed.

An experimental study by Napadensky (1969) using R-12 resulted in scattered values of heat transfer coefficients when plotted versus quality. A flux number, F1, proved to be an effective parameter in correlating their data, as follows:

\[
\frac{h_{TP}}{h_1} F1^{0.1} = 1.67 q^{0.39} \left( \frac{1 + X}{1 - X} \right)^{1.21} q^{0.018} \tag{2.11}
\]

Chaddock and Noerager (1966) correlated their experimental data using the pure convection mechanism of energy transport concept and the
Table 2.3. Summary of evaporation heat transfer studies of smooth tubes with R-

<table>
<thead>
<tr>
<th>Author</th>
<th>Refrigerant</th>
<th>Type of System</th>
<th>Method of Heating</th>
<th>Orientation</th>
<th>Tube Geometry</th>
<th>Saturation Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bryan and Quaint</td>
<td>R-11</td>
<td>Pump</td>
<td>Electrical</td>
<td>Horizontal</td>
<td>0.314 in. (8.03 mm) I.D., 10 ft (3.07 m) long</td>
<td>3.3-12 (-16 to -11)</td>
</tr>
<tr>
<td>(1951)</td>
<td>R-12</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bryan and Seigel</td>
<td>R-11</td>
<td>Pump</td>
<td>Electrical</td>
<td>Horizontal</td>
<td>0.55 in. (14.07 mm) I.D., 10 ft (3.07 m) long</td>
<td>4-16 (-15 to -9)</td>
</tr>
<tr>
<td>(1955)</td>
<td>R-12</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Chaddock and Noerager</td>
<td>R-12</td>
<td>Compressor</td>
<td>Electrical</td>
<td>Horizontal</td>
<td>0.5 in. (12.80 mm) O.D., 9 ft (2.76 m) long</td>
<td>54 (12)</td>
</tr>
</tbody>
</table>
## R-11 and R-12

<table>
<thead>
<tr>
<th>Mass Velocity (lbm/(hr*ft^2)) x 10^-5</th>
<th>Heat Flux (Btu/(hr*ft^2)) x 10^-3</th>
<th>Inlet-Outlet</th>
<th>Conditions</th>
<th>Data</th>
<th>Correlated</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.016-0.083 (2.16-11.25)</td>
<td>0.054-0.6 (0.17-1.89)</td>
<td>0-25</td>
<td>No</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.002-0.006 (0.27-0.81)</td>
<td>0.10-0.40 (0.32-1.26)</td>
<td>0.3-37</td>
<td>No</td>
<td></td>
<td></td>
</tr>
<tr>
<td>9-43 (1220.40-5830.80)</td>
<td>0.2-11.2 (0.63-35.34)</td>
<td>1-96</td>
<td>Yes</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Author</td>
<td>Refrigerant</td>
<td>Type of System</td>
<td>Method of Heating</td>
<td>Orientation</td>
<td>Tube Geometry</td>
</tr>
<tr>
<td>-----------------</td>
<td>-------------</td>
<td>----------------</td>
<td>-------------------</td>
<td>-------------</td>
<td>---------------</td>
</tr>
<tr>
<td>Napadensky (1969)</td>
<td>R-12</td>
<td>Pump</td>
<td>Electrical</td>
<td>Horizontal</td>
<td>0.90 in. (23.04 mm) I.D., L/D = 110</td>
</tr>
<tr>
<td>Uchida and Yamaguchi (1974)</td>
<td>R-12</td>
<td>---</td>
<td>Electrical</td>
<td>Horizontal</td>
<td>0.25 in. (6.35 mm) I.D., 3.20 ft (0.08 m) long</td>
</tr>
<tr>
<td>Izumi et al. (1975)</td>
<td>R-12</td>
<td>Compressor</td>
<td>Electrical</td>
<td>Horizontal</td>
<td>0.53 in. (13.56 mm) I.D.</td>
</tr>
<tr>
<td>Mass Velocity (lbm/(hr*ft^2))</td>
<td>Heat Flux (Btu/(hr*ft^2))</td>
<td>Conditions</td>
<td>q</td>
<td>Correlated</td>
<td></td>
</tr>
<tr>
<td>-----------------------------</td>
<td>--------------------------</td>
<td>------------</td>
<td>---</td>
<td>------------</td>
<td></td>
</tr>
<tr>
<td>1.98-19.75 x 10^{-5}</td>
<td>3-75</td>
<td>0-27.48</td>
<td>Yes</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(268.49-2678.10)</td>
<td></td>
<td>(0.85-1.51)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.03-3.88 x 10^{-3}</td>
<td>0-100</td>
<td>7.94-103.17</td>
<td>No</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(139.67-526.13)</td>
<td></td>
<td>(25.05-325.50)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.58-14.38 x 10^{-3}</td>
<td>20-70</td>
<td>15.87-79.36</td>
<td>Yes</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(78.65-1949.93)</td>
<td></td>
<td>(50.06-250.38)</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Martinelli parameter, $X_{tt}$, as follows:

$$
\frac{h_{av}}{h_1} = \frac{3.0}{(X_{tt})^{2/3}}
$$

(2.12)

Observations of their data set revealed that both heat flux and mass velocity have a significant effect on heat transfer coefficients. A detailed experimental study of various flow patterns and their relative effects on evaporation heat transfer was conducted by Uchida and Yamaguchi (1974). They concluded that an increase in quality resulted in a decrease in heat transfer coefficient during stratified flow. In contrast, the heat transfer coefficient increased with quality for annular flow. In addition, they observed that the effects of heat flux on the heat transfer coefficient were more decisive in stratified flows than in annular flows. Also, the heat transfer coefficient increased with mass velocity in annular flow.

**Summary** It is obvious from the above discussion that a considerable amount of research work has been conducted in support of understanding the behavior of heat transfer in evaporation of refrigerants. Various empirical correlations, generally having limited applicability, have been suggested. It can be concluded that evaporation heat transfer is a function of heat flux in the nucleate boiling region. In contrast, quality is an important parameter affecting heat transfer in the forced convection boiling regime. A general review of literature conducted by Shah (1982) indicates the correlation of Chen to be a reliable equation for predicting the evaporation heat transfer coefficients even though no refrigerant data were included in the data base.
A parametric study was performed on evaporation heat transfer coefficient using various correlations (Appendix A). A wide discrepancy in the predicted evaporation heat transfer coefficients using different correlations was noted. However, recently developed, statistically fitted correlations, which include the experimental data from most of the above-mentioned studies, are expected to predict the heat transfer coefficients more accurately.

**Correlations for evaporation heat transfer**

**Introduction** Over a period of years, many researchers have developed equations from their own experimental data for calculating evaporation heat transfer coefficients. However, most of these correlations are not reliable beyond the range of data on which they are based. In this section four different correlations, which are based on a wide range of data from several experimental studies, are presented. These correlations were developed by Chen (1966), Dembi et al. (1978), Shah (1982), and Kandlikar (1983). Table 2.4 itemizes the important features of these correlations.

**Chen's correlation** Chen (1966) attempted to correlate the experimental data for evaporation heat transfer coefficients obtained by various researchers using organic fluids. He considered macro and micro convective heat transfer mechanisms for representing saturated boiling heat transfer coefficients. Two non-dimensional functions, namely, a two-phase Reynolds number function, $F$, and a bubble growth suppression function, $S$, were considered for the development of heat transfer equations. The total heat transfer coefficient was then written as
Table 2.4. Summary of correlations for predicting smooth tube evaporation

<table>
<thead>
<tr>
<th>Correlation</th>
<th>Type</th>
<th>Fluid</th>
<th>Saturation Pressure (psi (kPa))</th>
<th>Inlet Velocity Inlet Velocity (ft/s (m/s))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chen (1966)</td>
<td>Local</td>
<td>Water</td>
<td>8-505 (55-3482)</td>
<td>0.2-14.7 (0-4.4)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Methanol</td>
<td>15 (103)</td>
<td>1.0-2.5 (0-4.8)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Cyclohexane</td>
<td>15 (103)</td>
<td>1.3-2.8 (0.4-2)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Pentane</td>
<td>15 (103)</td>
<td>0.9-2.2 (0.3-0.7)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Heptane</td>
<td>15 (103)</td>
<td>1.0-2.4 (0.0-0.7)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Benzene</td>
<td>15 (103)</td>
<td>1.0-2.4 (0.0-0.7)</td>
</tr>
<tr>
<td>Heat Flux (Btu/(hr-ft²))</td>
<td>Inlet (kW/m²)</td>
<td>Outlet (kW/m²)</td>
<td>Conditions</td>
<td>Orientation</td>
</tr>
<tr>
<td>-------------------------</td>
<td>---------------</td>
<td>----------------</td>
<td>-------------</td>
<td>-------------</td>
</tr>
<tr>
<td>y x 10⁻⁴</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.3-8.8 (41.02-277.64)</td>
<td>1-71</td>
<td>Vertical</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.7-1.7 (22.08-53.64)</td>
<td>1-4</td>
<td>Vertical</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.3-1.3 (9.45-41.02)</td>
<td>2-10</td>
<td>Vertical</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.3-1.2 (9.45-37.86)</td>
<td>2-12</td>
<td>Vertical</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.2-0.9 (6.31-28.40)</td>
<td>2-10</td>
<td>Vertical</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.4-1.3 (12.62-41.02)</td>
<td>2-9</td>
<td>Vertical</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Correlation</td>
<td>Type</td>
<td>Fluid</td>
<td>Saturation Temperature °F (°C)</td>
<td>Mass Velocity $g$ $(\text{lbm/(hr}\cdot\text{ft}^2)) \times 10^{-6}$ $(\text{kg/(m}^2\cdot\text{s}))$</td>
</tr>
<tr>
<td>-------------------</td>
<td>-------</td>
<td>-------------</td>
<td>-------------------------------</td>
<td>--------------------------------------------------</td>
</tr>
<tr>
<td>Dembi et al. (1978)</td>
<td>Local</td>
<td>R-11, R-12, R-22</td>
<td>14.2-104.2 (-10.0-40)</td>
<td>-0.011-2.26 (15.0-3100)</td>
</tr>
<tr>
<td>Shah (1982)</td>
<td>Local</td>
<td>Water</td>
<td>216-670 (102-354)</td>
<td>0.2-1.02 (271.20-752.20)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-11</td>
<td>32-103 (0-2)</td>
<td>0.009-0.39 (6.64-287.61)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-12</td>
<td>-103-60 (-75-15)</td>
<td>0.007-0.64 (5.16-471.98)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-22</td>
<td>-4-40 (-20-4)</td>
<td>0.023-0.255 (16.96-188.05)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-113</td>
<td>117-130 (47-54)</td>
<td>0.38-0.51 (280.24-376.11)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Cyclohexane</td>
<td>180 (82)</td>
<td>0.29-0.36 (213.86-265.49)</td>
</tr>
<tr>
<td>Heat Flux ( (\text{Btu/(hr}\cdot\text{ft}^2)) \times 10^{-4} )</td>
<td>Inlet Conditions</td>
<td>Outlet Conditions</td>
<td>Orientation</td>
<td></td>
</tr>
<tr>
<td>-------------------------------------------------</td>
<td>------------------</td>
<td>------------------</td>
<td>-------------</td>
<td></td>
</tr>
<tr>
<td>0.01-22.19 ( (0.35-700) )</td>
<td>0-10</td>
<td>0.3-100</td>
<td>Horizontal</td>
<td></td>
</tr>
<tr>
<td>1.4-25.0 ( (44.17-788.75) )</td>
<td>0-10</td>
<td>0.3-100</td>
<td>Vertical</td>
<td></td>
</tr>
<tr>
<td>0.05-2.2 ( (1.58-69.41) )</td>
<td>0-10</td>
<td>0.3-100</td>
<td>Horizontal</td>
<td></td>
</tr>
<tr>
<td>1.0-64.0 ( (31.55-2019.20) )</td>
<td>0-10</td>
<td>0.3-100</td>
<td>Vertical</td>
<td></td>
</tr>
<tr>
<td>2.3-25.5 ( (72.57-804.53) )</td>
<td>0-10</td>
<td>0.3-100</td>
<td>Horizontal</td>
<td></td>
</tr>
<tr>
<td>38.0-51.0 ( (198.90-1609.05) )</td>
<td>0-10</td>
<td>0.3-100</td>
<td>Horizontal</td>
<td></td>
</tr>
<tr>
<td>29.0-36.0 ( (914.95-1135.80) )</td>
<td>0.65-6</td>
<td></td>
<td>Vertical</td>
<td></td>
</tr>
<tr>
<td>Correlation</td>
<td>Type</td>
<td>Fluid</td>
<td>Mass Velocity (1bm/(hr*ft²)) (B x 10⁻⁶)</td>
<td>Saturation Pressure (psi (kPa))</td>
</tr>
<tr>
<td>---------------</td>
<td>-------</td>
<td>-------</td>
<td>----------------------------------------</td>
<td>---------------------------------</td>
</tr>
<tr>
<td>Kandlikar</td>
<td>Local</td>
<td>Water</td>
<td>0.25-8.26 (340-11200)</td>
<td>0.035-8.26 (340-11200)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-11</td>
<td>0.09-0.18 (12.2-250)</td>
<td>0.09-0.18 (12.2-250)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-12</td>
<td>0.08-0.33 (104-441)</td>
<td>0.08-0.33 (104-441)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-113</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-114</td>
<td>0.44-2.95 (600-4000)</td>
<td>0.44-2.95 (600-4000)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Nitrogen</td>
<td>0.03-0.33 (40-450)</td>
<td>0.03-0.33 (40-450)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Neon</td>
<td>0.06-0.10 (76-131)</td>
<td>0.06-0.10 (76-131)</td>
</tr>
</tbody>
</table>
| Heat Flux  
<table>
<thead>
<tr>
<th>(Btu/(hr*ft^2))</th>
<th>Inlet-Outlet Conditions</th>
<th>Orientation</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\times 10^{-4}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(kW/m^2)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$q$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.26-25.35</td>
<td>--</td>
<td>Horizontal and Vertical</td>
</tr>
<tr>
<td>(40-800)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.04-2.21</td>
<td>--</td>
<td>Horizontal</td>
</tr>
<tr>
<td>(1.3-70)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.06-2.21</td>
<td>--</td>
<td>Vertical</td>
</tr>
<tr>
<td>(2-70)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>--</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.59-19.02</td>
<td>--</td>
<td>Vertical</td>
</tr>
<tr>
<td>(50-600)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.01-12.67</td>
<td>--</td>
<td>Horizontal</td>
</tr>
<tr>
<td>(0.3-400)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.01-1.59</td>
<td>--</td>
<td>Horizontal</td>
</tr>
<tr>
<td>(0.3-50)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
\[ h_{TP} = h_{mic} + h_{mac} \quad (2.13) \]

where

\[ h_{mic} = 0.00122 \frac{K_1^{0.79} \rho_1^{0.45} \rho_l^{0.49} e_c^{0.25}}{\sigma^{0.5} \mu_1^{0.29} \frac{i_{fg}}{\rho_g^{0.24}}} (\Delta T)^{0.24} (\Delta P)^{0.75} S \quad (2.14) \]

and

\[ h_{mac} = 0.023 (Re_l)^{0.8} (Pr_l)^{0.4} (K_1/D) F \quad (2.15) \]

It should be noted that the English system of units (FPI) was used for computing the heat transfer coefficients using Eqs. 2.13 through 2.15.

The Reynolds number factor \( F \) is

\[ F = (Re_{TP}/Re_l)^{0.8} \quad (2.16) \]

and the suppression factor \( S \) is

\[ S = \left( \frac{\Delta T}{\Delta P} \right)^{0.24} \left( \frac{\Delta P}{\Delta T} \right)^{0.75} \quad (2.17) \]

A graphical representation of \( F \) factor correlates the parameter \( 1/X_{tt} \) as a straight line for \( 1.0 < 1/X_{tt} < 100 \). The effective vapor
pressure difference, $\Delta P_e$, corresponds to the effective superheat with the flow ($\Delta T_e$).

The correlation was tested against data for water and organic fluids. A total of 600 data points was within $\pm 12\%$ of the values obtained from the correlation. This correlation also predicts evaporation heat transfer coefficients for refrigerants quite satisfactorily (Dickson and Gouse, 1967; Reidle and Purcupile, 1973).

**Shah's correlation** Shah (1982) developed a correlation based on 800 data points from 18 independent experimental studies for the purposes of designing evaporators. The correlation used data for most common tube materials, horizontal and vertical orientations, circular and annular flow channels, upward and downward flow, as well as a wide range of heat and mass fluxes. He considered the boiling number ($Bo$), the Froude number ($Fr$), and the convection number ($Co$) as major parameters in developing this correlation. The boiling number proved to be important for the nucleate boiling regime. However, at higher mass qualities, the convection number was more applicable. In the intermediate range of qualities, both the boiling number and convection number were applicable. The correlation for horizontal flow is

$$h_{tp} = (\psi) (0.023) \left[ \frac{G(1-X)d}{\mu_1} \right]^{0.8} Pr_1^{0.4} (K_1/D) \quad (2.18)$$

The following procedure was used by the author in evaluating the two-phase multiplier, $\psi$:

$$N = 0.38 Fr_1^{-0.3} Co, \quad Fr_1 \leq 0.04 \quad (2.19)$$
For $N \geq 1.0$

$$\psi_{nb} = 230 \, \text{Bo}^{0.5}, \text{Bo} < 0.3 \times 10^{-4} \quad (2.20)$$

$$\psi_{nb} = 1 + 46 \, \text{Bo}^{0.5}, \text{Bo} < 0.3 \times 10^{-4} \quad (2.21)$$

and

$$\psi_{cb} = \frac{1.8}{N^{0.8}} \quad (2.22)$$

Then $\psi$ is taken as the larger of $\psi_{nb}$ and $\psi_{cb}$.

For $0.1 < N < 1.0$

$$\psi_{bs} = F \, \text{Bo}^{0.5} \exp \left(2.74 \, N^{-0.1}\right) \quad (2.23)$$

and $\psi_{cb}$ is calculated from Eq. (2.22). Again, $\psi$ is the larger of $\psi_{bs}$ and $\psi_{cb}$.

For $N < 0.1$

$$\psi_{bs} = F \, \text{Bo}^{0.5} \exp \left(2.74 \, N^{-0.15}\right) \quad (2.24)$$

and $\psi_{cb}$ is calculated from Eq. (2.22).
The constant is evaluated according to the boiling number as

\[ F = 14.7 \text{ for } Bo \geq 11 \times 10^{-4} \]  \hspace{1cm} (2.25)

and

\[ F = 15.43 \text{ for } Bo < 11 \times 10^{-4} \]  \hspace{1cm} (2.26)

\( \psi \) is the larger of \( \psi_{bs} \) and \( \psi_{cb} \).

The above correlation is actually a modified chart correlation which was developed earlier by Shah (1976). This new correlation (Shah, 1982) is capable of predicting the evaporation heat transfer coefficients in annuli. Various hydraulic diameters for a variety of gaps between the tube and annulus were recommended.

**Dembi et al. correlation**  Dembi et al. (1978) used a statistical analysis to develop a correlation from data by Lavin and Young (1965), Chawla (1967), and Rhee (1972). It should be noted that all these experimental data involved evaporating refrigerants inside tubes. The mass flow rate, vapor mass fraction, fluid properties, and tube diameter were some of the important parameters used in developing this correlation. A Weber number was also selected as one of the parameters since most of the evaporation process occurred with annular flow having a wavy interface. Instead of using a conventional Martinelli parameter \( X_{tt} \), the authors used the factor \( X^4(1 - X^2) \). They also devised a new parameter, \( G^2 \frac{i_{fg}}{g_1 \rho_1} \) which represents the ratio of the energy expended by the fluid in climbing along the wall to the energy
transported by evaporation. The resulting correlation is

$$\frac{hD}{\bar{K}_1} = 0.115 \left( X^4 (1 - X^2) \right)^{0.11} \left( \frac{G^2_{fj} \alpha_j}{g \sigma_1 \rho_1} \right)^{0.44} (Pr)_1^{0.77} \quad (2.27)$$

**Kandlikar's correlation** Recently, Kandlikar (1983) developed a new generalized correlation of data for water, a variety of refrigerants, and cryogenic fluids. This correlation is applicable for both horizontal and vertical orientations. His approach was similar to that of Chen (1966) in that he considered $h = h_{nucl} + h_{conv}$.

A convection number was used for calculating the convective part of heat transfer instead of the Martinelli parameter. A boiling number was used for the nucleate boiling region. Based on the orientation of the tube, the following equations were recommended:

**vertical flow**

$$h_{TP} = D_1(Co)^{D_2} \ h_1 + D_3(Bo)^{D_4} \ h_1 \ F_{fl} \quad (2.28)$$

**horizontal flow**

$$h_{TP} = D_1(Co)^{D_2} \ (25 \ Fr_1)^{0.5} \ h_1 + D_3(Bo)^{D_4} \ (25 \ Fr_1)^{D_6} \ h_1 \ F_{fl}$$

$$\quad (2.29)$$

The constants $D_1$-$D_6$ and $F_{fl}$ were evaluated for each fluid.
Introduction  Condensation heat transfer has been the topic of many analytical and experimental investigations since the pioneering work of Nusselt (1916). In-tube condensation of fluids having low surface tension generally results in film-wise condensation. Specifically, the tube wall is covered with an annular layer of condensate with vapor flowing in the center region. Hence, the condensing vapor has to transfer heat to the tube wall through the condensate film which is normally acted on by gravity, pressure, and interfacial shear forces. This, in turn, means that the condensate film is a major resistance to heat flow.

In this section, several studies related to in-tube condensation of refrigerants inside smooth tube are presented. It should be noted that these discussions are quite brief since detailed literature searches have already been presented by Shah (1978) and Luu (1979). Since annular flow conditions prevailed in most of the experiments performed in this study, more emphasis is put on past studies that relate to annular flow condensation. In addition, some past studies for vertical tubes may be applicable to horizontal tubes at high mass velocities.

Past studies  Recently, Shah (1978) used experimental data from several past investigations to develop a generalized correlation to predict condensation heat transfer. These studies included water, R-11, R-22, R-113, methanol, ethanol, benzene, toluene, and dichloroethylene condensing in horizontal, vertical, and inclined pipes of diameters ranging from 7 to 40 mm. Table 2.5 describes the experimental conditions of the studies using refrigerants R-113, R-22, R-11, and R-12.
Table 2.5. Summary of condensation heat transfer studies of smooth tube with Mas:

<table>
<thead>
<tr>
<th>Author</th>
<th>Refrigerant</th>
<th>Orientation</th>
<th>Tube Size</th>
<th>Saturation Temperature</th>
<th>Mas:</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>I.D., in.</td>
<td>°F (°C)</td>
<td>(lb)</td>
</tr>
<tr>
<td>Altman et al. (1959)</td>
<td>R-22</td>
<td>Horizontal</td>
<td>0.34</td>
<td>93.20-132.80</td>
<td>1.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(8.64)</td>
<td>(33.89-55.89)</td>
<td>(26!)</td>
</tr>
<tr>
<td>Powell (1961)</td>
<td>R-11</td>
<td>Horizontal</td>
<td>0.50</td>
<td>96.80</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(12.70)</td>
<td>(35.89)</td>
<td>(26!)</td>
</tr>
<tr>
<td>Goodykoontz and Dorsch (1967)</td>
<td>R-113</td>
<td>Vertical</td>
<td>0.29</td>
<td>98.6-190.0</td>
<td>5.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(7.37)</td>
<td>(36.89-87.67)</td>
<td>(73!)</td>
</tr>
<tr>
<td>Bae et al. (1969)</td>
<td>R-12</td>
<td>Horizontal</td>
<td>0.50</td>
<td>78.80-118.40</td>
<td>1.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(12.70)</td>
<td>(25.89-47.89)</td>
<td>(185!)</td>
</tr>
<tr>
<td>Bae et al. (1971)</td>
<td>R-22</td>
<td>Horizontal</td>
<td>0.50</td>
<td>78.80-102.20</td>
<td>2.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(12.70)</td>
<td>(25.89-38.89)</td>
<td>(288!)</td>
</tr>
<tr>
<td>Cavallini and Zecchin (1971)</td>
<td>R-11</td>
<td>Vertical</td>
<td>0.79</td>
<td>71.6-98.6</td>
<td>0.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(20.07)</td>
<td>(21.89-35.89)</td>
<td>(73!)</td>
</tr>
<tr>
<td>Traviss et al. (1971)</td>
<td>R-12</td>
<td>Horizontal</td>
<td>0.31</td>
<td>69.80-138.20</td>
<td>0.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(7.87)</td>
<td>(20.89-58.89)</td>
<td>(134!)</td>
</tr>
<tr>
<td>Azer et al. (1972)</td>
<td>R-12</td>
<td>Horizontal</td>
<td>0.50</td>
<td>98.60-122.0</td>
<td>1.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(12.70)</td>
<td>(36.89-49.89)</td>
<td>(185!)</td>
</tr>
<tr>
<td>Traviss et al. (1972)</td>
<td>R-22</td>
<td>Horizontal</td>
<td>0.31</td>
<td>75.20-116.60</td>
<td>1.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(7.87)</td>
<td>(23.89-47.00)</td>
<td>(168!)</td>
</tr>
<tr>
<td>Mass Velocity ((\text{lbm/(hr-ft^2)) \times 10^{-5} ((\text{kg/(m^2-s)})}))</td>
<td>Inlet Conditions</td>
<td>Heat Flux ((\text{Btu/(hr-ft^2)) \times 10^{-3} (\text{kW/m^2}))</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>---</td>
<td>---</td>
<td>---</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(1.99-5.62) ((269.84-762.07))</td>
<td>6-96</td>
<td>---</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(1.68) ((227.81))</td>
<td>25</td>
<td>5.07 ((16.00))</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(5.43-9.57) ((736.30-1297.69))</td>
<td>16-95</td>
<td>11.09-84.91 ((34.99-267.89))</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(1.37-4.93) ((185.77-668.51))</td>
<td>4.94</td>
<td>2.85-19.65 ((8.99-62.00))</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(2.13-4.30) ((288.82-583.08))</td>
<td>17-96</td>
<td>6.02-12.04 ((18.99-37.99))</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(0.54-2.01) ((73.22-272.56))</td>
<td>30-100</td>
<td>2.85-11.09 ((8.99-34.99))</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(0.99-10.01) ((134.24-1357.36))</td>
<td>7-95</td>
<td>1.90-18.07 ((5.99-57.01))</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(1.37-2.92) ((185.77-359.95))</td>
<td>30-99</td>
<td>---</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(1.24-6.56) ((168.14-889.54))</td>
<td>20-95</td>
<td>2.85-26.94 ((8.99-85.00))</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Many investigators have considered single-phase similarity in modeling the condensation process. The basic assumption for these studies is that the liquid film is a major thermal resistance and that the interfacial shear at the vapor-liquid surface can be incorporated in modifications to single-phase correlations for turbulent flow. These modifications were accomplished by using experimentally determined constants or by assuming a particular flow pattern for the condensing fluid. Both approaches result in correlations that have a restricted range of applicability. The studies of Akers et al. (1959), Murthy and Sharma (1972), Cavallini and Zecchin (1974), Izumi et al. (1976), and Shah (1978) were developed using these methods.

Carpenter and Colburn (1951) considered a "lumped model" in which each phase was averaged over the flow cross section and represented by a mean velocity. Using either a theoretical analysis or experimental data, mean velocities were then related to the other physical parameters. Soliman and his co-workers (1968) also derived a correlation for annular flow condensation using this approach.

The studies of Akers and Rosson (1960) and Kutateladze (1961) utilized dimensionless groups to represent physical quantities describing condensing systems. It should be noted that these correlations were restricted to a particular range of flow conditions.

Summary A review of past studies showed that the liquid condensate film constitutes a major thermal resistance during condensation. Also, a number of empirical correlations have been developed for predicting condensation heat transfer coefficients. Recently, Tandon et al. (1984) reviewed several of these correlations and concluded that
many discrepancies exist among them. Details of some of the correlations are described in the next section.

Correlations for condensation heat transfer

Introduction Numerous correlations for determining heat transfer coefficients have been reported in the literature; however, only three correlations were selected for a detailed comparison with the smooth tube results. These three correlations are selected due to their good performance in predicting the smooth tube condensation heat transfer coefficients for refrigerants. These correlations are: Traviss et al. (1972), Cavallini and Zecchin (1974), and Shah (1978). Details of these correlations are reported in Table 2.6. A comparison of the present experimental data for the smooth tube against these three correlations was used to certify the validity of the test apparatus.

Traviss et al. correlation The correlation of Traviss et al. (1972) was selected due to its good performance in predicting heat transfer coefficients for refrigerants. The von Karman velocity distribution in the annular film was used in developing this correlation. The correlation is

\[ h = \frac{K_1}{D} \frac{Pr_e \cdot Re_e^{0.9}}{F_2} F_1 \]  

(2.30)

where,

\[ Re_e = \frac{G(1-X)D}{\mu_1} \]
Table 2.6. Summary of correlations for predicting smooth tube condensation heat transfer

<table>
<thead>
<tr>
<th>Correlation</th>
<th>Type</th>
<th>Model</th>
<th>Mass Velocity (lbm/(hr*ft^2))</th>
<th>Heat Transfer (Btu/ft^2-hr)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Traviss et al.</td>
<td>Local</td>
<td>von Karman analogy</td>
<td>0.17 - 1.20 - 4.60</td>
<td>1.08 - (163 - 624)</td>
</tr>
<tr>
<td>(1972)</td>
<td></td>
<td></td>
<td>0.50</td>
<td>(3.4)</td>
</tr>
<tr>
<td>Cavallini and</td>
<td>Average</td>
<td>Single-phase similarity</td>
<td>--</td>
<td>--</td>
</tr>
<tr>
<td>Zecchin (1974)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Shah (1979)</td>
<td>Local</td>
<td>--</td>
<td>0.021 - 0.16 - 11.30</td>
<td>0.04 - (21.7 - 1532.5)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>0.44</td>
<td>(0.20)</td>
</tr>
</tbody>
</table>
heat transfer coefficients

<table>
<thead>
<tr>
<th>q [10^{-3} \text{kW/m}^2]</th>
<th>Inlet Conditions</th>
<th>Fluids</th>
</tr>
</thead>
<tbody>
<tr>
<td>8-26.99</td>
<td>2-100</td>
<td>R-12</td>
</tr>
<tr>
<td>4-85.17</td>
<td>31-100</td>
<td>R-11, R-21, R-14</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Water, Methanol, Benzene, Toluene, Ethanol, Trichloroethylene</td>
</tr>
<tr>
<td>06-60.0</td>
<td>0-100</td>
<td>R-11, R-12, R-113, R-22.</td>
</tr>
<tr>
<td>0-189.30</td>
<td></td>
<td>Methanol, Benzene, Toluene, Ethanol, Trichloroethylene</td>
</tr>
</tbody>
</table>
Also,

\[ F_1 = 0.15 \frac{1}{X_{tt}} + 2.85 X_{tt}^{-0.476} \]  
\[ F_2 = 0.707 \text{Pr}_e \text{Re}_e^{0.5} \text{ for } \text{Re}_e \leq 50 \]  
\[ F_2 = 5 \text{Pr}_1 + 5 \ln [1 + \text{Pr}_1 (0.09636 \text{Re}_e^{0.585} - 1)] \]  
\[ \text{for } 50 < \text{Re}_e \leq 1125 \]  
\[ F_2 = 5 \text{Pr}_1 + 5 \ln (1 + 5 \text{Pr}_1) + 2.5 \ln (0.00313 \text{Re}_e^{0.812}) \]  
\[ \text{for } \text{Re}_1 > 1125 \]

All properties are evaluated at the saturation temperature.

**Cavallini and Zecchin correlation** The Cavallini and Zecchin (1974) correlation (dimensionless form) is based on data obtained by condensing refrigerants inside a horizontal tube. The analytical results of the authors along with a study by Bae et al. (1969, 1971) confirmed the validity of this correlation. The correlation is

\[ h = 0.05 \frac{K_1}{D} \text{Re}_e^{0.8} \text{Pr}_1^{0.33} \]  
\[ \text{where} \]

\[ \text{Re}_e = \text{Re}_g \left( \frac{\mu}{\rho_1} \right)^{0.5} \left( \frac{\rho_1}{\rho_g} \right)^{0.5} \text{ + Re}_1 \]

All properties are evaluated at the saturation temperature.
Shah's correlation Shah (1978) developed a generalized correlation using experimental data available in the literature which incorporated a wide range of data obtained for condensing refrigerants (Table 2.5). The correlation is

\[
h = 0.023 \frac{K}{D} Re^{0.8} Pr^{0.4} [(1-X)^{0.8} + \frac{3.8 X^{0.76} (1-X)^{0.04}}{Pr^{0.38}}]
\]  

(2.37)

All properties are evaluated at the saturation temperature.

Pressure Drop Studies

Introduction

There are two major factors, namely, friction and momentum, which influence two-phase pressure drops in horizontal tubes. The pressure gradient can be stated as

\[
\frac{dP}{dz} = \frac{dP}{dz}_f + \frac{dP}{dz}_m
\]  

(2.38)

In forced convection two-phase flows, transverse momentum transfer between the vapor core at the center of the tube and the liquid film at the tube wall results in the frictional pressure gradient. Specifically, the interfacial shear stresses due to the vapor flow over the wavy interface, the viscous dissipation from the liquid film, and the formation and subsequent deposition of the droplets contribute to the frictional pressure drop. It should be noted that the interfacial shear stress is a function of tube geometry. Hence, it is expected that
different pressure drops will be obtained with different augmented tubes due to variations in fin geometry.

The momentum part of the pressure drop is due to the velocity change during conversion of vapor to liquid. In general, however, the momentum pressure drop in evaporator and condenser tubes is small when compared with the frictional pressure drops. Therefore, the momentum pressure drop is emphasized less than the frictional pressure drop in this study. This section briefly describes the literature for several pressure drop studies along with correlations for the frictional pressure drop.

Background

This section discusses studies related to frictional pressure drop. An early investigation by Bergelin et al. (1948) compared pressure drops for stratified and annular flow with values for bubbly, wavy, and slug flows. This comparison resulted in relatively stable pressure drops for annular and stratified flows. They concluded that this particular pressure drop characteristic was due to the fact that the liquid layer behaved as a rough pipe wall. These conclusions were later contradicted by Magiros and Dukler (1961). This latter study also found that pressure drop was only a weak function of surface tension and that viscosity effects were a strong function of the liquid flow rate.

A study conducted by Jameson (1971) concluded that the waviness of the liquid-vapor interface is a major factor contributing to pressure drop, especially at high gas velocities. Several other studies available in the literature also support the fact that a wavy motion
enhances the transverse transfer of momentum at the gas-liquid interface; hence, the pressure drop and heat transfer are increased. It should be noted, however, that these studies use simplifying assumptions which has limited their ability in practice to predict experimental results. Even so, most predictions are still made using these two-phase pressure drop correlations. Some of the correlations for predicting two-phase pressure drops are discussed in the following section.

Correlations for smooth tube pressure drop

There are two basic model types utilized in most pressure drop correlations, namely, the homogeneous model and the separated flow model. The homogeneous model is based on the following three assumptions: (1) both liquid and vapor phases have equal velocities, (2) there exists a thermodynamic equilibrium between the two phases, and (3) a single-phase friction factor can be defined for two-phase flow. Using this model, the frictional pressure drop is

\[
- \left( \frac{dP}{dZ} \right)_f = \frac{2 f_{TP} G^2 \mu_l}{D} \left[ 1 + X \left( \frac{\mu_f}{\mu_l} \right) \right]
\]

(2.39)

The two-phase friction factor, \( f_{TP} \), is sometimes replaced by \( f_{LO} \) which involves the liquid viscosity only. Equation (2.39) can be modified to

\[
- \left( \frac{dP}{dZ} \right)_f = - \left( \frac{dP}{dZ} \right)_{LO} \phi^2_{LO}
\]

(2.40)

where \( \phi^2_{LO} \) is known as the two-phase friction multiplier. Details of the
procedure for evaluating frictional pressure drop are given by Collier (1981).

The homogeneous model is simple and easy to use but it is probably suitable for low and high quality regions only. At intermediate qualities, the phases do not move at the same velocity and the homogeneous model may not be applicable. In general, the vapor phase moves faster than the liquid phase. This factor is incorporated in the separated flow model.

In the separated flow model, a two-phase frictional multiplier and the void fraction are related to the independent flow variables using empirical correlations. Since the two phases flow at different velocities, the frictional pressure drop for the total mass flow considered as a liquid, for the liquid phase alone, and for the vapor phase alone can be stated differently as follows:

\[ - \left( \frac{dP}{dz} \right)_f = - \left( \frac{dP}{dz} \right)_{f,\text{LO}} \phi_{\text{LO}}^2 = \left[ \frac{2 f_{\text{LO}} g^2}{\rho_1 D} \right] \phi_{\text{LO}}^2 \]  

(2.41)

\[ - \left( \frac{dP}{dz} \right)_f = - \left( \frac{dP}{dz} \right)_{f,1} \phi_1^2 = \left[ \frac{2 f_1 g^2 (1-x)^2}{\rho_1 D} \right] \phi_1^2 \]  

(2.42)

\[ - \left( \frac{dP}{dz} \right)_f = - \left( \frac{dP}{dz} \right)_{f,g} \phi_g^2 = \left[ \frac{2 f g^2 x^2}{\rho_1 D} \right] \phi_g^2 \]  

(2.43)

Lockhart and Martinelli (1949) developed a generalized parameter \( X \) which is defined as
where $C_1'$, $C_s'$, $m$, and $n$ are defined in Table 2.7. The values of these constants and exponents are dependent on the type of flow (turbulent or viscous) occurring in each phase. The Lockhart and Martinelli correlation was found to be reasonably accurate by Dukler et al. (1964) who tested the correlation against their own experimental data. A similar conclusion was reached by Traviss et al. (1972), Scheideman et al. (1975), and other investigators.

Martinelli and Nelson (1948) later developed a model to predict the two-phase pressure drop during forced circulation. The frictional multiplier for the gas ($\phi_g^2$) and the liquid phases ($\phi_l^2$) were correlated against a parameter $X_{tt}$ using Lockhart and Martinelli (1947) data. The integral value of the frictional multiplier over the length of the tube is required to obtain the total frictional pressure drop.

The Lockhart and Martinelli correlation is used in the present study for comparison with the experimental smooth tube data. Even though more correlations are cited in the literature, their validity for predicting experimental data beyond the range of their experimental parameters is questionable.

Augmented Heat Transfer

During the past 25 years, literature on heat (and mass) transfer enhancement has increased at a rapid rate. Augmentation techniques are classified into two groups: passive and active. No external power is
Table 2.7. Values of exponents m,n and constants $C'_1, C'_g$ for the Lockhart-Martinelli parameter in various flow types

<table>
<thead>
<tr>
<th>Flow Types</th>
<th>$tt$</th>
<th>$vt$</th>
<th>$tv$</th>
<th>$vv$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Re_1$</td>
<td>$&gt;2000$</td>
<td>$&lt;1000$</td>
<td>$&gt;2000$</td>
<td>$&lt;1000$</td>
</tr>
<tr>
<td>$Re_g$</td>
<td>$&gt;2000$</td>
<td>$&gt;2000$</td>
<td>$&lt;1000$</td>
<td>$&lt;1000$</td>
</tr>
<tr>
<td>$m$</td>
<td>0.2</td>
<td>1.0</td>
<td>0.2</td>
<td>1.0</td>
</tr>
<tr>
<td>$n$</td>
<td>0.2</td>
<td>0.2</td>
<td>1.0</td>
<td>1.0</td>
</tr>
<tr>
<td>$C'_1$</td>
<td>0.046</td>
<td>16</td>
<td>0.046</td>
<td>16</td>
</tr>
<tr>
<td>$C'_g$</td>
<td>0.046</td>
<td>0.046</td>
<td>16</td>
<td>16</td>
</tr>
</tbody>
</table>
required for the operation of passive augmentation techniques while active techniques do require external power.

Passive techniques consist of treated surfaces, rough surfaces, extended surfaces, displaced enhancement devices, swirl flow devices, coiled tubes, surface tension devices, and additives. Active techniques include mechanical aids, surface vibration, fluid vibration, electrostatic fields, injection, and suction. Advantages of both techniques for various modes of heat transfer are discussed in detail by Bergles et al. (1983).

With rapid developments in manufacturing techniques, a wide variety of tubes with internal fins have been developed and tested for heat transfer and pressure drop performance. For refrigeration and air conditioning applications, considerable interest has recently been shown in tubes with many small spiral fins. Several studies in the open literature have demonstrated good heat transfer and pressure drop performance for these "micro-fin" tubes. As mentioned in Chapter I, the micro-fin tubes have numerous smaller size fins. The area increases for micro-fin tubes are typically in the range of 1.1 to 1.6 times those of comparable smooth tubes. Two-phase heat transfer coefficients are about 1.5 to 2.5 times the smooth tube value (enhancement factors of 1.5 to 2.5) while the pressure drop increases are 1.2 to 2.0 times those for smooth tubes. It should be noted that higher heat transfer enhancement factors are obtained for condensation compared to evaporation. However, in general, the difference in the pressure drop enhancement factors for these two modes of heat transfer is marginal.
A detailed review of literature related to internally finned tubes for single-phase and two-phase heat transfer is reported in this section. Due to the large amount of research work performed in this particular area of heat transfer augmentation, studies related only to refrigerants R-113, R-22, R-11, and R-12 are reported. A detailed discussion of the limited data for micro-finned tubes is included in the following section.

**Single-phase**

Spiral fin tubes exhibiting 200% augmentation of single-phase heat transfer have been reported (Bergles et al., 1983). In general, secondary flows are caused by the spirals, thus, resulting in greater heat transfer than for axial fins. The present review covers literature which is relevant only to the understanding of heat transfer and fluid flow in spirally finned tubes. It should be noted that no specific study on augmentation of single-phase heat transfer using micro-fin tubes was found. Hence, some single-phase studies for tubes with higher fins relative to the micro-finned tubes are briefly discussed.

Bergles et al. (1970) tested seven different finned tubes, some of which had spiraled fins. They observed that stagnation occurred in spirally finned tubes at low mass velocities even though spiraled fins were expected to promote turbulence. The authors concluded that short fins with spirals result in less stagnation of the fluid. Watkinson and his co-workers (1972) concluded that a spirally finned tube with low pitch-to-diameter ratio or fewer fins at a given pitch-to-diameter ratio is best for single-phase heat transfer.
Recently, Carnavos (1980) developed empirical correlations from his experimental data using 21 internally finned tubes with air. Experiments for studying the effects of spiral angle and Prandtl number were also conducted on 11 tubes using water and a mixture of water and ethylene glycol. The empirical correlation developed by Carnavos included inside diameter of the tube, pitch, fin height, helix angle, and fin thickness as variables.

Augmented Evaporation

Introduction

In this section, enhancement of evaporation heat transfer using finned tubes is discussed for the refrigerants that are relevant to the present study. Details of these experimental studies along with important parameters are listed in Table 2.8. Studies that relate to the enhancement of heat transfer and pressure drop using micro-finned tubes are described in even greater detail in Table 2.9.

Internally finned tubes

A comparative study between spirally finned tubes and star-shaped inserts using short (~ 0.80 m) and long (~ 2.40 m) length test sections for R-22 was conducted by Kubanek and Miletti (1979). In general, spirally finned tubes gave better performance than star-shaped inserts. An interesting observation was that heat transfer coefficients decreased with increases in the mass flow rate using star inserts. In contrast, heat transfer coefficients increased with mass flow rates for spirally finned tubes. The experiments of Kubanek and Miletti also indicated an
Table 2.8. Summary of evaporation studies with internally finned tubes

<table>
<thead>
<tr>
<th>Author</th>
<th>Refrigerant</th>
<th>Type of System</th>
<th>Method of Heating</th>
<th>Orientation</th>
<th>Tube Geometry</th>
<th>Saturation Temperature °F (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boling et al.</td>
<td>R-12</td>
<td>Compressor</td>
<td>Water</td>
<td>Horizontal</td>
<td>Axially finned tube</td>
<td>42-54 (5-12)</td>
</tr>
<tr>
<td>(1953)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.5 in. (12.7 mm)</td>
<td>I.D.</td>
</tr>
<tr>
<td>Schlünder and Chawla</td>
<td>R-11</td>
<td>—</td>
<td>—</td>
<td>Horizontal</td>
<td>Axially finned tube</td>
<td>64 (18)</td>
</tr>
<tr>
<td>(1967)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.98 in. (24.89 mm)</td>
<td>I.D.</td>
</tr>
<tr>
<td>Tojo et al.</td>
<td>R-22</td>
<td>Compressor</td>
<td>Water</td>
<td>Horizontal</td>
<td>Micro-finned tube</td>
<td>14-41 (-10-5)</td>
</tr>
<tr>
<td>(1977)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.375 in. (9.52 mm)</td>
<td>O.D.</td>
</tr>
</tbody>
</table>

*gal/hr.

**Btu/hr.
<table>
<thead>
<tr>
<th>Mass Velocity (lbm/(hr*ft²)) x 10⁻⁵</th>
<th>Heat Flux (Btu/(hr*ft²)) x 10⁻³</th>
<th>Inlet ((kg/(m²*s))</th>
<th>Outlet (kW/m²)</th>
<th>G Conditions</th>
<th>q</th>
<th>Correlated</th>
</tr>
</thead>
<tbody>
<tr>
<td>-54</td>
<td>67.5-*</td>
<td>0-100</td>
<td>2450-**</td>
<td>No</td>
<td></td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>243</td>
<td></td>
<td>7200</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>54</td>
<td>0.57-</td>
<td>0-70</td>
<td>0.921-</td>
<td>Yes</td>
<td></td>
<td></td>
</tr>
<tr>
<td>18</td>
<td>7.28</td>
<td></td>
<td>11.06</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>(77.29-</td>
<td></td>
<td>(2.91-</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>987.17)</td>
<td></td>
<td>34.89)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>-41</td>
<td>0.86-2.87</td>
<td>0-100</td>
<td>---</td>
<td>No</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0-5</td>
<td>(116.62-</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>389.17)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
### Table 2.8. concluded

<table>
<thead>
<tr>
<th>Author</th>
<th>Refrigerant</th>
<th>Type of System</th>
<th>Method of Heating</th>
<th>Orientation</th>
<th>Saturation Temperature °F (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ito and Kimura</td>
<td>R-22</td>
<td>Compressor</td>
<td>--</td>
<td>Horizontal</td>
<td>Spirally finned tube (5) 40</td>
</tr>
<tr>
<td>(1979)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.45 in. (11.43 mm) I.D.</td>
</tr>
<tr>
<td>Kubanek and</td>
<td>R-22</td>
<td>Compressor</td>
<td>Water</td>
<td>Horizontal</td>
<td>Spirally finned tube (7-27) 44-80</td>
</tr>
<tr>
<td>Milette (1979)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.45 in. (11.43 mm) I.D.,</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>2.67-8.13 ft (0.81-2.50 mm) long</td>
</tr>
<tr>
<td>Kimura and Ito</td>
<td>R-12</td>
<td>Compressor</td>
<td>Direct heating</td>
<td>Horizontal</td>
<td>Micro-finned tube (1) 30</td>
</tr>
<tr>
<td>(1981)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.18 in. (4.57 mm) I.D.</td>
</tr>
<tr>
<td>Mass Velocity (lbm/(hr*ft^2))</td>
<td>Heat Flux (Btu/(hr*ft^2))</td>
<td>Inlet-Outlet Conditions</td>
<td>Data</td>
<td></td>
<td></td>
</tr>
<tr>
<td>------------------------------</td>
<td>--------------------------</td>
<td>-------------------------</td>
<td>------</td>
<td></td>
<td></td>
</tr>
<tr>
<td>-5 * 10^{-2}</td>
<td>-3 * 10^{-3}</td>
<td>G</td>
<td>q</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2-5</td>
<td>30-70</td>
<td></td>
<td>No</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(271.5-678.0)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>-80</td>
<td>20-70</td>
<td>37.92</td>
<td>No</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(57.80-119.57)</td>
<td>(2.50)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>-27</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>30-100</td>
<td>0.792</td>
<td></td>
<td>No</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(25.76-142.38)</td>
<td>(2.50)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Author</td>
<td>Tube Material</td>
<td>Tests</td>
<td>Diameter (D) in. (mm)</td>
<td>Thickness (l) in. (mm)</td>
<td>Pitch (P) in. (mm)</td>
</tr>
<tr>
<td>------------------</td>
<td>---------------</td>
<td>-------</td>
<td>-------------------------</td>
<td>--------------------------</td>
<td>----------------------</td>
</tr>
<tr>
<td>Ito et al. (1977)</td>
<td>—</td>
<td>R-22</td>
<td>—</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>Ito and Kimura</td>
<td>Copper</td>
<td>R-22</td>
<td>0.44 (0.11^{\text{.2}})</td>
<td>—</td>
<td>0.020-0.068 (0.5-1.75)</td>
</tr>
<tr>
<td>(1979)</td>
<td>Aluminum</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Kimura and Ito</td>
<td>Copper</td>
<td>R-12</td>
<td>0.18 (0.47^{\text{.5}})</td>
<td>—</td>
<td>0.020 (0.50^{\text{.0}})</td>
</tr>
<tr>
<td>(1981)</td>
<td>I.D.</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tatsumi et al.</td>
<td>Copper</td>
<td>R-22</td>
<td>0.375 in. (9.52) (0.41)</td>
<td>0.016 (0.41) (0.52)</td>
<td>—</td>
</tr>
<tr>
<td>(1982)</td>
<td>O.D.</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tojo et al.</td>
<td>Copper</td>
<td>R-22</td>
<td>0.375 (9.52) (0.35)</td>
<td>0.014 (0.35) (0.52)</td>
<td>—</td>
</tr>
<tr>
<td>(1984)</td>
<td>O.D.</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Fin tubes for evaporation

<table>
<thead>
<tr>
<th>Fin Height in. (mm)</th>
<th>Spiral Angle</th>
<th>Increase in Surface Area Referenced to Smooth Tube $A_{aug}/A_{smooth}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.020 (0-0.5)</td>
<td>0-90°</td>
<td>—</td>
</tr>
<tr>
<td>0.0068-0.016 (0.16-0.40)</td>
<td>0-75°</td>
<td>1.07-1.65</td>
</tr>
<tr>
<td>0.004 (0.10)</td>
<td>4°-30°</td>
<td>1.15</td>
</tr>
<tr>
<td>0.0047-0.006 (0.12-0.15)</td>
<td>7.5°</td>
<td>—</td>
</tr>
<tr>
<td>0.006-0.0079 (0.15-0.20)</td>
<td>10°-25°</td>
<td>—</td>
</tr>
</tbody>
</table>
increase in the heat transfer coefficient with a decrease in fin pitch. This means that a threshold fin pitch exists below which spiral fins are more effective than straight fins. An analysis of the experimental data for the spirally finned tubes showed that enhancement factors differed for the short and long test sections except at low mass velocities where they were quite comparable. Specifically, at the higher mass velocities much higher enhancement factors were observed for long test sections compared to short test sections. No specific reasons for this particular behavior of enhancement factors were given.

Micro-finned tubes

Ito et al. (1977) reported enhancement factors of 1.5 to 2.0 using triangular shaped fins (both peak and valley). Optimum spiral angles of 7° and 90° were recommended. They also concluded that fin heights of 0.008 in. (0.2 mm) or less result in an insignificant increase in pressure drop. A study by Ito and Kimura (1979) using R-22 suggested optimum fin pitches of 0.019 in. (0.5 mm) and 0.039 in. (1.0 mm) for 0.44 in. I.D. (11.2 mm) triangular-shaped micro-fin tubes. A fin height of 0.008 in. (0.2 mm) was recommended for evaporation heat transfer. Their study indicated that the heat transfer could be further improved by increasing the height of the fin, but at the cost of higher pressure drop. Also, the spiral angle was observed to be a key parameter with minimum heat transfer occurring at 45°. Kimura and Ito (1981) reported enhancement factors of 1.5 to 2.0 in annular flow (based on smooth tube data) with 0.188 in. (4.75 mm) O.D. triangular shaped micro-fin tubes.
An optimum spiral angle of 15° was recommended for R-12 evaporation heat transfer data.

Tatsumi et al. (1982) reported enhancement factors of 1.6 to 2.0 with negligible penalties on pressure drop using triangular and trapezoidal shaped fins. They could not rigorously investigate the effects of the fin shape because additional parameters were varied. Hence, a systematic performance study of their tubes was not carried out. A study of three different configurations of fin tip and valley geometry was performed by Tojo and co-workers (1984). The fin geometries were round tip and round valley, sharp tip and flat valley, and finally, flat tip and flat valley. They tested tubes with different fin heights, spiral angles, and number of fins. These micro-fin tubes resulted in enhancement factors of 1.6 to 2.0 in heat transfer and about 1.2 to 14 in pressure drop. Based on their results, a spiral angle of 10 degrees and a fin height of 0.008 in. (0.20 mm) was recommended.

Augmented Condensation

Introduction

In this section, the enhancement of condensation heat transfer using internally finned tubes is described. Details of several experimental studies are reported in Table 2.10. In addition, studies for micro-finned tubes, which are of special interest to the present investigation, are outlined in even greater detail in Table 2.11.
<table>
<thead>
<tr>
<th>Author</th>
<th>Refrigerant</th>
<th>Type of System</th>
<th>Method of Heating</th>
<th>Orientation</th>
<th>Tube Geometry</th>
<th>Saturation Temperature °F (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pearson and Weathers</td>
<td>R-22</td>
<td>Compressor</td>
<td>Water</td>
<td>Horizontal</td>
<td>Internally finned</td>
<td>105°F (40°C)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>tubes 3/4 in. (19.05 mm) O.D.</td>
<td></td>
</tr>
<tr>
<td>Ivanov et al. (1972)</td>
<td>R-12</td>
<td>Compressor</td>
<td>Water</td>
<td>Horizontal</td>
<td>Smooth finned tube</td>
<td>86-122°F (30-50°C)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.63 in. (16.00 mm) O.D.</td>
<td></td>
</tr>
<tr>
<td>Reisbig (1974)</td>
<td>R-12</td>
<td>Compressor</td>
<td>Water</td>
<td>Horizontal</td>
<td>Splined tube 3/8 in. (9.52 mm) O.D., 50.5 in. (1.28 m) long</td>
<td>5-70°F (-15-21°C)</td>
</tr>
</tbody>
</table>
Mass Velocity (lbm/(hr*ft^2)) | Heat Flux (Btu/(hr*ft^2)) | G Conditions | q | Correlated
---|---|---|---|---
| Inlet, x 10^-5 | Outlet x 10^-3 |

<table>
<thead>
<tr>
<th>Temperature (°C)</th>
<th>(kg/(m^2*s))</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>05</td>
<td>--</td>
<td>Super-heated vapor - 4-8°F (2-4°C) sub-cooled liquid</td>
<td>--</td>
</tr>
<tr>
<td>0)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>122</td>
<td>0.58-1.05</td>
<td>--</td>
<td>--</td>
</tr>
<tr>
<td>50</td>
<td>(78.65-)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>142.38</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>70</td>
<td>2-7.5</td>
<td>Saturated liquid, saturated vapor</td>
<td>0.02-2.0</td>
</tr>
<tr>
<td>15</td>
<td>(271.2-)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>(1017.0)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

---

**Note:** The table represents data for two different conditions: superheated vapor and sub-cooled liquid.
Table 2.10. continued

<table>
<thead>
<tr>
<th>Author</th>
<th>Refrigerant</th>
<th>Type of System</th>
<th>Method of Heating</th>
<th>Orientation</th>
<th>Tube Geometry</th>
<th>Saturation Temperature °F (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vrable et al. (1974)</td>
<td>R-12</td>
<td>Pump</td>
<td>Water</td>
<td>Horizontal</td>
<td>Axially finned tube</td>
<td>41-77 (5-25)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.43 in (10.92 mm) I.D., 22.31 ft (6.80 m) long</td>
<td></td>
</tr>
<tr>
<td>Luu and Bergles (1979, 1980)</td>
<td>R-113</td>
<td>Pump</td>
<td>Water</td>
<td>Horizontal</td>
<td>Internally-finned, rough tubes</td>
<td>170-245 (77-118)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.33 in (33.78 mm) I.D., 12 ft (3.65 m) long</td>
<td></td>
</tr>
<tr>
<td>Temperature (°C)</td>
<td>Mass Velocity (lbm/(hr*ft²)) x 10⁻⁵</td>
<td>Heat Flux (Btu/(hr*ft²)) x 10⁻³</td>
<td>Heat Transfer Conditions</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>------------------</td>
<td>-----------------------------------</td>
<td>---------------------------------</td>
<td>-------------------------</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>-77 -25</td>
<td>Inlet,</td>
<td>Outlet</td>
<td></td>
<td>Data (kW/m²) Correlated (°F or °C)</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>G Conditions</td>
<td>q</td>
<td>Yes</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>-77 -25</td>
<td>9-45°F</td>
<td></td>
<td>11-88</td>
<td>Yes</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Superheated vapor, subcooled liquid</td>
<td></td>
<td>34.71-277.64</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>245 18</td>
<td>0.63-5.6</td>
<td>Saturation</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>245 18</td>
<td>(85.42-759.36)</td>
<td></td>
<td>(34.71-759.36)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Superheated vapor, subcooled liquid</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Author</td>
<td>Refrigerant</td>
<td>Type of System</td>
<td>Method of Heating</td>
<td>Orientation</td>
<td>Tube Geometry</td>
<td>Saturation Temperature °F (°C)</td>
</tr>
<tr>
<td>----------------------</td>
<td>-------------</td>
<td>----------------</td>
<td>-------------------</td>
<td>-------------</td>
<td>---------------</td>
<td>-------------------------------</td>
</tr>
<tr>
<td>Said and Azer (1983)</td>
<td>R-113</td>
<td>Pump</td>
<td>Water</td>
<td>Horizontal</td>
<td></td>
<td>135-185 (57-84)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Straight finned, spirally finned</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.38 in. (3.5 cm)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>I.D., 8 ft (2.44 m)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>long</td>
<td></td>
</tr>
<tr>
<td>Tojo et al. (1984)</td>
<td>R-12</td>
<td>Compressor</td>
<td>Water</td>
<td>Horizontal</td>
<td></td>
<td>--</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Spirally finned tubes</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.375 in. (9.52 mm)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>O.D., 16.4 ft (5.00 m)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>long</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Spirally finned tube</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.54 in. (13.72 mm)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>I.D.</td>
<td></td>
</tr>
<tr>
<td>Inlet Temperature (°C)</td>
<td>Mass Velocity (lbm/(hr*ft^2)) x 10^-5</td>
<td>Heat Flux (Btu/(hr*ft^2)) x 10^-3</td>
<td>Outlet Conditions</td>
<td>q (kW/m^2)</td>
<td>Data Correlated</td>
<td></td>
</tr>
<tr>
<td>------------------------</td>
<td>--------------------------------------</td>
<td>---------------------------------</td>
<td>-------------------</td>
<td>------------</td>
<td>----------------</td>
<td></td>
</tr>
<tr>
<td>185</td>
<td>0.1-2.25 (13.56-305.10)</td>
<td>2.8-23.75</td>
<td>Super-heated vapor, subcooled liquid</td>
<td>8.83-74.93</td>
<td>Yes</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>20-70 saturated vapor, subcooled liquid</td>
<td>--</td>
<td>No</td>
<td></td>
</tr>
<tr>
<td>-25</td>
<td>0.12-0.63 (16.27-85.43)</td>
<td>183.9-608.7</td>
<td>Super-heated vapor, subcooled liquid</td>
<td>580.21-1920.45</td>
<td>Yes</td>
<td></td>
</tr>
</tbody>
</table>
Table 2.11. Summary of the geometrical characteristics of micro-fin tubes used for condensation

<table>
<thead>
<tr>
<th>Author</th>
<th>Tube Material</th>
<th>Fluid</th>
<th>Outside Diameter in. (mm)</th>
<th>Thickness in. (mm)</th>
<th>Fin Height in. (mm)</th>
<th>Spiral Angle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tatsumi et al.</td>
<td>Copper</td>
<td>R-22</td>
<td>0.375 (9.52)</td>
<td>0.016 (0.40)</td>
<td>0.0047-0.006</td>
<td>7.5°-25°</td>
</tr>
<tr>
<td>(1982)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tojo et al.</td>
<td>Copper</td>
<td>R-22</td>
<td>0.375 (9.52)</td>
<td>0.014 (0.36)</td>
<td>0.006-0.0079</td>
<td>10°-25°</td>
</tr>
<tr>
<td>(1984)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
**Internally finned tubes**

Luu and Bergles (1979) investigated the effects of twisted tapes, internally finned tubes, and rough tubes using R-113. Use of internally finned tubes gave considerably better performance than the twisted tapes, both in heat transfer and pressure drop. Said and Azer (1983) carried out a very similar study at lower mass velocities and low working pressures. They correlated their data for both internally finned tubes and twisted tapes. The results of Said and Azer confirmed the data of Luu and Bergles (1979, 1980).

Recently, Venkatesh and Azer (1985) tested five augmented tubes for condensation heat transfer using R-12. Three out of the five tubes were spirally finned tubes. Over the mass flow range tested, heat transfer enhancements as high as 1.55 were obtained.

**Micro-finned tubes**

Tatsumi et al. (1982) conducted two-phase condensation heat transfer and pressure drop studies using triangular and trapezoidal shape fins. Even though geometrical parameters such as fin height and the number of fins were varied, the effects of fin shape (peak and valley) were not studied. These investigators recommended a fin height of 0.007-0.008 in. (0.18-0.20 mm). They also observed that increases in the spiral angle resulted in increased heat transfer. Specifically, a sharp increase in the heat transfer coefficient was observed as the spiral angle was increased from $0^\circ$ to $10^\circ$. However, the increase in the heat transfer was moderate for further increases in the spiral angle.
from $10^\circ$ to $25^\circ$. No specific values of the spiral angles were recom-
mended by the authors for optimal condensation performance.

Tojo and co-workers (1984) conducted an experimental investigation
using fins having three different peak and valley geometries. Specifi-
cally, fins with a sharp tip and flat valley, a round tip and round
valley, and, finally, a flat tip and flat valley were considered. Heat
transfer enhancement factors of 1.8 to 2.0 were reported; however, the
pressure drop effects were not reported. The tube with a flat tip and a
flat valley resulted in about a 30% higher heat transfer coefficient
than the tube having a sharp tip and flat valley. The effects of round
tip and round valley were not effectively studied because additional
geometrical parameters were not varied.
CHAPTER III. AUGMENTATION OF R-113 IN-TUBE EVAPORATION AND CONDENSATION WITH MICRO-FIN TUBES (WATER HEATED/COOLED SHORT TEST SECTION)

Introduction

R-113 was used to screen tube geometries prior to performing experiments using R-22. Since properties of both refrigerants are quite similar, heat transfer enhancements should be similar for both fluids. Refrigerant R-113 was used because its higher saturation temperature at atmospheric pressure greatly simplified the test rig construction and shortened the time required to mount a tube in the rig.

The experimental program was divided into two phases. In the first phase, a smooth tube and nine augmented tubes were tested for overall heat transfer and pressure drop for both evaporation and condensation using 41 in. (1.0 m) long test sections. In the second phase (Chapter IV), a smooth tube and an augmented tube (Tube 10) were tested for local heat transfer and pressure drop using a 12 ft (3.65 m) long test section with evaporation. Details of the first phase are presented in this chapter while the second phase is presented in Chapter IV.

In this study, the effects of several geometrical parameters of micro-finned tubes were experimentally investigated for evaporation and condensation heat transfer. Specifically, the geometrical parameters studied were peak shape, valley shape, fin height, number of fins, and spiral angle. In addition, single-phase heat transfer data are also presented herein. These data are of interest in formulating
correlations since two-phase heat transfer is commonly referenced to single-phase heat transfer.

Experimental Facility

General

The test apparatus consisted of short test sections that were 41 in. (1.0 m) long. The use of short test sections reduced installation times. Additionally, since low pressure R-113 was used as a test fluid, it was possible to test and study a wide range of augmented tubes. A summary of the experimental operating conditions is presented in Table 3.1.

The experimental set-up consisted of two major flow loops: a refrigerant flow loop containing the test section and a water flow loop for heating or cooling of the refrigerant in the test section. The refrigerant was heated during evaporation and cooled during condensation and single-phase tests. Figure 3.1 is a schematic drawing of the complete test apparatus. A photographic view of the test rig is shown in Fig. 3.2.

Refrigerant loop

The R-113 loop consisted of a positive displacement pump, a preheater, a test-section, a drier and filter unit, a degassing tank, and an after-condenser. Details of each of these components are reported in Appendix B. The test fluid was pumped around the loop using a gear pump. After leaving the pump, the fluid was divided into a bypass stream, which passed through a dryer and filter unit, and a
Table 3.1. Operating parameter range for water-jacketed evaporation/condensation test facility (R-113 as a refrigerant)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass velocity</td>
<td>145,000 - 440,000 lbm/(hr<em>ft²) (197 - 594 kg/(m²</em>s))</td>
</tr>
<tr>
<td>Heat flux supplied to test section</td>
<td>3,400 - 17,000 Btu/(hr*ft²) (10.72 - 53.65 kW/m²)</td>
</tr>
<tr>
<td>Test fluid inlet pressure</td>
<td>45 - 51 psia (312 - 351 kPa)</td>
</tr>
<tr>
<td>Average quality in the test section</td>
<td>0.15 - 0.85</td>
</tr>
<tr>
<td>Quality change over the test section</td>
<td>0.2 - 0.3</td>
</tr>
</tbody>
</table>
Figure 3.1. Schematic diagram of flow loop
Figure 3.2. A photographic view of the test apparatus
main stream, which continued on to the preheater/preevaporator. The preheater consisted of a 1/2 in. (12.7 mm) O.D., 8.64 ft (2.63 m) long, stainless steel 304 tube which was heated using direct electrical current. The amount of electrical energy supplied to the preheater determined the vapor quality of the refrigerant entering the test section. After passing through the test section, the refrigerant was condensed and cooled in the after-condenser using water from the building supply.

Each end of the preheater was electrically isolated from the rest of the rig using 3 in. (76.2 mm) long rubber hoses. Similarly, sight glasses were mounted at each end of the test section for observing inlet and outlet conditions including flow patterns. The inlet conditions to the test section were of special importance during evaporation heat transfer experiments. Hence, a considerable effort was made to maintain a smooth flow stream through these components from the preheater exit to the test section inlet.

The test section was essentially a counter-flow, concentric-tube heat exchanger with refrigerant flowing through the inner tube and the heating water flowing in the annulus. The evaporation or condensation was carried out at a refrigerant pressure of approximately 35 psi (241 kPa) which corresponds to an R-113 saturation temperature of 169.6°F (76.4°C). After leaving the test section, the test fluid was condensed and then subcooled in the after-condenser to a temperature of approximately 70°F (21°C). The degassing tank was bypassed during normal operation; however, the tank served as a receiver to accommodate the expansion or contraction of fluid in the loop.
Test section

The test section consisted of a test tube surrounded by an annulus section so as to form a shell-and-tube counter flow heat exchanger. The copper test tubes were 3/8 in. (9.5 mm) O.D. with a typical wall thickness of 0.014 in. (0.36 mm) and a length of 48 in. (122 cm). Due to the sensitivity of the heat transfer coefficient to the saturation temperature, accurate fluid pressure measurements were necessary in the test section. Hence, two calibrated pressure gages were connected in parallel for this purpose. The pressure drop over the length of the test section was measured using a mercury manometer with a resolution of 0.025 in. (0.64 mm) of Hg. Pressure taps were formed from 1/8 in. (3.17 mm) compression fittings split longitudinally into two halves and then reduced to a height of 1/8 in. (3.17 mm). These fittings were soldered 41 in. (1.0 m) apart onto the test tube and then connected to the differential manometer using 1/8 in. (3.17 mm) O.D. stainless steel tubes. Two custom-made teflon spacers were placed 41 in. (1.0 m) apart at each end of the test section. Thus, restricting the annulus side water flow to 41 in. (1.0 m) length of the test section.

Wall temperature measurements were made with seven 30 gage copper-constantan thermocouples which were soldered onto the surface of the tube. Figure 3.3 indicates their locations on the tube surface. Each thermocouple bead was covered by a small layer of epoxy in the region where it was soldered on the tube wall. Adding this epoxy, along with attaching the thermocouple to the wall of the tube along its entire length, reduced errors in the wall temperature measurements. Thermocouples were passed through the outer annulus wall using a special
Figure 3.3. Details of the test section
pressure fitting. The refrigerant bulk temperature was measured at each end of the test section using thermocouples inserted in the fluid. The water-side bulk temperatures were measured using a pair of 36 gage copper-constantan thermocouples mounted in a Tee-section soldered at each end of the test section.

**Water loop**

The water flow loop consisted of a centrifugal pump, a filter, a flow meter, a resistance heater, a heat exchanger, an accumulator, and, finally, the test section described above. Prior to entering the test section, the water passed through a filter unit and then preheated with a resistance heater. A rotameter was used to measure the water flow rate. An accumulator was installed at the discharge side of the pump in order to maintain the pressure in the system and to dampen the flow fluctuations.

An average water temperature of $210^\circ F$ ($99^\circ C$) was required during the evaporation heat transfer tests. Since this temperature is very close to the boiling point of water at atmospheric pressure, the flow loop was pressurized up to 50 psia (344.7 kPa). Since the inlet and outlet bulk temperature difference was only 3-5$^\circ F$ (about 2$^\circ C$) for most of the experiments, the two thermocouples at each end of the tube wall were monitored continuously for a check on the bulk temperature measurements.

A calibrated rotameter was used to measure the test fluid flow rate. Eighteen 30 gage copper-constantan thermocouples measured
temperatures throughout the refrigerant loop. A computerized data acquisition system was used for data taking and data reduction.

Experimental Procedure

Single-phase

Prior to performing experiments using the above apparatus, the R-113 was degassed thoroughly. Details of the degassing process are reported in Appendix B.

After completing the degassing process, the system pressure of approximately 35 psi (241 kPa) was set using the pump bypass valve and flow regulating valve. The desired test fluid flow rate was then set using the flow regulating valve. The desired test fluid bulk temperature drop of approximately 30°F (17°C) across the test section was attained using the preheater. The after-condenser cooling water flow rate was then set to attain a preheater inlet temperature of approximately 70°F (21°C). At a steady/quasi-steady state, the data acquisition system was initiated and data were reduced.

Single-phase experiments checked out the heat balance between the shell and tube side of the test section. In general, a heat balance within ± 3% between the shell side and tube side of the test section was observed. Typically, for each tube under investigation, heat transfer coefficients were obtained at five different Reynolds numbers. The details of the data reduction procedure are described later.
After completing the degassing process, system pressure and flow rates were established by controlling the test fluid throttle valve, pump bypass valve, and flow regulating valve. Typically, the pressure in the test section was set at about 35 psi (241 kPa) while the pump discharge pressure was set at 70 psi (482 kPa). The pressure drop of approximately 35 psi (241 kPa) across the throttle valve prevented thermal-hydraulic instabilities in the system. The next step was to set the inlet quality of the refrigerant entering the test section by controlling the dc power connected to the preheater. The temperature and flow rate in the water loop were then set to attain a quality change of 0.2 to 0.3 over the length of the test section. The cooling water to the after-condenser was then adjusted so that the inlet test fluid temperature was reduced to 70°F (21°C) before entering the pump.

After the system had reached steady or quasi-steady state, as determined by repeated checks of temperature, pressure, and flow rates, the data acquisition system was initiated. Data monitoring, acquisition, reduction, and plotting were performed with a Hewlett-Packard data acquisition system. This system included a Hewlett-Packard Model 9825A computer, a Hewlett-Packard Model 3421A scanner, a Hewlett-Packard Model 3456A voltmeter, a Kaye Instruments Model K170-36C ice-point reference, and a Hewlett-Packard Model 8971A printer.

Data Reduction

Convective heat transfer coefficients were calculated from raw data using a data reduction computer program. Input to this program was
either automatic using the computerized data acquisition system or manual using hand-recorded data. For example temperature readings were recorded automatically while the absolute pressure, the pressure drop, and flow rates were read manually. The experimental data are recorded in the Heat Transfer Laboratory of the Department of Mechanical Engineering at Iowa State University.

**Single-phase heat transfer**

The properties for both fluids in the test section were calculated using averages of the inlet and outlet bulk fluid temperatures. The refrigerant side heat transfer is

\[
Q_F = m_F C_F (T_{F_{in}} - T_{F_{out}}) 
\]

and

\[
Q_F = h_i A_{sur} (T_{F_{av}} - T_{w_{av}}) 
\]

where \(T_{w_{av}}\) is the average wall temperature, and the surface area is based on the inside diameter of the tube.

The water-side heat transfer is

\[
Q_W = m_W C_W (T_{W_{out}} - T_{W_{in}}) 
\]

As mentioned previously, the heat transfer to the refrigerant and the heat loss from the water agreed to within ± 3%. For simplicity, the refrigerant side heat transfer was selected to calculate the tube side heat transfer coefficients as follows:
The average wall temperature in the above equation was obtained by first linearly averaging circumferential temperatures located at the center of the test section and then averaging wall thermocouple temperatures over the length of the test section. The Nusselt number was calculated using the inside diameter of the tube as the characteristic dimension.

Two-phase heat transfer

The vapor quality of fluid entering the test section was calculated from an energy balance on the preheater. The power supplied to the preheater was

$$Q_{\text{pre}} = 0.98 V_T I_s \quad (3.5)$$

where the factor of 0.98 accounted for heat losses through the insulation on the preheater tube wall. Since subcooled refrigerant entered the preheater tube, energy was required to both raise the fluid temperature to saturation and to evaporate it to the required quality, as follows:

$$Q_{\text{pre}} = Q_{\text{sens}} + Q_{\text{latent}} \quad (3.6)$$

where

$$Q_{\text{sens}} = m_F C_F (T_{\text{sat}} - T_{\text{pre_in}}) \quad (3.7)$$
and

$$Q_{\text{latent}} = m_p \cdot i_{fg} \cdot x_{\text{pre out}} \quad (3.8)$$

The latent heat of evaporation for the refrigerant in the preheater was calculated at the test-section inlet pressure.

The quality entering the test section is equal to the quality exiting the preheater, so that

$$x_{\text{in}} = \frac{Q_{\text{latent}}}{m_p \cdot i_{fg}} = x_{\text{pre out}} \quad (3.9)$$

The change of the refrigerant quality, neglecting heat losses from the test section, can be calculated as follows:

$$\Delta x = m_v \cdot C_v \cdot \frac{\Delta T_v}{m_p \cdot i_{fg}} \quad (3.10)$$

The properties of the test fluid were evaluated at the saturation temperature corresponding to the average pressure of the refrigerant flowing in the test section.

The heat transfer coefficient during evaporation or condensation can be calculated using the heat transfer from or to the water as follows:

$$h_i = \frac{Q_{\text{w}}}{A_{\text{sur}} \cdot \Delta T_v} \quad (3.11)$$

where
\[ \Delta T_v = \left( T_{v_{av}} - T_{sat} \right) \] for evaporation, and

\[ \Delta T_v = \left( T_{sat} - T_{v_{av}} \right) \] for condensation.

Results for Smooth Tube

A smooth tube study was included in the present investigation to qualify the experimental facility and to serve as a basis for obtaining the enhancement factors for finned tubes. A systematic comparison between smooth tube data and recent correlations is performed in this section.

Single-phase

Heat transfer Single-phase heat transfer tests were performed by cooling the refrigerant in the test section. Figure 3.4 shows that, the experimental data are higher than the classical Dittus-Boelter/McAdams correlation [Eq. (2.3)] by about 10%. The Petukhov-Popov equation [Eq. (2.5)], which is now widely accepted, is in excellent agreement with the data. Most of the experimental data are within ±10% of either correlation. The single-phase heat transfer coefficient was also measured using a Wilson plot technique (McAdams, 1942). At a Reynolds number of 7,025, the agreement was within 2.5% of the value obtained by a direct measurement of the wall temperature. Details of the single-phase repeatability tests are reported in Appendix C.
Figure 3.4. Comparison of experimental Nusselt numbers against predictions of the Dittus-Boelter/McAdams and Petukhov-Popov equations
It was not possible to obtain meaningful tube-side pressure drop measurements during single-phase flow, because of the limited accuracy and resolution of the installed differential manometer.

**Evaporation**

**Heat transfer** Evaporation tests for the smooth tube were important in that they formed the basis for enhancement factors presented in the next section. These smooth tube experiments were performed at about four different average qualities and at three different mass velocities. The resulting experimental heat transfer coefficient data are shown in Fig. 3.5 for the smooth tube and augmented Tube 5. As expected, the heat transfer coefficient generally increases with quality and mass flow rate. A straight line, which was plotted using a least square fit, correlated the data quite well. A series of repeatability tests showed a maximum variation of about ± 10% in the heat transfer coefficient data (Appendix C).

The evaporation data for the smooth tube were also compared with the correlations of Pujol and Stenning (1969), Shah (1982), Kandlikar (1983), and Chen (1966). All of these correlations predict heat transfer coefficients to within ± 30% of the experimental data. However, Kandlikar's correlation shows the best agreement, with the experimental data being within about ± 20% of the data as shown in Fig. 3.6.

**Pressure drop** The experimentally measured pressure drops over the test section during evaporation are shown in Fig. 3.7. A comparison with predicted pressure drops using the Lockhart and Martinelli (1949) correlation is also shown. The data points are within ± 40% of the
Figure 3.5. Evaporation heat transfer coefficients for smooth tube and augmented Tube 5
Figure 3.6. Comparison of evaporation heat transfer coefficients with predictions of Kandlikar (1983)
Figure 3.7. Comparison of smooth tube pressure drop data with predictions of the Lockhart-Martinelli (1949) correlation.
predicted values. The largest disagreement was observed at low and high mass flow rates.

**Condensation**

**Heat transfer** The condensation experiments for the smooth tube were performed at three different mass velocities and four different qualities. Typical sets of experimental data for a smooth tube and augmented Tube 7 are shown in Fig. 3.8. The data are described by linear curves. It is evident from Fig. 3.8 that the heat transfer coefficient increases with mass velocity and quality. The condensation heat transfer coefficients for the smooth tube are also compared with the correlations of Shah (1979), Cavallini and Zecchin (1974), and Traviss et al. (1972). Figure 3.9 indicates that the experimental data are within about $\pm 20\%$ of these correlations. A series of repeatability tests recorded a maximum variation of about $\pm 10\%$ in the heat transfer coefficients (Appendix C).

**Pressure drop** A comparison of experimentally measured pressure drop with the Lockhart-Martinelli (1949) correlation is presented in Fig. 3.10. The data points are within $\pm 40\%$ of the predicted values. Because of the limited accuracy and resolution of the differential manometer, the largest disagreement is observed at low mass velocities.

**Results for Micro-fin Tubes in Single-Phase Flow**

A cross-sectional view of a typical micro-fin tube was shown in Fig. 1.1. This figure also illustrates the important geometrical parameters. Table 3.2 itemizes these geometrical parameters for one
Figure 3.8. Condensation coefficients for smooth tube and augmented Tube 7
Figure 3.9. Comparison of condensation heat transfer coefficient data with predictions of Shah (1979), Traviss (1972), and Cavallini et al. (1974)
Figure 3.10. A comparison of pressure drop for smooth tube with the Lockhart-Martinelli (1949) correlation
Table 3.2. Selected geometrical parameters of the tubes

<table>
<thead>
<tr>
<th>Tube Number</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outside Diameter, O.D.</td>
<td>0.375 (9.525)</td>
<td>0.375 (9.525)</td>
<td>0.375 (9.525)</td>
<td>0.375 (9.525)</td>
<td>0.375 (9.525)</td>
<td>0.375 (9.525)</td>
</tr>
<tr>
<td>Root Diameter</td>
<td>— (8.738)</td>
<td>0.344 (8.712)</td>
<td>0.343 (8.890)</td>
<td>0.350 (8.890)</td>
<td>0.350 (8.915)</td>
<td></td>
</tr>
<tr>
<td>Tip Diameter</td>
<td>— (8.407)</td>
<td>0.331 (8.433)</td>
<td>0.332 (8.407)</td>
<td>0.331 (8.534)</td>
<td>0.336 (8.534)</td>
<td></td>
</tr>
<tr>
<td>Fin Height</td>
<td>— (0.19)</td>
<td>0.0074 (0.19)</td>
<td>0.0064 (0.16)</td>
<td>0.0063 (0.16)</td>
<td>0.0069 (0.17)</td>
<td></td>
</tr>
<tr>
<td>No. of Fins</td>
<td>— 65</td>
<td>60</td>
<td>65</td>
<td>70</td>
<td>60</td>
<td></td>
</tr>
<tr>
<td>Spiral Angle</td>
<td>— 20</td>
<td>20</td>
<td>25</td>
<td>20</td>
<td>8-10</td>
<td></td>
</tr>
<tr>
<td>Tip Geometry</td>
<td>— R***</td>
<td>F*</td>
<td>R</td>
<td>FR**</td>
<td>F</td>
<td></td>
</tr>
<tr>
<td>Valley Geometry</td>
<td>— F</td>
<td>F</td>
<td>R</td>
<td>FR</td>
<td>F</td>
<td></td>
</tr>
</tbody>
</table>

All dimensions in inch or inch² with bracketed values in mm or cm².

* F = flat tip geometry.

** FR = flat fin curving towards round.

*** R = round fin geometry.
<table>
<thead>
<tr>
<th></th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.375</td>
<td>0.375</td>
<td>0.375</td>
<td>0.375</td>
<td>0.375</td>
</tr>
<tr>
<td></td>
<td>0.351</td>
<td>0.350</td>
<td>0.352</td>
<td>0.348</td>
<td>—</td>
</tr>
<tr>
<td></td>
<td>(8.915)</td>
<td>(8.890)</td>
<td>(8.941)</td>
<td>(8.839)</td>
<td>—</td>
</tr>
<tr>
<td></td>
<td>0.342</td>
<td>0.337</td>
<td>0.336</td>
<td>0.333</td>
<td>—</td>
</tr>
<tr>
<td></td>
<td>(8.712)</td>
<td>(8.560)</td>
<td>(8.534)</td>
<td>(8.458)</td>
<td>—</td>
</tr>
<tr>
<td></td>
<td>0.0041</td>
<td>0.0061</td>
<td>0.0071</td>
<td>0.0070</td>
<td>0.0070</td>
</tr>
<tr>
<td></td>
<td>(0.10)</td>
<td>(0.15)</td>
<td>(0.18)</td>
<td>(0.18)</td>
<td>(0.22)</td>
</tr>
<tr>
<td>60</td>
<td>65</td>
<td>65</td>
<td>65</td>
<td>65</td>
<td>60</td>
</tr>
<tr>
<td>8-10</td>
<td>23-25</td>
<td>20</td>
<td>20</td>
<td>16-17.5</td>
<td></td>
</tr>
<tr>
<td>F</td>
<td>FR</td>
<td>F</td>
<td>FR</td>
<td>R</td>
<td></td>
</tr>
<tr>
<td>F</td>
<td>F</td>
<td>F</td>
<td>F</td>
<td>F</td>
<td></td>
</tr>
</tbody>
</table>
Table 3.2. concluded

<table>
<thead>
<tr>
<th>Tube Number</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pitch</td>
<td>--</td>
<td>0.016</td>
<td>0.017</td>
<td>0.016</td>
<td>0.015</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>(0.406)</td>
<td>(0.441)</td>
<td>(0.406)</td>
<td>(0.383)</td>
<td>(0.4)</td>
<td></td>
</tr>
<tr>
<td>Peak Width</td>
<td>--</td>
<td>--</td>
<td>0.0064</td>
<td>--</td>
<td>--</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>(0.16)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>(0.1)</td>
</tr>
<tr>
<td>Valley Width</td>
<td>--</td>
<td>0.0039</td>
<td>0.0043</td>
<td>--</td>
<td>--</td>
<td>0.0</td>
</tr>
<tr>
<td></td>
<td>(0.10)</td>
<td>(0.11)</td>
<td></td>
<td></td>
<td></td>
<td>(0.11)</td>
</tr>
<tr>
<td>Surface Area</td>
<td>--</td>
<td>60.48</td>
<td>56.16</td>
<td>56.16</td>
<td>61.92</td>
<td>47.4</td>
</tr>
<tr>
<td></td>
<td>(390.19)</td>
<td>(362.32)</td>
<td>(362.32)</td>
<td>(399.48)</td>
<td>(306.2)</td>
<td></td>
</tr>
<tr>
<td>$A_{\text{aug}} / A_{\text{smooth}}$</td>
<td>1.00</td>
<td>1.43</td>
<td>1.34</td>
<td>1.34</td>
<td>1.47</td>
<td>1.0</td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>7</td>
<td>8</td>
<td>9</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>----</td>
<td>------</td>
<td>------</td>
<td>------</td>
<td>------</td>
<td>------</td>
<td></td>
</tr>
<tr>
<td></td>
<td>0.018</td>
<td>0.016</td>
<td>0.016</td>
<td>0.016</td>
<td>0.018</td>
<td></td>
</tr>
<tr>
<td></td>
<td>(0.455)</td>
<td>(0.406)</td>
<td>(0.406)</td>
<td>(0.406)</td>
<td>(0.455)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>0.0041</td>
<td>—</td>
<td>0.0046</td>
<td>—</td>
<td>—</td>
<td></td>
</tr>
<tr>
<td></td>
<td>(0.10)</td>
<td>(0.12)</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td></td>
</tr>
<tr>
<td></td>
<td>0.0070</td>
<td>0.0036</td>
<td>0.0044</td>
<td>0.0030</td>
<td>0.008</td>
<td></td>
</tr>
<tr>
<td></td>
<td>(0.180)</td>
<td>(0.092)</td>
<td>(0.11)</td>
<td>(0.085)</td>
<td>(0.203)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>47.52</td>
<td>59.04</td>
<td>63.36</td>
<td>60.48</td>
<td>65.02</td>
<td></td>
</tr>
<tr>
<td></td>
<td>(306.58)</td>
<td>(380.90)</td>
<td>(408.77)</td>
<td>(390.19)</td>
<td>(419.48)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>1.13</td>
<td>1.40</td>
<td>1.50</td>
<td>1.43</td>
<td>1.54</td>
<td></td>
</tr>
</tbody>
</table>
smooth tube and the nine augmented tubes tested. Also shown are the increases in surface area referenced to the smooth tube. Typical profiles for nine different augmented fin tubes are shown in Fig. 3.11.

Experimental heat transfer coefficients were obtained for the eight different augmented tubes over a range of average Reynolds numbers from 5,000 to 11,000. These data were plotted in terms of the Nusselt-Prandtl parameter and then approximated as straight lines using least-square fits as shown in Fig. 3.12. The micro-fin tubes result in enhancement factors of 1.3 to 2.0. The corresponding increase in surface area were 1.1 to 1.5. Area is a major factor in enhancing the single-phase heat transfer. However, additional enhancement occurs due to the flow separation effects caused by the spiraling fins. The flow separation intensity is dependent on velocity; hence, many of the augmented tube curves are steeper than the smooth tube curve.

Results for Micro-fin Tubes in Evaporation

The evaporation tests for augmented tubes were performed at three different mass velocities rates and four different average qualities. A statistical analysis approach described in the next section was used to analyze the heat transfer and pressure drop data and to sort out the fin geometry effects. This section is primarily a study of the effects of mass velocity and quality. Heat transfer enhancement factors of approximately 1.3 to 2.6 were observed for the micro-fin tubes investigated in this study.
Figure 3.11. Fin profiles for enhanced tubes
Figure 3.12. Single-phase enhanced heat transfer results

*SEE TABLE 3.2 FOR TUBE DIMENSIONS

ENHANCED TUBES
SINGLE-PHASE DATA

NUSSELT-PRAKTIK PARAMETRER, Nu/Pt 0.3

REYNOLDS NUMBER, Re

10000
12000
3000
4
5
6
7
8
9
10
100
9
8
7
6
5
4
3
2
1

10
7
4
9
8
6
5
1

100
9
8
7
6
5
4
3
2
1

ENHANCED TUBES
SINGLE-PHASE DATA

*SEE TABLE 3.2 FOR TUBE DIMENSIONS
General

Heat transfer  Figures 3.13 through 3.15 indicate that heat transfer coefficients for augmented tubes are only a weak function of quality. In contrast, increases in mass velocity resulted in large increases in the heat transfer coefficient. Tube 5 and Tube 10 provided the best enhancement at the high mass velocity with the average enhancement factor being about 1.75. For the medium and high quality range at medium mass velocity, comparable enhancement factors in the range of 1.8 to 2.1 were observed for Tube 5 and Tube 10. At low mass velocity and high average quality, Tube 10 resulted in an enhancement factor of 2.6. In summary, Tube 5 and Tube 10 performed equally well at medium and high mass velocities while Tube 10 performed the best at low mass velocity.

Pressure drop  The increases in pressure drops for micro-fin tubes relative to the smooth tube, expressed as a pressure drop enhancement factor, were in the range of 0.88 to 1.8 (Figs. 3.16 to 3.18). The pressure drop enhancement factors of less than unity could have resulted from the changes in the flow patterns affecting the frictional component of the total pressure drop as well as inaccuracies in the pressure drop measurements at lower qualities. In general, the pressure drop enhancement factor was observed to be lower than the heat transfer enhancement factor.

The pressure drop increased with increases in the average quality and the mass velocity. However, at a given mass velocity, the pressure drop was not a strong function of quality. Tube 6 had the least pressure drop at low and medium mass velocity. This was to be expected
Figure 3.13. Evaporation heat transfer coefficients at low mass velocity

MASS VELOCITY = 197-241 kg/(m²·s)
PRESSURE = 312-346 kN/m²

*SEE TABLE 3.2 FOR TUBE DIMENSIONS
Figure 3.14. Evaporation heat transfer coefficients at medium mass velocity

MASS VELOCITY = 340-402 kg/(m^2·s)
PRESSURE = 321-345 kN/m^2

*SEE TABLE 3.2 FOR TUBE DIMENSIONS
Figure 3.15. Evaporation heat transfer coefficients at high mass velocity
MASS VELOCITY = 197-241 kg/(m²·s)
PRESSURE = 312-346 kN/m²

*SEE TABLE 3.2 FOR TUBE DIMENSIONS

Figure 3.16. Evaporation pressure drops at low mass velocity
MASS VELOCITY = 340-402 kg/(m^2·s)
PRESSURE = 321-345 kN/m^2

*SEE TABLE 3.2 FOR TUBE DIMENSIONS

Figure 3.17. Evaporation pressure drops at medium mass velocity
Figure 3.18. Evaporation pressure drops at high mass velocity
since the fin height was 35% less than the other tubes. Surprisingly, at high mass velocity, Tube 8 had smaller pressure drops.

**Analysis of results**

An investigation of the effects of geometrical parameters such as tip shape, valley shape, fin height, number of fins, and spiral angle required a detailed analysis of the data. The methodology for this analysis and the results are presented in this section.

**Methodology**

An extensive analysis was performed on the heat transfer and pressure drop data for the smooth tube and the nine augmented tubes. Since the experimental data was measured at different values of quality, a procedure for obtaining data at the same value of quality was developed. This procedure required plotting straight lines through the heat transfer coefficient/quality data plotted in Figs. 3.13 through 3.15. Thus, straight-line equations were obtained for each tube at each of the three mass velocities. Using these equations, heat transfer coefficients were then recalculated at qualities of 0.25, 0.50, and 0.75. The enhancement factors were then obtained by comparing the augmented tube heat transfer coefficient data with the smooth tube data at the three values of quality. The above analysis was performed for low, medium, and high mass velocities. It should be noted that the uncertainty in the experimental data was approximately ± 10% due to propagation of error (Appendix D). Tables 3.3 through 3.5 summarize the heat transfer enhancement factors for all tubes.

The above procedure was repeated for the pressure drop data reported in Figs. 3.16 through 3.18. Tables 3.3 through 3.5 summarize
Table 3.3. Evaporation heat transfer (pressure drop) enhancement factors for low mass velocity

<table>
<thead>
<tr>
<th>Tube</th>
<th>$X = 0.25$</th>
<th>$X = 0.50$</th>
<th>$X = 0.75$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td></td>
<td>(1.00)</td>
<td>(1.00)</td>
<td>(1.00)</td>
</tr>
<tr>
<td>2</td>
<td>1.67</td>
<td>1.65</td>
<td>1.64</td>
</tr>
<tr>
<td></td>
<td>(2.80)</td>
<td>(1.32)</td>
<td>(1.07)</td>
</tr>
<tr>
<td>3</td>
<td>1.75</td>
<td>1.68</td>
<td>1.63</td>
</tr>
<tr>
<td></td>
<td>(2.12)</td>
<td>(1.36)</td>
<td>(1.23)</td>
</tr>
<tr>
<td>4</td>
<td>1.96</td>
<td>1.90</td>
<td>1.85</td>
</tr>
<tr>
<td></td>
<td>(3.43)</td>
<td>(1.46)</td>
<td>(1.13)</td>
</tr>
<tr>
<td>5</td>
<td>1.68</td>
<td>1.88</td>
<td>2.04</td>
</tr>
<tr>
<td></td>
<td>(2.34)</td>
<td>(1.45)</td>
<td>(1.30)</td>
</tr>
<tr>
<td>6</td>
<td>1.65</td>
<td>1.74</td>
<td>1.81</td>
</tr>
<tr>
<td></td>
<td>(1.25)</td>
<td>(1.11)</td>
<td>(1.09)</td>
</tr>
<tr>
<td>7</td>
<td>1.57</td>
<td>1.65</td>
<td>1.72</td>
</tr>
<tr>
<td></td>
<td>(1.93)</td>
<td>(1.28)</td>
<td>(1.17)</td>
</tr>
<tr>
<td>8</td>
<td>1.43</td>
<td>1.59</td>
<td>1.73</td>
</tr>
<tr>
<td></td>
<td>(1.61)</td>
<td>(1.37)</td>
<td>(1.33)</td>
</tr>
<tr>
<td>9</td>
<td>1.44</td>
<td>1.62</td>
<td>1.77</td>
</tr>
<tr>
<td></td>
<td>(2.50)</td>
<td>(1.39)</td>
<td>(1.20)</td>
</tr>
<tr>
<td>10</td>
<td>1.72</td>
<td>2.22</td>
<td>2.64</td>
</tr>
<tr>
<td></td>
<td>(2.12)</td>
<td>(1.54)</td>
<td>(1.44)</td>
</tr>
</tbody>
</table>

*Underlining indicates best performance.
Table 3.4. Evaporation heat transfer (pressure drop) enhancement factors for medium mass velocity

<table>
<thead>
<tr>
<th>Tube</th>
<th>$X = 0.25$</th>
<th>$X = 0.50$</th>
<th>$X = 0.75$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td></td>
<td>(1.00)</td>
<td>(1.00)</td>
<td>(1.00)</td>
</tr>
<tr>
<td>2</td>
<td>1.44</td>
<td>1.51</td>
<td>1.56</td>
</tr>
<tr>
<td></td>
<td>(1.72)</td>
<td>(1.54)</td>
<td>(1.45)</td>
</tr>
<tr>
<td>3</td>
<td>1.52</td>
<td>1.55</td>
<td>1.57</td>
</tr>
<tr>
<td></td>
<td>(1.17)</td>
<td>(1.46)</td>
<td>(1.60)</td>
</tr>
<tr>
<td>4</td>
<td>1.41</td>
<td>1.53</td>
<td>1.63</td>
</tr>
<tr>
<td></td>
<td>(0.97)</td>
<td>(1.26)</td>
<td>(1.42)</td>
</tr>
<tr>
<td>5</td>
<td>1.65</td>
<td>1.83</td>
<td>1.97</td>
</tr>
<tr>
<td></td>
<td>(1.32)</td>
<td>(1.63)</td>
<td>(1.79)</td>
</tr>
<tr>
<td>6</td>
<td>1.41</td>
<td>1.51</td>
<td>1.59</td>
</tr>
<tr>
<td></td>
<td>(0.88)</td>
<td>(1.14)</td>
<td>(1.30)</td>
</tr>
<tr>
<td>7</td>
<td>1.67</td>
<td>1.62</td>
<td>1.58</td>
</tr>
<tr>
<td></td>
<td>(1.16)</td>
<td>(1.43)</td>
<td>(1.57)</td>
</tr>
<tr>
<td>8</td>
<td>1.55</td>
<td>1.57</td>
<td>1.58</td>
</tr>
<tr>
<td></td>
<td>(1.17)</td>
<td>(1.46)</td>
<td>(1.61)</td>
</tr>
<tr>
<td>9</td>
<td>1.62</td>
<td>1.57</td>
<td>1.53</td>
</tr>
<tr>
<td></td>
<td>(1.26)</td>
<td>(1.49)</td>
<td>(1.61)</td>
</tr>
<tr>
<td>10</td>
<td>1.81</td>
<td>1.99</td>
<td>2.13</td>
</tr>
<tr>
<td></td>
<td>(1.07)</td>
<td>(1.40)</td>
<td>(1.57)</td>
</tr>
</tbody>
</table>

*Underlining indicates best performance.*
Table 3.5. Evaporation heat transfer (pressure drop) enhancement factors for high mass velocity

<table>
<thead>
<tr>
<th>Tube</th>
<th>X = 0.25</th>
<th>X = 0.50</th>
<th>X = 0.75</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td></td>
<td>(1.00)</td>
<td>(1.00)</td>
<td>(1.00)</td>
</tr>
<tr>
<td>2</td>
<td>1.37</td>
<td>1.50</td>
<td>1.60</td>
</tr>
<tr>
<td></td>
<td>(0.97)</td>
<td>(1.08)</td>
<td>(1.15)</td>
</tr>
<tr>
<td>3</td>
<td>1.43</td>
<td>1.61</td>
<td>1.74</td>
</tr>
<tr>
<td></td>
<td>(1.04)</td>
<td>(1.22)</td>
<td>(1.33)</td>
</tr>
<tr>
<td>4</td>
<td>1.62</td>
<td>1.69</td>
<td>1.74</td>
</tr>
<tr>
<td></td>
<td>(1.13)</td>
<td>(1.31)</td>
<td>(1.42)</td>
</tr>
<tr>
<td>5</td>
<td>1.70</td>
<td>1.73</td>
<td>1.76</td>
</tr>
<tr>
<td></td>
<td>(1.38)</td>
<td>(1.47)</td>
<td>(1.52)</td>
</tr>
<tr>
<td>6</td>
<td>1.50</td>
<td>1.59</td>
<td>1.64</td>
</tr>
<tr>
<td></td>
<td>(0.98)</td>
<td>(1.18)</td>
<td>(1.30)</td>
</tr>
<tr>
<td>7</td>
<td>1.69</td>
<td>1.70</td>
<td>1.72</td>
</tr>
<tr>
<td></td>
<td>(1.03)</td>
<td>(1.33)</td>
<td>(1.51)</td>
</tr>
<tr>
<td>8</td>
<td>1.66</td>
<td>1.62</td>
<td>1.59</td>
</tr>
<tr>
<td></td>
<td>(1.21)</td>
<td>(1.37)</td>
<td>(1.47)</td>
</tr>
<tr>
<td>9</td>
<td>1.46</td>
<td>1.43</td>
<td>1.42</td>
</tr>
<tr>
<td></td>
<td>(1.19)</td>
<td>(1.33)</td>
<td>(1.41)</td>
</tr>
<tr>
<td>10</td>
<td>1.82</td>
<td>1.78</td>
<td>1.75</td>
</tr>
<tr>
<td></td>
<td>(1.03)</td>
<td>(1.38)</td>
<td>(1.60)</td>
</tr>
</tbody>
</table>

*Underlining indicates best performance.*
the increase in the pressure drop with respect to the smooth tube. However, it should be noted that the extrapolation curves at low mass velocity and low qualities do not result in valid pressure drops because of the resolution of the differential manometer.

It should be stressed that two-phase enhancement factors could have also been computed using heat transfer coefficients and pressure drops predicted by correlations for the smooth tube. However, due to large amounts of uncertainties in the predictions, the present analysis was carried out using experimental data only (Appendix A).

A total of 162 enhancement factors for heat transfer coefficients and pressure drops was created using the above procedure. It is important to note that a specific performance evaluation criteria for two-phase flows has not been established in the literature. In order to identify the most promising fin geometry, tubes with heat transfer enhancement factors within 10% (based on uncertainty) of the maximum value at each quality have been marked in the appropriate tables. In addition, tubes with pressure drop enhancement factors within 10% of the minimum value (based on uncertainty), also a desirable characteristic, have been noted in the Tables 3.3 through 3.5. Table 3.6 summarizes the number of times that each tube was within 10% of the maximum and minimum heat transfer and pressure drop, respectively. It should be noted that the outer limits of the enhancement factors could be more than that of 10%. Also, though a large range of mass velocities were reported in Figs. 3.15 through 3.15, the majority of the data points were obtained in a relatively narrower range. Additional subset analysis using the
Table 3.6. Tube performance ranking for evaporation heat transfer and pressure drop

<table>
<thead>
<tr>
<th>Tube No.</th>
<th>First Category in Heat Transfer</th>
<th>First Category in Pressure Drop</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(X = 0.25) (X = 0.50) (X = 0.75) Total</td>
<td>(X = 0.25) (X = 0.50) (X = 0.75) Total</td>
</tr>
<tr>
<td>2</td>
<td>- - 1 1</td>
<td>1 - 2 4</td>
</tr>
<tr>
<td>3</td>
<td>- 1 1 2</td>
<td>1 - - 1</td>
</tr>
<tr>
<td>4</td>
<td>2 1 1 4</td>
<td>1 - 2 3</td>
</tr>
<tr>
<td>5</td>
<td>2 2 2 6</td>
<td>- - - 0</td>
</tr>
<tr>
<td>6</td>
<td>- - 1 1</td>
<td>3 3 2 8</td>
</tr>
<tr>
<td>7</td>
<td>2 1 1 4</td>
<td>1 - 1 2</td>
</tr>
<tr>
<td>8</td>
<td>1 1 1 3</td>
<td>- - - 0</td>
</tr>
<tr>
<td>9</td>
<td>1 - - 1</td>
<td>- - 1 1</td>
</tr>
<tr>
<td>10</td>
<td>2 3 3 8</td>
<td>- - - 0</td>
</tr>
</tbody>
</table>
methodology described above was used to analyze a specific geometrical parameter.

Effect of geometrical parameters

Surface area  In the present analysis, the heat transfer enhancement factors of 1.3 to 2.6 correspond to increases in surface area of 1.13 to 1.54. It should, however, be noted that eight of the total micro-fin tubes had a narrow range of area increase, i.e., 1.34 to 1.54. A careful examination of Tables 3.3 through 3.5 and Figs. 3.19 through 3.21 indicates that the heat transfer coefficients generally increase with increasing surface area, notably at medium and high mass velocities. A similar correspondence of increase in pressure drop with surface area is also noted from Tables 3.3 through 3.5.

Tip shape

Heat transfer  Tubes 2, 8, and 9 can be used to investigate variations in the fin tip geometry since their other geometrical parameters are similar. Specifically, Tube 2 has a sharp tip, Tube 8 a flat tip, and Tube 9 a round tip. The subset ranking of these three tubes resulted in the best performance for Tube 8, but Tube 2 performed very close to it. However, both of these tubes performed better than Tube 9.

It can be speculated that fins with wide tip geometry result in a locally higher heat transfer area for thin film evaporation. This could explain the better performance of Tube 8. It is also believed that fins with sharp peak geometries result in generally thinner liquid films which, in turn, result in higher heat transfer.
MASS VELOCITY = 197 - 241 kg/(m².s)
PRESSURE = 312 - 346 kN/m²

△ LOW QUALITY, X = 0.25
○ MEDIUM QUALITY, X = 0.50
□ HIGH QUALITY, X = 0.75

Figure 3.19. Effect of surface area on evaporation enhancement factors at low mass velocity
Figure 3.20. Effect of surface area on evaporation enhancement factors at medium mass velocity
Figure 3.21. Effect of surface area on evaporation enhancement factors at high mass velocity

MASS VELOCITY = 483 - 593 kg/(m²·s)
PRESSURE = 317 - 351 kN/m²

△ LOW QUALITY, X = 0.25
○ MEDIUM QUALITY, X = 0.50
□ HIGH QUALITY, X = 0.75
It is evident from the results that peak geometry does not affect the heat transfer performance appreciably. The fact that the flat tip geometry yields slightly better heat transfer performance is important since mechanical expansion will flatten fin tips.

Pressure drop Among the three tubes mentioned above, Tube 2 with a sharp tip resulted in the lowest increase in pressure drop. The pressure drops for Tube 8 and Tube 9 at medium and high mass velocities were comparable.

Valley shape

Heat transfer  Tube 4 with a round valley and Tube 7 with a flat valley can be compared since their other geometrical parameters are similar. Even though Tube 7 has a slightly less tip curvature and a slightly lower fin height than Tube 4, the tubes are similar enough so that a qualitative comparison should be valid.

The subset analysis indicates that Tube 4 is better than Tube 7. Also, at low mass velocities, Tube 4 resulted in a maximum of 25% higher heat transfer coefficient than Tube 7. At medium and high mass velocities, both tubes performed equally well.

Pressure drop  Tube 4 with a rounded valley resulted in lower pressure drops than Tube 7 with a flat valley. However, it is important to note that the difference in the increase of pressure drop between these tubes is only about 10 to 15%. Hence, the effect of valley geometry on pressure drop is concluded to be insignificant.

Fin height

Heat transfer  Tube 7 has a 20% lower fin height than Tube 9. Both fin configurations are similar in that they have a round tip
geometry and a flat valley geometry. At a low mass velocity, the evaporation heat transfer for both tubes was approximately the same. Enhancement factors of 1.5 to 1.7 at medium mass velocities for both tubes are noted. At high mass velocity, 20% higher heat transfer coefficients were recorded for Tube 7. Comparing the data for the entire range of experimental test conditions, Tube 7 resulted in as much as 30% higher heat transfer coefficients.

**Pressure drop** Tube 7 had pressure drop lower than those for Tube 9 by up to 15%.

**Number of fins**

**Heat transfer** Tube 9 with 65 fins and Tube 5 with 70 fins were compared using subset analysis. This represents only a 7.7% difference in the number of fins. It should also be noted that a minor difference in the valley geometry exists; Tube 9 has a flat valley while Tube 5 has a flat valley region with a minor curvature in it. Excluding a comparable performance of both tubes at medium and high mass velocities for low qualities, Tube 5 outperformed Tube 9. Hence, micro-fin tubes having a larger number of fins are more desirable.

**Pressure drop** At high mass velocities and high qualities, Tube 5 had up to 15% higher pressure drops than Tube 9. At low mass velocities, the pressure drop data for both tubes are comparable. However, at a medium mass velocity, Tube 9 performed better. Overall, insignificant differences in the pressure drop were observed for the two tubes.
Spiral angle

Heat transfer
Tube 6 with a spiral angle of 8-10° and Tube 7 with a 23-25° spiral angle are compared. It should be noted that a difference in the fin height between these two tubes exists. In addition, Tube 7 has round peak geometry as compared to flat peak shape of Tube 6. In general, Tube 7 performed better than Tube 6, especially at lower qualities. The performance of Tube 6 was comparable with Tube 7, even though the fin height differed considerably. Tube 6 should be better because of lower fin height but Tube 7 performed better. This indicates a strong effect of spiral angle on heat transfer performance. Considering the fluid to follow the grooves of the micro-fin tubes, it can be speculated that higher fluid velocities occur with higher spiral angles. This, in turn, results in higher heat transfer coefficients.

Pressure drop
Overall, Tube 6 resulted in the lowest pressure drops of all the micro-finned tubes. This is probably due to the lower fin height.

Results for Micro-fin Tubes in Condensation

The condensation tests for all augmented tubes were performed at three ranges of mass velocities and four different average qualities. Figures 3.22 through 3.24 show least squares fit curves of heat transfer coefficients as a function of quality for the nine augmented tubes.

General

Heat transfer
From the present analysis, it is clear that increases in quality and mass velocity result in increases in the heat
Figure 3.22. Condensation heat transfer coefficients at low mass velocity
MASS VELOCITY = 367-389 kg/(m²⋅s)
PRESSURE = 333-389 kN/m²
*SEE TABLE 3.2 FOR TUBE DIMENSIONS

Figure 3.23. Condensation heat transfer coefficients at medium mass velocity
Figure 3.24. Condensation coefficients at high mass velocity

MASS VELOCITY = 565-579 kg/(m²·s)
PRESSURE = 321-340 kN/m²

*SEE TABLE 3.2 FOR TUBE DIMENSIONS
transfer coefficients; however, quality has a more pronounced effect on heat transfer at high mass velocities. These figures also show that higher enhancement factors occur at high mass velocities. For example, the maximum enhancement factor of 3.83 occurred at a high mass velocity while the highest enhancement factor at medium or low mass velocities was 2.20.

**Pressure drop** The increase in pressure drop of micro-finned tubes relative to the smooth tube was in the range of 4% to 102%. Figures 3.25 through 3.27 indicate the weak dependence of pressure drop on the quality. However, the pressure drop increases with increasing mass velocity.

**Analysis of results**

**Methodology** The procedure was for analyzing the geometry effects for the nine augmented tubes is similar to that described for evaporation. This procedure involved using the straight line equations plotted in Figs. 3.22 through 3.24 to recalculate heat transfer coefficients at qualities of 0.25, 0.50, and 0.75. Enhancement factors were then calculated at each quality for the three different mass velocity ranges. Tables 3.7 through 3.9 itemize the enhancement factors for all tubes. A similar objective analysis was repeated for the pressure drop data. The results are graphically presented in Figs. 3.25 through 3.27. Tables 3.7 through 3.9 summarize the increase in the pressure drop referenced to the smooth tube. Again, the usage of correlations for predicting the smooth tube heat transfer coefficients and pressure drops was eliminated due to large uncertainties in the predicted values.
Table 3.7. Condensation heat transfer (pressure drop) enhancement factors at low mass velocity

<table>
<thead>
<tr>
<th>Tube</th>
<th>X = 0.25</th>
<th>X = 0.50</th>
<th>X = 0.75</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td></td>
<td>(1.00)</td>
<td>(1.00)</td>
<td>(1.00)</td>
</tr>
<tr>
<td>2</td>
<td>1.84</td>
<td>1.82</td>
<td>1.82</td>
</tr>
<tr>
<td></td>
<td>(1.83)</td>
<td>(1.64)</td>
<td>(1.61)</td>
</tr>
<tr>
<td>3</td>
<td>2.10*</td>
<td>2.01</td>
<td>1.96</td>
</tr>
<tr>
<td></td>
<td>(3.23)</td>
<td>(2.02)</td>
<td>(1.79)</td>
</tr>
<tr>
<td>4</td>
<td>1.77</td>
<td>1.60</td>
<td>1.51</td>
</tr>
<tr>
<td></td>
<td>(1.40)</td>
<td>(1.14)</td>
<td>(1.08)</td>
</tr>
<tr>
<td>5</td>
<td>2.15</td>
<td>2.14</td>
<td>2.13</td>
</tr>
<tr>
<td></td>
<td>(2.63)</td>
<td>(1.70)</td>
<td>(1.53)</td>
</tr>
<tr>
<td>6</td>
<td>1.51</td>
<td>1.61</td>
<td>1.67</td>
</tr>
<tr>
<td></td>
<td>(0.61)</td>
<td>(1.05)</td>
<td>(1.13)</td>
</tr>
<tr>
<td>7</td>
<td>1.78</td>
<td>1.94</td>
<td>2.03</td>
</tr>
<tr>
<td></td>
<td>(1.67)</td>
<td>(1.36)</td>
<td>(1.30)</td>
</tr>
<tr>
<td>8</td>
<td>1.71</td>
<td>1.86</td>
<td>1.93</td>
</tr>
<tr>
<td></td>
<td>(1.29)</td>
<td>(1.27)</td>
<td>(1.27)</td>
</tr>
<tr>
<td>9</td>
<td>2.19</td>
<td>2.20</td>
<td>2.20</td>
</tr>
<tr>
<td></td>
<td>(1.98)</td>
<td>(1.47)</td>
<td>(1.37)</td>
</tr>
<tr>
<td>10</td>
<td>2.22</td>
<td>2.80</td>
<td>3.10</td>
</tr>
<tr>
<td></td>
<td>(1.65)</td>
<td>(1.60)</td>
<td>(1.59)</td>
</tr>
</tbody>
</table>

*Underlining indicates best performance.
Table 3.8. Condensation heat transfer (pressure drop) enhancement factors at medium mass velocity

<table>
<thead>
<tr>
<th>Tube</th>
<th>X = 0.25</th>
<th>X = 0.50</th>
<th>X = 0.75</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td></td>
<td>(1.00)</td>
<td>(1.00)</td>
<td>(1.00)</td>
</tr>
<tr>
<td>2</td>
<td>1.67</td>
<td>1.81</td>
<td>1.89</td>
</tr>
<tr>
<td></td>
<td>(1.54)</td>
<td>(1.48)</td>
<td>(1.46)</td>
</tr>
<tr>
<td>3</td>
<td>1.65</td>
<td>1.83</td>
<td>1.93</td>
</tr>
<tr>
<td></td>
<td>(1.66)</td>
<td>(1.38)</td>
<td>(1.29)</td>
</tr>
<tr>
<td>4</td>
<td>1.50</td>
<td>1.53</td>
<td>1.54</td>
</tr>
<tr>
<td></td>
<td>(1.35)</td>
<td>(1.37)</td>
<td>(1.38)</td>
</tr>
<tr>
<td>5</td>
<td>1.84</td>
<td>2.05*</td>
<td>2.16</td>
</tr>
<tr>
<td></td>
<td>(1.52)</td>
<td>(1.54)</td>
<td>(1.54)</td>
</tr>
<tr>
<td>6</td>
<td>1.45</td>
<td>1.73</td>
<td>1.88</td>
</tr>
<tr>
<td></td>
<td>(1.60)</td>
<td>(1.45)</td>
<td>(1.40)</td>
</tr>
<tr>
<td>7</td>
<td>1.51</td>
<td>2.10</td>
<td>2.42</td>
</tr>
<tr>
<td></td>
<td>(1.37)</td>
<td>(1.47)</td>
<td>(1.50)</td>
</tr>
<tr>
<td>8</td>
<td>1.91</td>
<td>2.15</td>
<td>2.28</td>
</tr>
<tr>
<td></td>
<td>(1.04)</td>
<td>(1.12)</td>
<td>(1.14)</td>
</tr>
<tr>
<td>9</td>
<td>2.13</td>
<td>2.26</td>
<td>2.34</td>
</tr>
<tr>
<td></td>
<td>(1.25)</td>
<td>(1.31)</td>
<td>(1.32)</td>
</tr>
<tr>
<td>10</td>
<td>2.13</td>
<td>2.54</td>
<td>2.76</td>
</tr>
<tr>
<td></td>
<td>(1.97)</td>
<td>(1.65)</td>
<td>(1.56)</td>
</tr>
</tbody>
</table>

*Underlining indicates best performance.
Table 3.9. Condensation heat transfer (pressure drop) enhancement factors at high mass velocity

<table>
<thead>
<tr>
<th>Tube</th>
<th>X = 0.25</th>
<th>X = 0.50</th>
<th>X = 0.75</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td></td>
<td>(1.00)</td>
<td>(1.00)</td>
<td>(1.00)</td>
</tr>
<tr>
<td>2</td>
<td>1.87</td>
<td>1.95</td>
<td>1.99</td>
</tr>
<tr>
<td></td>
<td>(1.40)</td>
<td>(1.41)</td>
<td>(1.41)*</td>
</tr>
<tr>
<td>3</td>
<td>1.65</td>
<td>2.24</td>
<td>2.53</td>
</tr>
<tr>
<td></td>
<td>(1.65)</td>
<td>(1.57)</td>
<td>(1.54)</td>
</tr>
<tr>
<td>4</td>
<td>1.65</td>
<td>2.06</td>
<td>2.25</td>
</tr>
<tr>
<td></td>
<td>(1.36)</td>
<td>(1.33)</td>
<td>(1.32)</td>
</tr>
<tr>
<td>5</td>
<td>2.33</td>
<td>2.05</td>
<td>1.90</td>
</tr>
<tr>
<td></td>
<td>(1.56)</td>
<td>(1.37)</td>
<td>(1.30)</td>
</tr>
<tr>
<td>6</td>
<td>1.65</td>
<td>1.76</td>
<td>1.81</td>
</tr>
<tr>
<td></td>
<td>(1.22)</td>
<td>(1.27)</td>
<td>(1.29)</td>
</tr>
<tr>
<td>7</td>
<td>1.77</td>
<td>2.30</td>
<td>2.56</td>
</tr>
<tr>
<td></td>
<td>(1.34)</td>
<td>(1.39)</td>
<td>(1.42)</td>
</tr>
<tr>
<td>8</td>
<td>2.20</td>
<td>3.29</td>
<td>3.83</td>
</tr>
<tr>
<td></td>
<td>(1.44)</td>
<td>(1.48)</td>
<td>(1.50)</td>
</tr>
<tr>
<td>9</td>
<td>1.88</td>
<td>2.46</td>
<td>2.74</td>
</tr>
<tr>
<td></td>
<td>(1.32)</td>
<td>(1.33)</td>
<td>(1.33)</td>
</tr>
<tr>
<td>10</td>
<td>2.05</td>
<td>2.39</td>
<td>2.56</td>
</tr>
<tr>
<td></td>
<td>(1.43)</td>
<td>(1.50)</td>
<td>(1.52)</td>
</tr>
</tbody>
</table>

*Underlining indicates best performance.
Figure 3.26. Condensation pressure drops at medium mass velocity

MASS VELOCITY = 197-250 kg/(m²·s)
PRESSURE = 320-366 kN/m²
*SEE TABLE 3.2 FOR TUBE DIMENSIONS
MASS VELOCITY = 367-389 kg/(m²·s)
PRESSURE = 333-361 kN/m²
*SEE TABLE 3.2 FOR TUBE DIMENSIONS

Figure 3.27. Condensation pressure drops at high mass velocity
Figure 3.27. Condensation pressure drops at high mass velocity

MASS VELOCITY = 565-576 kg/(m²·s)
PRESSURE = 321-340 kN/m²
*SEE TABLE 3.2 FOR TUBE DIMENSIONS
Instead, smooth tube experimental data were used for computing the two-phase enhancement factors.

All of those tubes that performed within \( \pm 10\% \) of the maximum heat transfer enhancement factors and within \( \pm 10\% \) of the minimum pressure drop enhancement factors were considered to perform similarly. As mentioned earlier, the uncertainties in computing the enhancement factors could be more than \( \pm 10\% \) for both heat transfer and pressure drop. Again, a large range of mass velocities were reported for the present data set in Figs. 3.22 through 3.24. However, the majority of the experimental data were obtained in a relatively narrower range. A comprehensive summary of the number of firsts (\( \pm 10\% \) of the maximum) for heat transfer coefficients and for pressure drops (\( \pm 10\% \) of the minimum) is presented in Table 3.10. Tube 10 had outstanding heat transfer performance and moderate increases in pressure drop. Additional subset analysis using the methodology described above was used to analyze a specific geometric parameter.

**Factors responsible for condensation enhancement**

Three major factors are responsible for the improvement in condensation heat transfer for these finned surfaces: the increase in surface area, thinning of the condensate film by surface tension forces, and disturbances in the film by the fins. The surface area increases were in the range of 1.13 to 1.54; however, seven of the tubes had a narrow range of area increase, i.e., 1.34 to 1.54 (Figs. 3.28 through 3.30). In spite of this limited distribution of area, a careful examination of Tables 3.7 through 3.9 indicates that the heat transfer
Table 3.10. Tube performance ranking for condensation heat transfer and pressure drop

<table>
<thead>
<tr>
<th>Tube No.</th>
<th>First Category in Heat Transfer</th>
<th>First Category in Pressure Drop</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$X = 0.25$ $X = 0.50$ $X = 0.75$</td>
<td>$X = 0.25$ $X = 0.50$ $X = 0.75$</td>
</tr>
<tr>
<td>2</td>
<td>-</td>
<td>0</td>
</tr>
<tr>
<td>3</td>
<td>1 1</td>
<td>2</td>
</tr>
<tr>
<td>4</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>5</td>
<td>2 1 1</td>
<td>4</td>
</tr>
<tr>
<td>6</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>7</td>
<td>0 1</td>
<td>2</td>
</tr>
<tr>
<td>8</td>
<td>2 1 1</td>
<td>4</td>
</tr>
<tr>
<td>9</td>
<td>2 1 0</td>
<td>3</td>
</tr>
<tr>
<td>10</td>
<td>3 2 2</td>
<td>7</td>
</tr>
</tbody>
</table>
Figure 3.28. Effect of surface area on condensation enhancement factors at low mass velocity

Mass velocity = 197 - 250 kg/(m²·s)
Pressure = 320 - 366 kN/m²

- Low quality, X = 0.25
- Medium quality, X = 0.50
- High quality, X = 0.75
Figure 3.29. Effect of surface area on condensation enhancement factors at medium mass velocity.
Figure 3.30. Effect of surface area on condensation enhancement factors at high mass velocity
coefficient generally increases with increasing surface area for all mass velocities and qualities. In general, pressure drop also increases with increasing surface area.

Estimates of the void fraction were made to give an indication of the condensate film thickness. At low qualities, specifically at the reference quality of $X = 0.25$, it is likely that the fins are completely submerged in condensate. At the higher qualities of $X = 0.5$ and $X = 0.75$, however, the grooves (valleys) are only partially filled. Under these conditions, the surface-tension-driven cross-flows (Gregorig effect) are important. The film at the fin tip is thinned, whereas the film is thicker in the valley. For gravity-driven flows, the net result is usually a considerable increase in heat transfer coefficient based on either nominal (envelope) or total surface area.

Film disturbance is evidently responsible for the observed enhancement beyond the area increase at low qualities. With the spiraling fins, flow separation can occur so that there is a secondary flow in the valleys.

The influence of the detailed geometrical characteristics should be interpreted in terms of the surface tension and flow disturbance effects. In other words, it is important how the increased area is configured.

**Effect of geometrical parameters**

**Tip shape** The effects of fin peak geometry can be inferred from the data for Tubes 2, 9, and 8, which have sharp, round, and flat peaks, respectively. The other geometrical parameters are quite similar for
these three tubes. At low mass velocities, Tube 9 (round peak) performed the best for all qualities. Tube 8 (flat peak) resulted in the highest enhancement factors at high mass velocities for all qualities. However, at medium mass velocities, the performances of Tube 8 and Tube 9 were comparable. In general, Tube 2 (sharp peak) resulted in the lowest heat transfer coefficient of all three tubes for the entire range of experimentation.

It is likely that the good performance of Tube 8 is due to the large tip area for thin film evaporation. The performance improvement is especially pronounced at high qualities where, as noted earlier, the surface tension effects should be strongest. Similar conclusions concerning peak shapes were made by Tojo and co-workers (1984). Their experimental study resulted in approximately 30% better performance for micro-fin tubes with flat peaks as compared to tubes with sharp peaks. This is a fortunate result, as mechanical expansion is likely to produce flat fin tips in all tubes regardless of the initial configuration.

The generally better pressure drop performance of Tube 8 is a welcome accompaniment to the good heat transfer performance. It is possible that this is because of less drag due to flow separation although the high mass velocity data are contrary to this supposition.

Valley shape Tube 4, having a round valley, and Tube 7, having a flat valley, are suitable to compare variations in the valley shape, since other geometrical parameters are quite similar. In general, the subset ranking of these two tubes concluded that Tube 7 performs better than Tube 4; specifically, Tube 7 resulted in better heat transfer performance at medium and high qualities for all three mass velocities.
However, the performance was comparable at low qualities for all three mass velocities.

These results are quite reasonable since valley geometry is expected to be important for liquids that have very low contact angles such as refrigerants. The liquid film thickness will be less when the valley is flat than when the valley is round; hence, a higher average heat transfer coefficient is expected for the flat valley. This effect is expected at higher qualities where the fins are not flooded. For gravity-drained tubes, Mori et al. (1981) actually recommend fins having a flat valley with steep sides which, of course, are difficult to manufacture.

Tube 4 (round valley) resulted in better pressure drop performance than Tube 7 (flat valley) particularly at low mass velocities. At higher mass velocities, the pressure drop increases with Tube 7 are really quite small and do not outweigh the gain in heat transfer performance.

**Fin height**

Tubes 7 and 9 are suitable for an assessment of the influence of fin height on condensing performance, since the latter has a 20% greater fin height while other parameters are essentially the same. With only one exception, Tube 9 has higher heat transfer coefficients at all flow rates and qualities. At high qualities, the higher fins will have less of the surface covered with condensate, which means that more area is available for the effective thin film condensation. At low qualities where the fins are flooded, the higher fins should cause greater flow disturbance.
The pressure drops for Tube 7 are greater at high and medium mass velocities and less at low mass velocity. This is possibly related to the larger flooded area of the lower height fins. In any event, the higher heat transfer coefficients of Tube 9 are accompanied in many cases by lower pressure drop.

**Number of fins**  
A limited comparison of the influence of number of fins can be made by comparing the results for Tubes 5 (70 fins) and 9 (65 fins). With one exception, Tube 9 performs better over the entire range of mass velocities and qualities. This is likely due to the larger interfin spacing or valley flow area. Part of the additional enhancement with Tube 9 is likely due to the flat valley.

Once again, lower pressure drops accompany the higher enhancement. This is probably due to the larger flooded area of the tube with more fins.

**Spiral angle**  
Tubes 6 (8 to 10°) and 7 (23 to 25°) represent the spiral angle extremes; however, it is not possible to compare the performance because of the difference in heights. In addition, Tube 7 has slightly more roundness in its peak geometry. Even so, Tube 7 is consistently better, most likely because of both height and angle. It is unlikely that the gentle twist represented by Tube 6 is as effective as tighter twists. Considering larger angles, the enhancement is greater for Tube 5 (20°) than for Tube 4 (25°). Although the comparison is not strictly valid because of the lower area of Tube 4 and varying roundness in the peak and valley geometry, the data suggest that intermediate angles are more effective.
Conclusions

The water-cooled annular test section has proven to be an effective arrangement for studying condensation of refrigerants in augmented tubes.

Single-phase study

The heat transfer enhancement factors for single-phase flow were in the range of 1.31 to 2.0 for the nine micro-fin tubes tested. The increase in the surface area and the flow separation effects caused by spiraling fins appear to be the major factors in enhancing the single-phase heat transfer.

Evaporation study

Evaporation heat transfer and pressure drop for the micro-fin tubes, increased with mass velocity and quality, similarly to smooth tubes. Evaporation heat transfer enhancement factors were in the range of 1.3 to 2.6 for all nine augmented tubes. The pressure drop increased at most by a factor of 1.80. Additional conclusions are

1. The fin peak geometry had only a minor effect on heat transfer performance. However, tubes having flat or sharp peak geometry are preferred. No significant effect of peak shape on pressure drop was observed.

2. Rounded valleys increased heat transfer performance. In contrast, the valley shape did not affect the pressure drop.
3. Surprisingly, Tube 7 performed better than Tube 9 even though it had a lower fin height. However, insignificant effects of fin heights on pressure drop were noticed.

4. Tube 5 having 70 fins outperformed Tube 9 with 65 fins. Hence, tubes having larger numbers of fins are more desirable. Overall, increasing the number of fins (approximately 7.7%) resulted in insignificant differences in the pressure drops.

5. From the present analysis, tubes having spiral angles greater than 10° are preferred. The velocity of the fluid following the grooves of the micro-fin tubes increases with an increase in the spiral angle, thus resulting in higher heat transfer.

Overall, Tube 10 with a round (sharp) peak, a flat valley, and 60 fins, had the best heat transfer performance of all nine augmented tubes. It should, however, be stressed that the better heat transfer performance of Tube 10 was accomplished by a modest increase in pressure drop.

Condensation study

Increases in the mass velocity and quality resulted in increases in the heat transfer and pressure drop during condensation. The effects of quality on heat transfer coefficients are more significant at high mass velocity. Enhancement factors up to 3.83 (283% increase in heat transfer coefficient) were observed. The enhancement is generally related to surface area. The specific conclusions are
1. Increases in the fin height result in increases in the heat transfer performance. However, the fin height variations did not affect the pressure drop performance significantly.

2. Tubes having 65 fins had better heat transfer performance than the tubes having 70 fins. Also, the pressure drop is lower with 65 fins.

3. Although the present data are not definitive, intermediate spiral angles (20°) appear to be more effective than smaller (~9°) or larger (~24°) angles.

Overall, Tube 10 having a round peak and a flat valley, resulted in the best heat transfer performance. Tube 6, with the lowest fin height, resulted in the best pressure drop performance, but it is unlikely that it would be considered because of its poor heat transfer characteristics.
CHAPTER IV. AUGMENTATION OF R-113 IN-TUBE EVAPORATION WITH A MICRO-FIN TUBE (ELECTRICALLY HEATED LONG TEST SECTION)

Introduction

The R-113 test rig described in the previous chapter was modified to determine the local heat transfer coefficients using a 12.5 ft (3.81 m) long electrically heated tube. This study was important for understanding the fundamental mechanisms of in-tube evaporation. Further, this improved understanding can lead to improved designs of refrigeration system evaporators. The experiments were performed on a smooth tube and a micro-fin tube (Tube 10) using the nearly constant wall heat flux boundary condition provided by the electrical heating. A detailed comparison between local and average heat transfer coefficients and pressure drops is also presented in this chapter. This comparison was carried out to verify the experimental facility described in Chapter II, and to study the effects of entrance effects on evaporation. Additional topics studied were the effect of heat flux on the heat transfer coefficient and the dependence of the mass flow rate and quality on the critical heat flux for the smooth tube.

Experimental Facility

General

The electrically heated test-section was built parallel to the water cooled/heated test section described in Chapter III. This earlier test section along with the water flow loop was isolated during the tests described in this chapter. The after-condenser loop was not
modified for the present study; hence, that description is not repeated. Since portions of the refrigerant loop were modified, the refrigerant loop is described in detail. Figures 4.1 and 4.2 are a schematic and a photographic view of the apparatus, respectively.

Refrigerant loop

Two major changes were made in the experimental apparatus described in Chapter III. The preheater was replaced by a 11.2 kW capacity boiler and a 12.5 ft (3.81 m) long test section was installed and heated using direct electrical current. Details of these new components are reported in Appendix B. The modified refrigerant loop consisted of a positive displacement pump, a boiler, a test section, a combination filter and drier unit, a degassing tank, and an after-condenser. After leaving the pump, the test fluid entered the test section via filter-drier unit and a boiler. The 11.2 kW heat capacity boiler was electrically heated using six Chromalox resistance immersion heaters. The test fluid exiting from the test section was condensed and then cooled in the after-condenser. A degassing tank was connected at the exit of the pump for the initial removal of noncondensable gases.

Two sight glasses 4 in. (10.0 cm) long and 1/2 in. (12.7 mm) I.D., were provided at each end of the test section. In addition, the test section was electrically isolated from the rest of the system using 3 in. (7.62 cm) long rubber hoses also installed at each end of the test tube. Due to considerable radial temperature gradients between the test fluid and the ambient air, a guard heater had to be installed around the
Figure 4.1. Schematic diagram of flow loop for testing enhanced tubing in evaporation
Figure 4.2. A photographic view of the test loop used for determining the local evaporation heat transfer coefficients
test section. Details of the construction and operation of the guard heater are presented in the next section.

The evaporation tests were carried out at a test fluid pressure of approximately 35 psi (241 kPa) which corresponds to a R-113 saturation temperature of approximately 180°F (82.2°C). Most of these experiments were performed with the test fluid entering the test section at 20°F (11.1°C) subcooling. Depending on the mass flow rate and the heat flux, an exit quality in the range of 0.6 to 0.98 was attainable. However, a maximum exit quality of 0.98 was achieved only during critical heat flux tests at low mass velocities. The range of parameters for the experimentation is listed in Table 4.1.

A calibrated rotameter was used to measure the test fluid flow rate. A total of 54 copper-constantan (30-gage) thermocouples measured temperatures throughout the refrigerant loop. A Meriam type differential manometer was used to measure the pressure drop along the test section. A data acquisition system was used to read and analyze the data.

Test section

The two tubes that were tested in this study were 13 ft (3.96 m) long, 3/8 in. (9.525 mm) O.D. copper tubes with approximate wall thicknesses of 0.014 in. (0.035 mm). The tubes were heated by passing electrical current through the tube wall. Two brass bushings, 1 in. (25.4 mm) long and 1/2 in. (12.7 mm) O.D., were soldered 12.5 ft (3.81 m) apart onto the test tube. These bushings were then connected to the current-carrying copper bus bars.
Table 4.1. Operating parameter range for electrically heated evaporation test facility (R-113 as a refrigerant)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass velocity</td>
<td>162,090 – 392,000 lbm/(hr*ft²)</td>
</tr>
<tr>
<td></td>
<td>(248 – 600 kg/(m²*s))</td>
</tr>
<tr>
<td>Heat flux supplied to test section</td>
<td>5,390 – 12,680 Btu/(hr*ft²)</td>
</tr>
<tr>
<td></td>
<td>(17.0 – 40.0 kW/m²)</td>
</tr>
<tr>
<td>Test fluid inlet pressure</td>
<td>46 – 50 psia</td>
</tr>
<tr>
<td></td>
<td>(324 – 343 kPa)</td>
</tr>
<tr>
<td>Quality change in the test section</td>
<td>20°C subcooled inlet – 0.98 at outlet</td>
</tr>
</tbody>
</table>
tube wall for at least 6 feet (1.82 m) in order to minimize lead loss errors. The instrumented test tube was then insulated using a 1/2 in. (12.7 mm) thick layer of Armaflex insulation manufactured by Armstrong. Additionally, a 1 in. (25.4 mm) thick layer of fiberglass insulation was wrapped onto the Armaflex insulation. The entire test section was then enclosed in a guard heater unit as explained in the next section.

In addition to evaporation heat transfer and pressure drop tests, critical heat flux experiments were performed using a smooth tube. The additional instrumentation was seven more thermocouples on the tube wall at the downstream end of the test section. Figure 4.3 indicates their location on the tube wall. These additional thermocouples were necessary for continuous monitoring and control of the tube wall temperature.

Guard heater

During the trial runs, heat transfer coefficients were observed to be significantly higher than those obtained using the short (41 in. or 1.0 m long) test sections. After a number of tests, the cause was traced to thermocouple errors. These errors were caused by the large radial temperature gradient from the tube wall (180-190°F or 82.2-87.7°C) to the room air (80°F or 26.7°C) and by the fact that the wall thermocouple is actually insulated from the tube wall. Even though these errors were small, they still had a significant effect on the heat transfer coefficient measurements because of the small temperature differences (~4-7°F or 2-4°C) between the fluid and the wall.
* ADDITIONAL THERMOCOUPLES INSTALLED FOR THE DRYOUT TESTS

1-23: THERMOCOUPLES ON THE WALL OF THE TUBE

1-5: PRESSURE TAPS ON THE TUBE

1-13: LOCATION OF DATA POINTS OBTAINED ALONG THE LENGTH OF THE TUBE

Figure 4.3. Location of thermocouples and pressure taps along the test section
As explained in the previous chapter, an accurate measurement of the test fluid pressure in the test section was necessary due to its importance in the data reduction. Hence, two calibrated pressure gages were connected in parallel for absolute pressure measurements. Five pressure taps (1/8 in. or 3.18 mm) compression fitting with one end removed, were soldered onto the test section at intervals of 3 ft (0.91 m) along the test tube. A spacing of 12 ft (3.65 m) between inlet and outlet pressure taps was maintained in each tube. These fittings were then connected to a differential manometer via a pressure selector switch using a 1/8 in. (3.18 mm) O.D. plastic (electrically-nonconductive) tube made by Imperial-Eastman. The selector switch enabled the measurement of four different pressure drops along the length of the test section using a single differential manometer. Figure 4.3 shows the location of pressure taps on the test tube surface.

A total of 23 thermocouples was used to measure the test tube wall temperatures. Figure 4.3 indicates their locations on the tube surface. It is important to note that it was necessary to electrically isolate the wall thermocouples from the test tube because of the voltage applied to the test tube. This was achieved by wrapping electrical insulation tape around the tube so that a thin layer of tape was placed between the tube wall surface and the thermocouple. Electrical tape (3M-33) was satisfactory for this purpose.

A 30 gage copper-constantan thermocouple was placed on top of each tape wrap. The thermocouple wire was then held in position using two more layers of the electrical insulation tape. After installing the thermocouple bead, each thermocouple lead was kept in contact with the
The problem was solved by eliminating the radial temperature gradient using a guard heater. The guard heater was basically designed and fabricated to maintain the outermost insulation temperature very close to the tube wall temperature (i.e., 180°F or 82.2°C). Schematic and side views of the guard heater are shown in Fig. 4.4.

The guard heater consisted of a 3 in. (7.62 cm) I.D. copper pipe with a 1/4 in. (6.35 mm) wall thickness. The selection of the inner diameter of the guard heater, i.e., 3 in. (7.62 cm), was based on the outermost diameter of the insulated test section. The 3 in. (7.62 cm) I.D. copper pipe was split in half axially. One half of the pipe was then cut into four equal segments, each 3 ft (0.91 m) in length. The other half of the copper pipe, which was 12 ft (3.65 m) long was left unsegmented. The five pieces of pipe were reassembled around the insulated test tube. Four heating tapes manufactured by Omega, each 1/2 in. (12.7 mm) wide by 12 ft (3.65 m) long with 0.73 kW capacity, were then wrapped around the four guard heater sections. Finally, the entire assembly was insulated using 1 in. (25.4 mm) thick fiberglass insulation to reduce the heat loss.

The guard heater was divided into four equal length segments with independent heating for each section because of the axial temperature gradient in the tube wall. The individual power supply for each heating tape resulted in a very controlled guard heater temperature which could be maintained close to the linear average of the wall temperatures for each section. A set of three thermocouples was installed on each 3 ft (0.91 m) long section of the guard heater for monitoring these guard heater temperatures.
GUARD HEATER THERMOCOUPLES FOR A SPLIT SECTION

HEATING TAPE
1 in. THICK FIBER GLASS INSULATION
3 in. I.D. COPPER PIPE
1/2 in. THICK GLASS INSULATION
1/2 in. THICK ARMAFLEX INSULATION
12 ft LONG TEST SECTION

Figure 4.4. Guard heater cross section
A typical wall temperature profile, with and without guard heater, is shown in Fig. 4.5. It is observed from the figure that the installation of the guard heater was essential due to the sensitivity of wall temperature measurements on the local heat transfer coefficients. However, it should be noted that the guard heater did not cover the entire test-section. This could have a direct effect on the two extreme locations, i.e., locations 1 and 13, of the test section. Additionally, the cooling of the cables could have also resulted in conduction errors at these two locations. Hence, data points at locations 1 and 13 were neglected in the present analysis. Details of the guard heater operation are included in the next section.

Experimental Procedure

Single-phase

Single-phase experiments were used to check the heat balance between the test fluid and the electrical (dc) energy supply. Additionally, single-phase data can also be used to predict the heat transfer coefficients of superheated vapor flow. As mentioned in Chapter III, the test fluid (R-113) was degassed thoroughly prior to performing experiments using the above apparatus. After completing the degassing process, the system pressure of approximately 35 psi (241 kPa) was set using the flow regulating valve and the pump bypass valve. The fluid throttle valve was opened completely. Based on the desired Reynolds numbers, the test fluid mass flow rate was set using a flow regulating valve. The test section was then heated to attain a temperature rise of approximately 45°F (25°C) in the test fluid. The
Figure 4.5. Effect of the guard heater on the wall temperature profile at medium mass velocity.
cooling water to the after-condenser was then set to attain an inlet test fluid temperature of approximately 70°F (21°C). The boiler was not used to set the test section inlet conditions during single-phase testing. The power supply to the heating tape in the guard heater was then adjusted so as to approximately equalize the temperature on the outside of the guard heater with the sectional-average tube wall temperature.

A heat balance of approximately ± 4% between the electrical power supplied to the test tube and the refrigerant side heat transfer was considered to be acceptable. After the system had reached steady state, data acquisition was initiated. A detailed description of the data taking and data reduction is presented in a subsequent section.

Evaporation

The evaporation study was conducted at three different mass flow rates and at a system pressure of approximately 35 psi (241 kPa). The degassed test fluid was set to the desired system pressure and flow rate using the fluid throttle valve, pump bypass valve, and flow regulating valve. Typically, the pressure in the test section was set to about 35 psi (241 kPa) while the pump discharge pressure was set at 70 psi (482 kPa). The pressure drop of approximately 35 psi (241 kPa) across the throttle valve was necessary for ensuring the thermal-hydraulic stability of the system. The test fluid was then heated using the boiler so as to attain a subcooling of approximately 20°F (11°C) at the inlet of the test section. This was accomplished by controlling the power supplied to the boiler using a four-gang variac system made by
General Electric. The test tube was then heated to the desired exit quality using direct electrical current. Finally, the flow rate of the coolant in the after-condenser was set so as to attain an approximate fluid temperature of $70^\circ F$ ($21^\circ C$) at the inlet of the pump. The guard heater power was then adjusted so that the guard heater temperature was close to the wall temperature of the test tube. The system was judged to have reached a steady-state condition when the temperatures, pressures, and mass flow rates of the test fluid remained unchanged for approximately 30 minutes. The data were then taken and analyzed using the data acquisition system.

Additional experiments were performed to evaluate the critical heat flux at three different mass flow rates. The critical heat flux or dryout is caused by a complete evaporation of the liquid film at some particular value of quality. This results in a significant decrease in the heat transfer coefficient which, in turn, causes a sudden rise in the tube wall temperature. Hence, it was very important to continuously monitor the wall temperatures at the downstream end during experiments, especially since the silver-soldered pressure fittings had a limiting temperature of about $400^\circ F$ ($204^\circ C$). It should be noted that dryout conditions could be sustained; however, the physical burnout of the test tube had to be avoided. Therefore, the heat flux was very carefully increased just to the point of dryout and then data were taken using the data acquisition system.
Data Reduction

Local convective heat transfer coefficients at the thirteen different locations along the test section were calculated from raw data using a data reduction program. However, due to large axial conduction losses through the bus bars soldered at each end of the test section, the two end thermocouples were not considered. As mentioned in Chapter III, the input to this data reduction program was either automatic or manual. Specifically, temperatures were read automatically while the absolute and differential pressures were read manually.

Single-phase heat transfer

The heat transfer from the test fluid was calculated as

\[ Q_F = m_F \cdot C_F \cdot (T_{F_{\text{out}}} - T_{F_{\text{in}}}) \]  \hspace{1cm} (4.1)

The specific heat, \( C_F \), was calculated at the average bulk fluid temperature, \( T_{F_{\text{av}}} = (T_{F_{\text{out}}} + T_{F_{\text{in}}})/2 \). Also, the electrical input to the test section was

\[ Q_{\text{test}} = V_T \cdot I_s \cdot 3.412 \]  \hspace{1cm} (4.2)

For most of the data points, a heat balance of approximately \( \pm 4\% \) was observed between the electrical power supplied to the test section, \( Q_{\text{test}} \), and the refrigerant side heat transfer, \( Q_F \).

As shown in Fig. 4.4, the test tube was divided into twelve sections, each 1 ft (0.30 m) long, between pressure taps 1 and 5. This
enabled the evaluation of heat transfer coefficients at thirteen equally spaced (12 in. or 0.3 m apart) locations along the test section. However, it should be noted that the data points obtained at the inlet (location 1) and outlet (location 13) of the test section were not considered in the present analysis. This was necessary due to the conduction losses resulting from the bus bars and the cooling water circulating in the cable lines.

Two different methods were used for determining the local bulk temperature of the test fluid. They are described briefly in this section. One of the methods assumed a linear variation in the bulk fluid temperatures between the inlet and outlet. In contrast, the other method used the uniform electrical energy, $Q_{\text{test}}$, to evaluate the rise in the local bulk fluid temperature for each 12 in. (0.3 m) long subsection of the test tube as follows:

$$\Delta T = \frac{Q_{\text{test}}}{A_{\text{sur}} m_P C_F} \quad (4.3)$$

The specific heat, $C_F$, in the above equation was calculated at an average bulk fluid temperature between two consecutive locations. It should be noted that the inlet (location 1) and outlet (location 13) bulk temperatures were measured experimentally. Due to the good heat balance, insignificant differences in the bulk temperatures obtained by both of these methods were observed.

A single wall thermocouple was mounted at most locations along the test section. However, at locations 1, 4, 7, 10, and 13, linear
averaging of three thermocouples, which were located at the top, bottom, and side of the test tube (90° apart), was required.

As mentioned earlier, the rate of heat transfer to the refrigerant and the input electrical power agreed to within ±4%. Therefore, the refrigerant side heat transfer was used to calculate the local heat transfer coefficient as follows:

\[ h_i = \frac{Q_f}{A_{\text{sur}} (T_w - T_{FL})} \]  \hspace{1cm} (4.4)

where \( T_{FL} \) is the linear average of the local bulk fluid temperatures computed using two different methods described above. The local Nusselt numbers were then calculated using the inside diameter of the tube as the characteristic dimension.

**Local evaporation heat transfer**

Local heat transfer coefficients for evaporation were obtained for three different mass flow rates. The power supplied to the test section was evaluated using Eq. (4.2). Since subcooled refrigerant entered the test tube, energy was required to raise the fluid temperature to saturation and to evaporate it to a desired quality, as follows:

\[ Q_{\text{test}} = Q_{\text{latent}} + Q_{\text{sens}} \]  \hspace{1cm} (4.5)

where,
and, 

\[ Q_{\text{latent}} = Q_{\text{test}} - Q_{\text{sens}} \]  

(4.7)

Using the heat transfer area, the average heat flux can be calculated as

\[ q_{\text{test}} = \frac{Q_{\text{test}}}{A_{\text{sur}}} \]  

(4.8)

As mentioned earlier, the test tube was divided into twelve 1-ft (0.3 m) long subsections having 11 different locations (neglecting values at locations 1 and 13) for determining heat transfer coefficients (see Fig. 4.3). Therefore, the test section length required to raise the inlet fluid temperature to the saturation temperature can be calculated as follows:

\[ N = \frac{Q_{\text{sens}}}{(q_{\text{test}} \frac{mD}{L_{\text{sect}}})} = \frac{Q_{\text{sens}} L}{Q_{\text{test}} L_{\text{sect}}} \]  

(4.9)

It is possible that this section is a fractional number. For example, if \( N = 2.3 \), then at location 4 the quality is
\[ X_{\text{in}} = \left( q_{\text{test}} \, m_D \right) \left( 0.7 \, L_{\text{sect}} \right) / m_F \, \Delta q_g \]  
(4.10)

For additional locations in the direction of fluid flow, the quality is calculated as

\[ X_{\text{location}} = X_{(\text{location} - 1)} + \Delta X \]  
(4.11)

where

\[ \Delta X = q_{\text{test}} \left( m_D \, L_{\text{sect}} \right) / m_F \, \Delta q_g \]  
(4.12)

Pressures were experimentally measured at locations 1, 4, 7, and 13, and linear distributions of pressure between these locations were used to estimate the local pressure values at all thirteen locations. From this, the latent heat of vaporization could be calculated for each section for use in Eq. (4.12).

Finally, the local heat transfer coefficient during evaporation was calculated using the refrigerant side heat transfer as follows:

\[ h_i = \frac{Q_{\text{test}}}{A_{\text{sur}} \left( T_w - T_{\text{sat,L}} \right)} \]  
(4.13)

Results for Smooth Tube

The results of the local heat transfer experiments for single-phase flow and evaporation are discussed in this section. As mentioned earlier, the smooth tube was used to verify the experimental facility by
comparing the results with several well-known correlations. In addition, the results served as a basis for obtaining enhancement factors for augmented tubes. The effects of heat flux on boiling heat transfer are also discussed. Finally, critical heat fluxes for several mass velocities are described.

Heat transfer

The Reynolds number range is similar to the range that was used during experiments with the short test section presented in Chapter III, namely, 5,000 to 11,000. Thus, a comparison between long and short test sections can be performed. Bulk temperature and wall temperature distributions along the length of the tube are shown in Fig. 4.6 for a typical test run. For each data run, 11 values of the local heat transfer coefficient were obtained.

Experimental data for single-phase heat transfer coefficients were compared with the classical Dittus-Boelter/McAdams equation (Eq. 2.3) and the Petukhov-Popov correlation (Eq. 2.5). The data are in good agreement (+15%) with both correlations (Figs. 4.7 and 4.8); however, the Petukhov-Popov correlation describes the data better (+10%). A detailed comparison between the short and the long test section data sets is contained in the next section.

Evaporation heat transfer coefficients were measured at three different mass flow rates (i.e., low, medium, and high). Most of the tests were conducted with 20°F (11°C) subcooling at the inlet of the test section. A typical wall temperature profile at a medium mass flow rate is shown in Fig. 4.9. Typically, for most of the evaporation tests
Figure 4.6. Wall temperature profile for single-phase heat transfer test using the long test section
Figure 4.7. Comparison of experimental single-phase heat transfer coefficient with predictions of Dittus-Boelter/McAdams correlation.
Figure 4.8. Comparison of experimental single-phase Nusselt numbers with predictions of Petukhov-Popov correlation
Figure 4.9. Wall temperature profile for evaporation heat transfer at medium mass flow rate using the long test section.

AVERAGE SYSTEM PRESSURE = 327 kN/(m$^2$·s)
MASS VELOCITY = 374 kg/(m$^2$·s)

- WALL TEMPERATURES
- LIQUID TEMPERATURES
a temperature difference of 5 to 7°F (3 to 4°C) was observed between the tube wall and the test fluid. Heat transfer coefficients for the three mass velocities are plotted in Fig. 4.10. It is interesting to observe that heat transfer coefficients are only weakly dependent on the quality. In contrast, increases in the mass velocity resulted in large increases in the heat transfer coefficients. A comparison of experimental data attained using two different length test sections (i.e., short and long) is reported in the latter part of this chapter. The short test section data were reported earlier in Chapter III.

Additional analysis consisted of comparing the experimental heat transfer coefficient data with correlations by Pujol and Stenning (1969), Kandlikar (1983), and Shah (1982). As shown in Figs. 4.11 to 4.13, most of the data were within ± 30% of the predicted values. However, it should be noted that Kandlikar's correlation shows the best agreement with the experimental data.

A study by Gouse and Dickson (1966), among others, indicated a significant dependence of the heat transfer coefficient on quality. However, this was not observed in the present case. It should be noted that the recent correlations reported by Shah (1982) and Kandlikar (1983) also predict insignificant dependence of quality on heat transfer coefficient.

The effect of heat flux on the heat transfer coefficient is shown in Figs. 4.14 through 4.16 for three different mass flow rates. The heat flux was varied by 24% to 40%, with the maximum variation of 40% occurring at low mass flow velocities. In general, an increase in heat flux increases the heat transfer coefficients at lower qualities where
Figure 4.10. Local evaporation heat transfer coefficients for the smooth tube at low, medium, and high mass velocities.
Figure 4.11. Comparison between experimental and predicted evaporation heat transfer coefficients at low mass velocity
Figure 4.12. Comparison between experimental and predicted evaporation heat transfer coefficients at medium mass velocity.

MEDIUM MASS VELOCITY = 390 Kg/(m$^2$·s)
HEAT FLUX = 24387 W/m$^2$

- SHAH CORRELATION (1982)
- KANDLIKAR CORRELATION (1983)
- PUJOL AND STENNING CORRELATION (1969)

[-30%] [-30%] [+30%] [+30%]
Figure 4.13. Comparison between experimental and predicted evaporation heat transfer coefficients at high mass velocity.
Figure 4.14. Effect of heat flux on evaporation heat transfer coefficient for the long, smooth tube at low mass velocity.
Figure 4.15. Effect of heat flux on evaporation heat transfer coefficient for the long, smooth tube at medium mass velocity
Figure 4.16. Effect of heat flux on evaporation heat transfer coefficient for the long, smooth tube at high mass velocity
nucleate boiling occurs. However, insignificant effects of heat flux were observed at higher qualities. At higher qualities, a flow pattern transition from bubbly or slug to annular flow results in the "forced convective region" which is characterized by a thin annular liquid film with convective and conduction heat transfer at the liquid-vapor interface. Past studies have shown that in the convection boiling region, variations in heat flux do not greatly affect the heat transfer coefficients (Rohsenow et al., 1985). The present data confirms that the effect of heat flux in the convective boiling region is not large.

Heat fluxes at dryout were also measured for the smooth tube. The dryout condition is characterized by a sharp increase in the wall temperature which is caused by complete evaporation of the liquid film at some critical value of quality. As is usually observed with uniformly heated tubes, the dryout is initiated at the tube exit. A typical wall temperature profile during dryout for a low mass velocity is shown in Fig. 4.17. Table 4.2 lists the dryout conditions for low, medium, and high mass velocities.

**Pressure drop**

A comparison of experimental pressure drop data with the Lockhart and Martinelli (1949) correlation is shown in Fig. 4.18. It should be noted that no correction was made for the acceleration component since it is so small compared to the friction component. The data points are within ± 40% of the predicted values with the maximum disagreement occurring at the low mass velocity. It should be noted that the rather large deviation from the correlation is not unusual for two-phase flow.
Figure 4.17. Wall temperature profile for low mass flow rate at the dryout condition
### Table 4.2. Dryout heat flux for different mass velocities (smooth tube)

<table>
<thead>
<tr>
<th>No.</th>
<th>Mass Velocity</th>
<th>Dryout Heat Flux</th>
<th>Dryout Quality</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>251.56 kg/(m²·s)</td>
<td>19.43 kW/m²</td>
<td>0.98</td>
</tr>
<tr>
<td></td>
<td>(Low mass velocity)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>464.73 kg/(m²·s)</td>
<td>33.61 kW/m²</td>
<td>0.89</td>
</tr>
<tr>
<td></td>
<td>(Medium mass velocity)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>558.00 kg/(m²·s)</td>
<td>41.02 kW/m²</td>
<td>0.67</td>
</tr>
<tr>
<td></td>
<td>(High mass velocity)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Figure 4.18. Comparison of smooth tube evaporation pressure drop with the predictions of Lockhart-Martinelli (1949)
As noted earlier, pressure drop data for single-phase flow are not reported due to the limited accuracy and resolution of the installed differential manometer.

Results for Micro-fin Tube

The results of local heat transfer coefficients for single-phase flow and evaporation using micro-fin Tube 10 are reported in this section. A systematic analysis was carried out to compute the local evaporation heat transfer enhancement factors. Additionally, increases in the pressure drops referenced to the smooth tube are reported.

Heat transfer

The single-phase heat transfer tests were conducted in the Reynolds number range of 5,000 to 11,000. Figure 4.19 indicates enhancement factors for single-phase heat transfer of approximately 1.2 to 2.0 for micro-fin Tube 10. A detailed comparison of the single-phase heat transfer data between the short and long test section is reported in the latter part of this chapter.

Evaporation heat transfer coefficients for micro-fin Tube 10 were measured at three different mass velocities. Figure 4.20 shows that the increase in the mass velocity increases the evaporation heat transfer coefficients. Additionally, the heat transfer coefficients at medium and high mass velocities were observed to be independent of the quality. In contrast, an increase in the quality increases the heat transfer coefficients at low mass velocity.
Figure 4.19. Single-phase heat transfer coefficients for micro-fin Tube 10 for the long test section
Figure 4.20. Evaporation heat transfer coefficients for micro-fin Tube 10 for the long test section
A systematic study of local enhancement factors was performed using both experimental and predicted values of the heat transfer coefficient for a smooth tube. To accomplish this, the heat transfer coefficients for the smooth tube at pressures and temperatures comparative to the enhanced tube test conditions were required. Obtaining similar test conditions for the smooth tube and enhanced tube was accomplished by either of two methods: interpolating experimental data or using predictions obtained from Kandlikar's correlation. Using the experimental approach, the local heat transfer coefficients at any appropriate quality were calculated by assuming a linear variation between known data points. Enhancement factors for Tube 10 based on these values are reported in Tables 4.3 through 4.5.

The Kandlikar correlation was also used to evaluate the local enhancement factors. This correlation was selected due to its excellent agreement with the present data set. It is important to stress that predictions of heat transfer coefficients using Kandlikar's correlation were within \( \pm 10\% \) of the experimental data. The smooth tube heat transfer coefficients were evaluated at pressures, qualities, and mass velocities corresponding to those of augmented Tube 10. Local enhancement factors were then computed at appropriate qualities for three different mass velocities. These local enhancement factors were then computed and are reported in Tables 4.3 through 4.5.

Enhancement factors obtained by using both methods were in good agreement with each other at higher qualities (i.e., convective boiling region). However, at lower qualities, enhancement factors obtained using experimental data were lower by as much as 16%. Hence, the local
Table 4.3. Local heat transfer enhancement factors for micro-fin Tube 10 at low

<table>
<thead>
<tr>
<th>No.</th>
<th>Local Quality</th>
<th>Experimental Heat Transfer Coefficients (Tube, 10) W/(m²·K)</th>
<th>Experimental Heat Transfer Coefficients (Smooth Tube) W/(m²·K)</th>
<th>Predicted Heat Transfer Coefficients Using Kandlikar's Correlation (Smooth Tube) W/(m²·K)</th>
<th>Local Heat Transfer Coefficients (Experimental)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.13</td>
<td>5462</td>
<td>3158</td>
<td>2153</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>0.20</td>
<td>6969</td>
<td>2758</td>
<td>2277</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>0.28</td>
<td>6165</td>
<td>2674</td>
<td>2388</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>0.36</td>
<td>5402</td>
<td>2526</td>
<td>2483</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>0.43</td>
<td>5712</td>
<td>2516</td>
<td>2552</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>0.51</td>
<td>6280</td>
<td>2526</td>
<td>2612</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>0.60</td>
<td>7163</td>
<td>2632</td>
<td>2661</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>0.64</td>
<td>5781</td>
<td>2947</td>
<td>2689</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>0.74</td>
<td>6581</td>
<td>3000</td>
<td>2703</td>
<td></td>
</tr>
</tbody>
</table>
at low mass velocity

<table>
<thead>
<tr>
<th>Local Enhancement Factors (Experimental Data)</th>
<th>Local Enhancement Factors (Kandlikar's Correlation)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.73</td>
<td>2.54</td>
</tr>
<tr>
<td>2.53</td>
<td>3.06</td>
</tr>
<tr>
<td>2.31</td>
<td>2.58</td>
</tr>
<tr>
<td>2.14</td>
<td>2.18</td>
</tr>
<tr>
<td>2.27</td>
<td>2.24</td>
</tr>
<tr>
<td>2.49</td>
<td>2.40</td>
</tr>
<tr>
<td>2.72</td>
<td>2.69</td>
</tr>
<tr>
<td>1.96</td>
<td>2.15</td>
</tr>
<tr>
<td>2.19</td>
<td>2.43</td>
</tr>
<tr>
<td>No.</td>
<td>Local Quality</td>
</tr>
<tr>
<td>-----</td>
<td>---------------</td>
</tr>
<tr>
<td>1</td>
<td>0.13</td>
</tr>
<tr>
<td>2</td>
<td>0.20</td>
</tr>
<tr>
<td>3</td>
<td>0.28</td>
</tr>
<tr>
<td>4</td>
<td>0.36</td>
</tr>
<tr>
<td>5</td>
<td>0.42</td>
</tr>
<tr>
<td>6</td>
<td>0.50</td>
</tr>
<tr>
<td>7</td>
<td>0.58</td>
</tr>
<tr>
<td>8</td>
<td>0.65</td>
</tr>
<tr>
<td>9</td>
<td>0.72</td>
</tr>
</tbody>
</table>
Local Enhancement Factors (Experimental Data) | Local Enhancement Factors (Kandlikar's Correlation)
---|---
1.78 | 2.19
1.71 | 2.11
1.72 | 1.99
1.97 | 2.01
1.90 | 1.97
1.82 | 1.85
1.60 | 1.64
1.64 | 1.74
1.72 | 1.83
Table 4.5. Local heat transfer enhancement factors for micro-fin Tube 10 at high air mass flow rate

<table>
<thead>
<tr>
<th>No.</th>
<th>Local Quality</th>
<th>Experimental Heat Transfer Coefficients (Tube 10) ( \text{W}/(\text{m}^2 \cdot \text{K}) )</th>
<th>Experimental Heat Transfer Coefficients (Smooth Tube) ( \text{W}/(\text{m}^2 \cdot \text{K}) )</th>
<th>Predicted Heat Transfer Coefficients Using Kandlikar's Correlation (Smooth Tube) ( \text{W}/(\text{m}^2 \cdot \text{K}) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.10</td>
<td>7306</td>
<td>5337</td>
<td>4005</td>
</tr>
<tr>
<td>2</td>
<td>0.16</td>
<td>7728</td>
<td>5289</td>
<td>4279</td>
</tr>
<tr>
<td>3</td>
<td>0.23</td>
<td>7038</td>
<td>4947</td>
<td>4534</td>
</tr>
<tr>
<td>4</td>
<td>0.31</td>
<td>7332</td>
<td>5000</td>
<td>4763</td>
</tr>
<tr>
<td>5</td>
<td>0.37</td>
<td>6828</td>
<td>5020</td>
<td>4945</td>
</tr>
<tr>
<td>6</td>
<td>0.44</td>
<td>6052</td>
<td>5211</td>
<td>5124</td>
</tr>
<tr>
<td>7</td>
<td>0.52</td>
<td>6718</td>
<td>5658</td>
<td>5298</td>
</tr>
<tr>
<td>8</td>
<td>0.58</td>
<td>6764</td>
<td>5737</td>
<td>5445</td>
</tr>
<tr>
<td>9</td>
<td>0.65</td>
<td>7099</td>
<td>5789</td>
<td>5599</td>
</tr>
</tbody>
</table>
at high mass velocity

<table>
<thead>
<tr>
<th>Local Enhancement Factors (Experimental Data)</th>
<th>Local Enhancement Factors (Kandlikar's Correlation)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.37</td>
<td>1.82</td>
</tr>
<tr>
<td>1.46</td>
<td>1.81</td>
</tr>
<tr>
<td>1.42</td>
<td>1.55</td>
</tr>
<tr>
<td>1.47</td>
<td>1.54</td>
</tr>
<tr>
<td>1.36</td>
<td>1.38</td>
</tr>
<tr>
<td>1.16</td>
<td>1.18</td>
</tr>
<tr>
<td>1.18</td>
<td>1.27</td>
</tr>
<tr>
<td>1.18</td>
<td>1.24</td>
</tr>
<tr>
<td>1.23</td>
<td>1.27</td>
</tr>
</tbody>
</table>
enhancement factors computed using the experimental data are considered for the present analysis.

A detailed comparison of the local and average evaporation heat transfer coefficients is discussed in the following section.

Pressure drop

Figure 4.21 shows the pressure drop for microfin Tube 10 at three different mass velocities. It should be noted that pressure drops for sectional average qualities of the four different 3 ft (0.99 m) long test sections were plotted. In general, an increase in mass velocity or quality increases pressure drop.

Comparison Between Short and Long Test Sections

Smooth tube

Heat transfer A systematic comparison between the data obtained using the short (41 in. or 1.0 m) and long (12.5 ft or 3.83 m) test sections was performed for both single-phase and evaporation heat transfer. The differences in these heat transfer coefficients for single-phase flow were insignificant as can be observed in Fig. 4.22.

Comparison of evaporation heat transfer coefficients using the two different length test sections at three different mass velocities are shown in Figs. 4.23 through 4.25. At low and medium mass velocities, the evaporation heat transfer coefficients are within ±10%. However, at the high mass velocity, the local heat transfer coefficients are approximately 20-40% higher than the sectional average values, with maximum differences at low qualities. In general, the heat transfer
AVERAGE SYSTEM PRESSURE = 324-343 kN/m²
HEAT FLUX = 18107-39647 W/m²
TUBE MASS VELOCITY kg/(m²·s)

△ LOW ○ MEDIUM ■ HIGH

- SMOOTH 248 390 590
- TUBE 10 248 369 600

Figure 4.21. Evaporation pressure drop for micro-fin Tube 10 for the long test section
Figure 4.22. Comparison of single-phase heat transfer coefficients for the short and long smooth tubes
Figure 4.23. Comparison of evaporation heat transfer coefficients for the short and long smooth tubes at low mass velocity.
Figure 4.24. Comparison of evaporation heat transfer coefficients for the short and long smooth tubes at medium mass velocity.
Figure 4.25. Comparison of evaporation heat transfer coefficients for the short and long smooth tubes at high mass velocity.
coefficients are weak functions of quality for both data sets. In contrast, a strong dependence of the heat transfer coefficients on the mass velocity can be observed.

There are several possible explanations that could account for the difference between the local and average heat transfer coefficients. First, the short and long test section experiments were performed with different tube wall boundary conditions. For example, the short test section was water heated so that the wall boundary condition can be roughly described as constant temperature. In contrast, the long test section was electrically heated so as to approximate a constant heat flux boundary condition. Second, heat transfer coefficients for the short test section are average values measured at a linear mean quality while the heat transfer coefficients for the long test section are local values. Due to higher heat flux conditions, the local evaporation tests were expected to result in higher heat transfer coefficients at lower qualities (Fig. 4.25). However, insignificant differences in the two different sets of data were expected at high qualities. Third, the test section inlets were different in both test rigs. Plotting data on a Baker flow regime map (Baker, 1954) showed that most of the short-section data points at the high mass velocity were in the annular flow regime. It is a well known fact that heat transfer for annular flow evaporation is very sensitive to the inlet condition (Roshenow et al., 1985) since the annular film entering the test section can be easily disturbed. Normally, a disturbance of the annular flow increases the heat transfer coefficient; however, a degradation of the coefficient cannot be ruled out.
In summary, data sets using short and long test sections agreed well at low and medium mass velocities. However, heat transfer coefficients were lower for the high mass velocity, especially at low qualities. The deviation may be due to the difference in heat flux level at low quality and inlet disturbance at high quality.

In the past, a comparison of heat transfer coefficients for evaporation was carried out by Kubanek and Miletti (1979). For a short test section quality change of 0.2 and a long test section quality change of 0.7, they showed insignificant differences in the heat transfer coefficients.

**Pressure drop** A systematic comparison of pressure drops for the short and long test sections is shown in Fig. 4.26. Pressure drops at average qualities are reported for both tubes. In the case of the short test section, pressure gradients for the average of inlet and the outlet qualities are plotted. However, for the long test section, pressure gradients have been plotted for sectional average qualities for the four different 3 ft (0.99 m) long test sections.

The results indicate excellent agreement between the two data sets at the low mass velocity. However, the long test section resulted in approximately 20-25% higher pressure drops at the medium mass velocity. Specifically, at the medium mass velocity, data for the long test section were noted to be approximately 20% higher at a quality of 0.6. More divergence in the data was observed at higher qualities. Similar results of approximately 25% higher pressure drops with long length test sections were reported by Kubanek and Miletti (1979). No specific
Figure 4.26. Comparison of evaporation pressure drops for the short and long smooth tubes
reason for this phenomenon was cited by the authors. In summary, the pressure drop agreement between two different data sets was reasonable.

**Micro-finned tube**

**Heat transfer** As mentioned earlier, Tube 10, was also tested for single-phase and evaporation heat transfer. As shown in Fig. 4.27, there is a difference in the slopes for the single-phase heat transfer data using two different lengths of the test section. This could be due to the data measurement method along with the usual experimental errors. Overall, local and average single-phase heat transfer coefficients are in good agreement.

Evaporation heat transfer coefficients are plotted in Figs. 4.28 through 4.30 for Tube 10 at three different levels of mass velocity. At low and medium mass velocities, experimental data for the short and long test sections are generally in excellent agreement. However, the average heat transfer coefficients (Table 4.6) are moderately higher than the local values for the higher qualities at medium and high mass flow velocities. The differences in the heat transfer coefficient between the short and long test section data sets could possibly be due to the variations in the mass velocity and the heat flux condition. In general, considering experimental uncertainties, it can be stated that two sets of data are in rather good agreement (Tables 4.3 through 4.6; Figs. 4.28-4.31).

**Pressure drop** A comparison of pressure gradients for Tube 10 using short and long test sections is shown in Fig. 4.31. The pressure gradients obtained using the short test section were approximately
Figure 4.27. Comparison of single-phase heat transfer results for the short and long micro-fin tubes (Tube 10).
Figure 4.28. Comparison of evaporation heat transfer coefficients for the short and long micro-fin tubes (Tube 10) at low mass velocity.
Figure 4.29. Comparison of evaporation heat transfer coefficients for the short and long micro-fin tubes (Tube 10) at medium mass flow velocity.
Figure 4.30. Comparison of evaporation heat transfer coefficients for the short and long micro-fin tubes (Tube 10) at high mass velocity.
Table 4.6. Evaporation heat transfer enhancement factors for micro-fin Tube 10 using the short test section

<table>
<thead>
<tr>
<th>Mass Flow Rate</th>
<th>Quality</th>
<th>$X = 0.25$</th>
<th>$X = 0.50$</th>
<th>$X = 0.75$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low</td>
<td></td>
<td>1.72</td>
<td>2.22</td>
<td>2.64</td>
</tr>
<tr>
<td>Medium</td>
<td></td>
<td>1.81</td>
<td>1.99</td>
<td>2.13</td>
</tr>
<tr>
<td>High</td>
<td></td>
<td>1.82</td>
<td>1.78</td>
<td>1.75</td>
</tr>
</tbody>
</table>
Figure 4.31. Comparison of evaporation pressure drops for the short and long micro-fin tubes (Tube 10)
10-15% higher than those of the long test section. Specifically, pressure gradients using the longer test sections were approximately 15% higher at high mass flow velocities. However, comparable pressure gradients were recorded for low and medium mass velocities.

Local pressure drop enhancement factors were not evaluated due to the large uncertainty (approximately ± 40%) in the available pressure drop correlations for smooth tubes. However, it is evident from the experimental data that approximately 20% higher pressure drop enhancement factors were obtained using the short length test section at medium and high mass velocities.

Conclusions

Single-phase study

The single-phase heat transfer enhancement factors of 1.2-2.0 were recorded using micro-fin Tube 10. The present investigation also resulted in insignificant differences in the single-phase heat transfer enhancement factors using short and long test sections.

Evaporation study

Heat transfer coefficients and pressure drops increased with increases in mass velocity for both the smooth tube and a micro-fin tube (Tube 10). Surprisingly, the local heat transfer coefficient was observed to be independent of quality. Heat flux increases resulted in increases in the local heat transfer coefficient in the nucleate boiling region. However, heat flux did not affect heat transfer coefficients in the forced convection boiling region. In general, the local heat
transfer enhancement factors of 1.16-2.53 were recorded using micro-fin Tube 10. It should also be noted that the heat transfer enhancement factors decreased with increases in mass velocity.

The heat transfer coefficients and enhancement factors using two different lengths (i.e., short and long) of the test section were comparable for both smooth and augmented Tube 10. When compared with data for the short test section, the smooth tube resulted in approximately 20-25% higher pressure drops at medium and high mass velocities, specifically, at higher qualities with the longer test section. The higher pressure drop using a longer test section was also reported by Kubanek and Miletti (1979). In contrast, Tube 10 resulted in approximately 10-15% higher pressure drops using the short test section at medium and high mass velocities. Overall, the pressure drop data using two different length test sections are comparable at low and medium mass velocities.
CHAPTER V. AUGMENTATION OF R-22 IN-TUBE EVAPORATION WITH A MICRO-FIN TUBE (ELECTRICALLY HEATED LONG TEST SECTION)

Introduction

Many commercial refrigeration and air-conditioning units operate with R-22; hence, a new test apparatus was built to perform in-tube evaporations tests using R-22. A smooth tube and a micro-fin tube (Tube 10) were tested for local evaporation heat transfer coefficients. This particular study was undertaken to determine local enhancement factors for Tube 10 using R-22, and to study the validity of using R-113 to model R-22 evaporation. It should be noted that local evaporation test results with micro-fin Tube 10 using R-113 were reported in Chapter IV. The effects of variations in mass velocities, qualities, and heat flux were investigated in detail. A systematic parametric study of several smooth tube correlations is also reported in this chapter.

Experimental Facility

The test apparatus was constructed similar to the R-113 experimental setup which was used previously (Chapter IV) for determining local heat transfer coefficients. However, due to the thermodynamic properties of R-22, the apparatus had to be maintained at a relatively high system pressure, ~ 150 psia (1.03 MPa), to ensure that a liquid phase always existed at the pump suction. This high system pressure was achieved using a bladder type accumulator which was attached to the discharge line of the pump. In addition to the accumulator the low operating temperatures of the evaporation tests required the
installation of a 5-ton capacity refrigeration (R-12) unit in the test apparatus. Additional features of the test apparatus included a diaphragm pump and a positive displacement flow meter. Local evaporation heat transfer tests were conducted using a 3/8 in. (9.525 mm) O.D. by 12.1 ft (3.68 m) long electrically heated test tube. A schematic of the test apparatus and a photographic view are shown in Figs. 5.1 and 5.2, respectively. Table 5.1 summarizes the operating parameters for the test facility.

Refrigerant loop

The refrigerant flow loop included a diaphragm pump, a preheater, a test section, an accumulator, a filter-dryer unit, and an after-condenser. The test fluid was circulated through the filter-drier unit, the preheater, the test section, and the after-condenser (in that order) using the diaphragm pump. A 1/2 in. (12.7 mm) O.D. preheater was electrically heated (ac power) using a 3 kW capacity Nichrome wire. The test section was a 3/8 in. (9.525 mm) O.D. by 13 ft (3.96 m) long electrically (dc power) heated copper tube. The test fluid exiting from the test section was condensed and cooled in a condenser.

Two custom-built, 4 in. (10.12 cm) long by 1/2 in. (12.7 mm) O.D., sight glasses were provided at each end of the test section. Similarly, rubber hoses were mounted at each end of the test section to electrically isolate the test section from the rest of the test rig. A rubber hose was also installed just upstream of the preheater for the purpose of electrically isolating the preheater from the rest of the test apparatus. An accumulator installed at the exit of the pump served two
Figure 5.1. Refrigerant R-22 test loop for evaporation heat transfer
Figure 5.2. A photographic view of test apparatus
Table 5.1. Operating parameter range for electrically heated evaporation test facility (R-22 as a refrigerant)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass velocity</td>
<td>177,000 – 352,000 lbm/(hr*ft²)</td>
</tr>
<tr>
<td></td>
<td>(271 – 539 kg/m²·s)</td>
</tr>
<tr>
<td>Heat flux</td>
<td>7,064 – 12,870 Btu/(hr*ft²)</td>
</tr>
<tr>
<td></td>
<td>(22,287 – 40,605 W/m²)</td>
</tr>
<tr>
<td>Saturation pressure of</td>
<td>123 – 136 psia</td>
</tr>
<tr>
<td>the test fluid</td>
<td>(848 – 937 kN/m²)</td>
</tr>
<tr>
<td>Quality range</td>
<td>subcooled inlet – 78% outlet quality</td>
</tr>
</tbody>
</table>
purposes: to maintain the test fluid in a liquid phase throughout the refrigerant flow loop and to dampen flow vibrations caused by the diaphragm pump.

A total of 44 copper-constantan (30 gage) thermocouples measured temperatures throughout the flow loop. A calibrated piston type positive displacement flow meter manufactured by Connometer measured the test fluid flow rate. Two dial type pressure gages were used to read the absolute pressure at the inlet of the test section and the differential pressure across the test section. In addition, a strain gage type pressure transducer manufactured by Sensotek was used as a check for the absolute pressure measurement. A data acquisition system, which is described in Appendix E, was used to read and analyze the experimental data.

Test section

The test section was similar to that used in determining the local heat transfer coefficients using R-113. (See Chapter IV.) However, the guard heater was not necessary for this test facility due to the operating temperature range of 60-80°F (15-27°C). A 13 ft (3.96 m) long by 3/8 in. (9.525 mm) O.D. copper tube with a typical wall thickness of 0.014 in. (0.35 mm) was used as a test tube. Two brass bushings, 1 in. (25.4 mm) long and 1/2 in. (12.7 mm) O.D., were soldered 12.1 ft (3.68 m) apart onto the test tube for electrical power connections.

Five pressure taps, constructed from 1/4 in. (6.35 mm) O.D. brass flare fittings, were welded onto the test section at intervals of 3 ft (0.91 m) along the test section. A 1/4 in. (6.35 mm) O.D. length of
rubber tubing made by Imperial Eastman connected each pressure tap to
the pressure gages and pressure transducer via a scanning valve switch. Unfortunatel
leaks in the scanning valve switch restricted the local pressure drop measurements. However, the total pressure drop across the entire length of the test section was measured using two different pressure gages. This established the local pressure drops along the test section. It should also be noted that the changes in saturation temperature were not particularly sensitive to pressure changes due to the higher operating pressures. Hence, the critical measurement of the system pressure was not essential.

A total of 23 copper-constantan thermocouples (30 gage) measured the tube wall temperature. Figure 5.3 shows the location of thermocouples and the pressure taps along the length of the test tube. The installation of wall thermocouples on the test tube was similar to that explained earlier in Chapter IV. After installation, the test tube was insulated using 1/2 in. (12.7 mm) thick Armaflex insulation material. Finally, the entire test section was enveloped in a 1 in. (25.4 mm) thick layer of fiberglas insulation.

**Condenser loop**

The condenser was used to condense and cool the vapor exiting from the test section. A mixture of water and ethylene glycol, mixed in equal proportion by weight, was used as the condenser flow loop fluid. The use of a water and ethylene glycol mixture was necessary due to the relatively low operating temperatures (~ 30°F or 10°C) of the flow loop. The low temperatures in this flow loop were achieved by using a 5 ton
Figure 5.3. Location of thermocouples and pressure taps on the test tube
capacity refrigeration unit, details of which are discussed in Appendix E.

The condenser flow loop is depicted in Fig. 5.1. The coolant fluid was circulated through a filter, rotameter, shell-and-tube heat exchanger, storage tank, and heater, in that order. The water-glycol mixture passed through the shell side of the heat exchanger thus condensing and subcooling the test fluid (R-22) on the tube side.

The water-glycol mixture was cooled to approximately 20°F (-7°C) in the storage tank using a 5 ton (R-12) refrigeration unit manufactured by Lennox. The refrigeration unit evaporator, consisting of a coil of 1/2 in. (12.7 mm) O.D. and 25 ft (7.62 m) long smooth copper tubing, was installed directly in the storage tank where it absorbed heat from the condenser test fluid (ethylene glycol-water mixture). The storage tank was heavily insulated in order to minimize heat losses. A stirrer and motor assembly mounted on top of the storage tank was used to enhance heat transfer from the water-glycol mixture to the evaporator coil.

Eight (30 gage) copper-constantan thermocouples installed throughout the condenser flow loop provided temperature measurements for controlling the test apparatus. A Bourdon type pressure gage installed at the exit of the pump measured the pressure. In addition, a calibrated rotameter manufactured by Brooks was used to measure the flow rate of the water-glycol mixture.
Experimental Procedure

Single-phase

Single-phase experiments provided a method for checking the heat balance between the electrical energy supplied to the test section and heat transfer to the R-22. Additionally, heat transfer from the superheated vapor can be evaluated using the single-phase experimental data.

The experimental procedure consisted of evacuating the test fluid flow loop for at least 24 hours prior to filling it with refrigerant R-22. This was necessary so as to eliminate noncondensable gases in the system. After setting the system pressure to approximately 180 psi (1.24 MPa), the flow regulating valve was used to attain a desired value of the Reynolds number. The test section was then electrically heated to attain a temperature difference of approximately 40°F (22.2°C) across the test section. It should be noted that the R-12 refrigeration unit was not essential for conducting single-phase experiments. However, the refrigeration unit was used since the lower system pressures resulted in relatively easier operation of the test apparatus.

In general, a heat balance of approximately ± 5% was attained between the electrical (dc) power supplied to the test section and the refrigerant side heat transfer. At steady state, experimental data were taken and then analyzed using the computerized data acquisition system.

Evaporation

The evaporation tests were performed at three different mass flow rates and a system pressure of approximately 105 psi (723 kPa). The refrigeration unit was operated so as to obtain a storage tank
temperature of approximately 15°-20°F (-6 to -9°C). Next, the water-glycol mixture was circulated through the condenser so that the saturation pressure of the test fluid could be lowered. Since the fluid saturation pressure at room temperature (~ 70°F or 21°C) is approximately 160 psi (1.10 MPa), lower saturation temperatures could only be achieved by operating the refrigeration unit and adjusting the nitrogen pressure in the bladder of the accumulator. The test fluid diaphragm pump was then started. The flow regulating valve, the fluid bypass valve, and the system throttle valve were then used to set the system pressure to approximately 105 psi (723 kPa) and the pump back pressure to about 150 psi (1.03 MPa). The test section was then electrically heated using dc electrical power. The water-glycol mixture flow was regulated so as to attain approximately 20°F (11°C) subcooling at the inlet of the test section. In addition, the exit quality of the test fluid was normally maintained in the range of 0.6 to 0.8.

Checks on the test fluid temperatures, pressures, and mass flow rates were used to judge whether a steady-state condition had been achieved. The data acquisition system was then initiated for data taking and data analyses. The data acquisition system included a Hewlett-Packard Model 9825A computer, a Hewlett-Packard Model 9856A scanner, a Hewlett-Packard Model 3456A voltmeter, an Omega electronic ice-point, two Hewlett-Packard Model 3435A digital multimeters, and a printer made by Digital. A detailed description of the procedure for taking and reducing data was presented in Chapter IV.
Results for Smooth Tube

The smooth tube study verified the experimental facility and served as a basis for determining the local enhancement factors of the augmented tube. In this section, the results for single-phase heat transfer and evaporation heat transfer and pressure drop are discussed. An attempt was also made to understand the effects of heat flux and mass velocity.

Heat transfer

Single-phase heat transfer tests were conducted in the Reynolds number range of 21,000 to 40,000. A typical tube wall temperature profile is shown in Fig. 5.4. The experimental data agreed within $\pm 10\%$ with the Dittus-Boelter/McAdams equation (Eq. 2.3; Fig. 5.5) and Petukhov-Popov correlation (Eq. 2.5; Fig. 5.6). However, the Petukhov-Popov correlation showed the better agreement with experimental data.

Typical tube wall temperature and saturation bulk fluid temperature profiles for evaporation tests are shown in Fig. 5.7. The evaporation tests were conducted at three different mass flow rates and the results showed that the heat transfer coefficient (Fig. 5.8) increased with increasing mass velocity. For example, a three fold increase in mass velocity increased the heat transfer coefficient by a factor of two. In contrast, the heat transfer coefficients were not dependent on quality.

Figures 5.9 through 5.11 shows the effects of heat flux on the heat transfer coefficients at three different mass velocities. Insignificant effects of heat flux were observed at low mass velocity. In contrast,
Figure 5.4. Wall temperature profile for single-phase heat transfer in a smooth tube
Figure 5.5. Comparison of experimental single-phase heat transfer coefficient data with Dittus-Boelter/McAdams correlation for the smooth tube
Figure 5.6. Comparison of experimental single-phase Nusselt numbers with Petukhov-Popov correlation for the smooth tube
Figure 5.7. Wall temperature profile for evaporation heat transfer at low mass flow rate for the smooth tube.
Fig. 5.8. Evaporation heat transfer coefficients at low, medium, and high mass velocities for the smooth tube.
Figure 5.9. Effect of heat flux on local evaporation heat transfer coefficients for the smooth tube at low mass velocity.
Figure 5.10. Effect of heat flux on local evaporation heat transfer coefficients for the smooth tube at medium mass velocity.
Figure 5.11. Effect of heat flux on evaporation heat transfer coefficients for the smooth tube at high mass velocity.
heat transfer coefficients increased with heat flux at high mass velocity for the entire quality range.

The experimental data were also compared with correlations of Shah (1979), Kandlikar (1983), and Pujol and Stenning (1969). Figures 5.12 through 5.14 indicates that the predicted heat transfer coefficients are within ± 30% of the experimental values. However, it should be noted that the experimental data agreed to within ± 20% with Kandlikar’s correlation.

In conclusion, increases in mass velocity increased heat transfer coefficients. In contrast, the heat transfer coefficients were independent of quality. The heat transfer coefficients were relatively more dependent on the heat flux at higher mass velocity. This is probably due to the fact that the nucleate boiling regime prevailed even at the higher qualities for high mass velocity test runs.

Pressure drop

A comparison of two-phase pressure drops with the Lockhart-Martinelli correlation (1949) shows good agreement between the experimental data and predicted values (Fig. 5.15). It should be noted that the pressure drops plotted were obtained at average qualities of approximately 0.35.

Results for Micro-finned Tube

A micro-finned tube, Tube 10, was tested for both single-phase and two-phase heat transfer performance using R-22. Local heat transfer enhancement factors for Tube 10 for three different mass velocities are
AVERAGE SYSTEM PRESSURE = 848 - 969 kN/m²
HEAT FLUX = 7064 - 12816 W/m²

Figure 5.12. Comparison of evaporation heat transfer data with predictions of Kandlikar (1983)
Figure 5.13. Comparison of evaporation heat transfer data with predictions of Shah (1982)
AVERAGE SYSTEM PRESSURE
= 848 - 969 kN/m^2
HEAT FLUX = 7064 - 12826 W/m^2

Figure 5.14. Comparison of experimental data with predictions of Pujol and Stenning (1969)
Figure 5.15. Comparison of evaporation pressure drop with predictions of Lockhart-Martinelli (1949) equation.
reported, herein. In addition, a qualitative comparison of heat transfer enhancement factors using two different refrigerants (R-22 and R-113) is described. Finally, a general pressure drop enhancement factor analysis is discussed.

**Heat transfer**

Augmented Tube 10 was tested for single-phase heat transfer in the Reynolds number range of 20,000 to 40,000. Figure 5.16 indicates that the heat transfer coefficient is 60-80% above the smooth tube correlation. The Nusselt–Prandtl parameter \((\text{Nu}/\text{Pr}^{0.4})\) vs. Reynolds number \((\text{Re})\) for Tube 10 was much steeper than that predicted by the Dittus–Boelter equation. This could be caused by flow separation effects due to the spiraling fins. The extrapolated values of enhancements factors using R-22 as a refrigerant are in good agreement with those of R-113 reported in Chapter IV.

Evaporation tests were conducted at three different mass velocities and at a system pressure of 120 psi (826.8 kPa). Figure 5.17 compares heat transfer coefficients for Tube 10 and the smooth tube. In general, heat transfer coefficients are a weak function of quality. However, increases in the mass velocity considerably increase the heat transfer coefficient. For example, at high mass velocity, the heat transfer coefficients are approximately 90% higher than at low mass velocities.

An analysis similar to that described in Chapter IV was used to obtain enhancement factors for Tube 10. However, only experimental data were used to get the smooth tube reference. Local heat transfer coefficients at the quality of interest were obtained by linear
Figure 5.16. Single-phase heat transfer coefficients for micro-fin Tube 10
Figure 5.17. Evaporation heat transfer coefficients for micro-fin Tube 10 at low, medium, and high mass velocities.
extrapolation. It should be noted that variations in the average system pressures, heat fluxes, and mass velocities are neglected in the present analysis. Tables 5.2 through 5.4 reports local enhancement factors at three different mass velocities. In general, enhancement factors of approximately 1.75, 1.25, and 1.20 were attained at low, medium, and high mass velocities, respectively. Therefore, it can be concluded that the heat transfer enhancement decreases with mass velocity. Similar observations were also noted for the experimental data with R-113 as a test fluid (Chapter IV).

A systematic comparison was made of the local enhancement factors for R-113 and R-22. Figure 5.18 indicates approximately 30% and 36% higher enhancement factors for R-113 at low and medium mass velocities, respectively. However, this difference decreased to approximately 10% at the higher mass velocities, especially at higher qualities.

It should be noted that the mass velocities for R-22 were 11-12% higher than those for R-113. Additionally, the scaling of two different refrigerants (i.e., R-113 and R-22) using the density ratio ($\rho_g/\rho_l$) and system pressure ($P_{sat}$) requires to operate R-22 testing at a system pressure of approximately 80 psia (551 kPa) (Fig. 5.19). In the present study, the system pressures were considerably higher. Considering the differences in the mass velocity, the dependence of enhancement factors on the mass velocities, the system pressures based on the density ratios, and the uncertainties in data taking and data analyses, it can be concluded that the enhancement factors for Tube 10 using both refrigerants agreed well. It can, therefore, be concluded that R-113 can be used as a test fluid for screening a variety of test tubes. This
Table 5.2. Local heat transfer enhancement factors for micro-fin Tube 10 at low mass velocity

<table>
<thead>
<tr>
<th>No.</th>
<th>Local Quality</th>
<th>Local Experimental Heat Transfer Coefficients (Tube 10) W/(m²•K)</th>
<th>Local Experimental Heat Transfer Coefficients for Smooth Tube W/(m²•K)</th>
<th>Local Enhancement Factors</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.11</td>
<td>5795</td>
<td>3344</td>
<td>1.73</td>
</tr>
<tr>
<td>2</td>
<td>0.17</td>
<td>6126</td>
<td>3478</td>
<td>1.76</td>
</tr>
<tr>
<td>3</td>
<td>0.24</td>
<td>5656</td>
<td>3171</td>
<td>1.78</td>
</tr>
<tr>
<td>4</td>
<td>0.31</td>
<td>5480</td>
<td>3087</td>
<td>1.78</td>
</tr>
<tr>
<td>5</td>
<td>0.38</td>
<td>6244</td>
<td>3152</td>
<td>1.98</td>
</tr>
<tr>
<td>6</td>
<td>0.44</td>
<td>5440</td>
<td>3150</td>
<td>1.73</td>
</tr>
<tr>
<td>7</td>
<td>0.51</td>
<td>5687</td>
<td>3552</td>
<td>1.60</td>
</tr>
<tr>
<td>8</td>
<td>0.58</td>
<td>6121</td>
<td>3323</td>
<td>1.84</td>
</tr>
<tr>
<td>9</td>
<td>0.64</td>
<td>6050</td>
<td>3166</td>
<td>1.91</td>
</tr>
<tr>
<td>10</td>
<td>0.71</td>
<td>6449</td>
<td>----</td>
<td>----</td>
</tr>
<tr>
<td>11</td>
<td>0.78</td>
<td>6360</td>
<td>----</td>
<td>----</td>
</tr>
</tbody>
</table>
Table 5.3. Local heat transfer enhancement factors for micro-fin Tube 10 at medium mass velocity

<table>
<thead>
<tr>
<th>No.</th>
<th>Local Quality</th>
<th>Local Experimental Heat Transfer Coefficients ( \text{W/(m}^2\text{K)} )</th>
<th>Local Experimental Heat Transfer Coefficients for Smooth Tube ( \text{W/(m}^2\text{K)} )</th>
<th>Local Enhancement Factors</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.10</td>
<td>5844</td>
<td>5333</td>
<td>1.28</td>
</tr>
<tr>
<td>2</td>
<td>0.16</td>
<td>7067</td>
<td>5159</td>
<td>1.37</td>
</tr>
<tr>
<td>3</td>
<td>0.22</td>
<td>6530</td>
<td>5038</td>
<td>1.29</td>
</tr>
<tr>
<td>4</td>
<td>0.28</td>
<td>6377</td>
<td>5039</td>
<td>1.26</td>
</tr>
<tr>
<td>5</td>
<td>0.35</td>
<td>7164</td>
<td>4868</td>
<td>1.47</td>
</tr>
<tr>
<td>6</td>
<td>0.41</td>
<td>6365</td>
<td>5010</td>
<td>1.27</td>
</tr>
<tr>
<td>7</td>
<td>0.47</td>
<td>6612</td>
<td>5286</td>
<td>1.25</td>
</tr>
<tr>
<td>8</td>
<td>0.53</td>
<td>6738</td>
<td>5583</td>
<td>1.21</td>
</tr>
<tr>
<td>9</td>
<td>0.59</td>
<td>6922</td>
<td>5483</td>
<td>1.26</td>
</tr>
<tr>
<td>10</td>
<td>0.65</td>
<td>7130</td>
<td>5154</td>
<td>1.38</td>
</tr>
<tr>
<td>11</td>
<td>0.71</td>
<td>7009</td>
<td>5144</td>
<td>1.36</td>
</tr>
</tbody>
</table>
### Table 5.4. Local heat transfer enhancement factors for micro-fin Tube 10 at high mass velocity

<table>
<thead>
<tr>
<th>No.</th>
<th>Quality</th>
<th>Local Experimental Heat Transfer Coefficients (Tube 10) W/(m²·K)</th>
<th>Local Experimental Heat Transfer Coefficients for Smooth Tube W/(m²·K)</th>
<th>Local Enhancement Factors</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.14</td>
<td>8136</td>
<td>6463</td>
<td>1.26</td>
</tr>
<tr>
<td>2</td>
<td>0.19</td>
<td>7285</td>
<td>6296</td>
<td>1.16</td>
</tr>
<tr>
<td>3</td>
<td>0.25</td>
<td>7340</td>
<td>6168</td>
<td>1.19</td>
</tr>
<tr>
<td>4</td>
<td>0.31</td>
<td>7966</td>
<td>5966</td>
<td>1.33</td>
</tr>
<tr>
<td>5</td>
<td>0.36</td>
<td>7340</td>
<td>6141</td>
<td>1.19</td>
</tr>
<tr>
<td>6</td>
<td>0.42</td>
<td>7414</td>
<td>6091</td>
<td>1.22</td>
</tr>
<tr>
<td>7</td>
<td>0.47</td>
<td>7359</td>
<td>6385</td>
<td>1.15</td>
</tr>
<tr>
<td>8</td>
<td>0.53</td>
<td>7605</td>
<td>6254</td>
<td>1.22</td>
</tr>
<tr>
<td>9</td>
<td>0.59</td>
<td>7827</td>
<td>6149</td>
<td>1.27</td>
</tr>
<tr>
<td>10</td>
<td>0.64</td>
<td>7961</td>
<td>6168</td>
<td>1.29</td>
</tr>
</tbody>
</table>
Figure 5.18. Comparison of enhancement factors for evaporation heat transfer using refrigerants R-113 and R-22
Figure 5.19. Pressure-density dependence of R-113 and R-22
is especially useful due to simplicity of construction and operation of a R-113 test apparatus. In addition, conclusions that were made in Chapters III and IV using R-113 are likely to be applicable to the heat transfer performance characteristic for the same tubes using R-22.

Due to the previously mentioned limitations of the test apparatus, only total pressure drops across the entire test tube were obtained. Hence, a systematic comparison of pressure drop enhancement factors was not possible.

Conclusions

For both smooth tube and micro-fin Tube 10, an increase in the mass velocity results in an increase in the evaporation heat transfer coefficient. However, the effect of quality on the evaporation heat transfer coefficient is insignificant at all three mass velocities. Also, an increase in the heat flux results in an increase in the heat transfer coefficient. This increase is especially pronounced at lower qualities and higher mass velocities.

The specific conclusions for micro-fin tubes are:

1. The single-phase enhancement factors are 1.6-1.8 for the Reynolds number range of 20,000 to 40,000.
2. Evaporation enhancement factors of 1.2 to 1.75 were recorded for R-22. An increase in the mass velocity results in a decrease in the evaporation enhancement factor. A similar trend was obtained using R-113 (Chapter IV).
3. R-22 resulted in relatively lower enhancement factors when compared with R-113 using micro-fin Tube 10. However, comparable evaporation enhancement factors were recorded at higher mass velocities. Hence, tests conducted with R-113 as a refrigerant should be quite comparable to those using R-22.
CHAPTER VI. AUGMENTATION OF R-22 IN-TUBE EVAPORATION
AND CONDENSATION WITH A MICRO-FIN TUBE
(WATER HEATED/COOLED LONG TEST SECTION)

Introduction

The final phase of experimentation consisted of performing R-22 experiments under test conditions approaching those found in actual commercial refrigeration and air-conditioning systems. The test fluid was either heated or cooled in a 12 ft long test tube enclosed by a water jacket. The average heat transfer coefficients inside the test tube during evaporation/condensation were derived from the experimentally attained overall heat transfer coefficients, and knowledge of the annulus side heat transfer coefficients. This latter coefficient was obtained from calibration tests using a Wilson plot technique. Details of the design, description, and operation of the test apparatus are discussed in this section. Data for a smooth tube and macro-fin Tube 10 were obtained with this apparatus.

Experimental Apparatus

Design

The design of the test section was based on a heat exchanger of the shell-and-tube type. Specifically, the test section was designed with water flowing in the annulus and refrigerant flowing inside the tube being tested (Fig. 6.1). The inside heat transfer coefficient was obtained from measurements of the overall heat transfer coefficient and
Figure 6.1. Annulus design drawing
knowledge of the shell side heat transfer coefficient using the procedure described below. The heat transfer between fluids is

\[ Q = U A_{\text{sur}} (\text{LMTD}) \]  

(6.1)

where LMTD is the log mean temperature difference across the test section. This heat transfer is also equal to the heat entering/leaving the water in the annulus:

\[ Q = Q_w = m_w C_w (T_{\text{in}} - T_{\text{out}}) \]  

(6.2)

Therefore, using experimental data for flow rate and temperature, it was possible to obtain \(Q_w\), LMTD, and, hence, the overall heat transfer coefficient. Ignoring the very small wall thermal resistance, the overall heat transfer coefficient is

\[ \frac{1}{U_i} = \frac{1}{h_i} + \left( \frac{A_i}{A_o} \right) \frac{1}{h_o} \]  

(6.3)

If \(h_o\) is known, the above equation can be rearranged to calculate \(h_i\), which is the average heat transfer coefficient on the refrigerant side.

\[ h_i = \frac{1}{\frac{1}{U_i} - \left( \frac{A_i}{A_o} \right) \frac{1}{h_o}} \]  

(6.4)
The outside heat transfer coefficient, \( h_o \), can be estimated from correlations available in the literature (Shah and Johnson, 1981; Rohsenow et al., 1985). However, due to the effects of the spacers in the annulus, it did not seem appropriate to use these correlations. An alternative approach was then adopted, namely, to measure the annulus side heat transfer coefficients using a Wilson plot technique as illustrated in Fig. 6.2. For a given flow rate and set of conditions in the annulus, the overall heat transfer coefficient is plotted for increasing tube-side mass flow rate. If the plotted curve is extrapolated to the coordinate, then the thermal resistance on the inside of the tube is negligible and the measured overall heat transfer coefficient is essentially equal to the outside heat transfer coefficient (i.e., annulus side heat transfer coefficient).

Design parameters and operating conditions for the experimental facility are described in Table 6.1. A 3/4 in. (19.1 mm) O.D. tube was selected as an appropriate size for the outer annulus. This selection was based on having measurable fluid temperature changes through the annulus during condensing and evaporating heat transfer tests and on obtaining outside heat transfer coefficients high enough to use a Wilson Plot technique for the calibration of the annulus.

Test facility

The test apparatus was based on the R-22 test rig that was used for determining the local evaporation coefficients in Chapter V. Figure 6.3 shows a schematic drawing of the modified test rig. In addition, a photographic view of the test apparatus is shown in Fig. 6.4. The
Figure 6.2. Graph indicating Wilson plot technique
Table 6.1. Operating parameter range for water-jacketed evaporation/condensation test facility (R-22 as a refrigerant)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass velocity</td>
<td>96,000-386,000 lbm/(hr*ft²)</td>
</tr>
<tr>
<td></td>
<td>(2143-590 (kg/m²*s))</td>
</tr>
<tr>
<td>Heat flux</td>
<td>7,064 - 12,870 Btu/(hr*ft²)</td>
</tr>
<tr>
<td></td>
<td>(22,287 - 40,605 W/m²)</td>
</tr>
<tr>
<td>Saturation temperature of the test fluid</td>
<td>Condensation: 110-130°F (43-54°C)</td>
</tr>
<tr>
<td></td>
<td>Evaporation: 50-70°F (10-21°C)</td>
</tr>
<tr>
<td>Quality range</td>
<td>Condensation: saturated vapor inlet-</td>
</tr>
<tr>
<td></td>
<td>saturated liquid outlet</td>
</tr>
<tr>
<td></td>
<td>Evaporation: 10% inlet quality-</td>
</tr>
<tr>
<td></td>
<td>saturated vapor outlet</td>
</tr>
</tbody>
</table>
Figure 6.3. A schematic of the R-22 test loop
Figure 6.4. A photographic view of the test apparatus
apparatus consisted of three major flow loops: a refrigerant flow loop, a water flow loop, and a condenser loop. This section describes the refrigeration flow loop and water flow loop. The condenser loop was identical to that described in Chapter V; hence, it is not discussed.

The test-section construction was simplified by the use of an unsegmented 12 ft long test tube which, in turn, resulted in an average heat transfer coefficient over the tube length. A smooth tube and augmented Tube 10 was tested for heat transfer and pressure drop using this approach. Table 6.1 summarizes the range of operating parameters for the tests.

**Refrigerant loop** There were two major modifications to the refrigerant flow loop that was described in Chapter V. First, an electrically (dc) heated boiler and superheater unit were installed just upstream of the test section. Second, the electrically heated test section was replaced by a counterflow heat exchanger with refrigerant flowing in the test tube and water flowing in the annulus. Therefore, an additional water flow loop was added to the modified test apparatus.

The refrigerant loop included a diaphragm pump, which circulated the test fluid through a dryer and filter unit, a boiler, a superheater, a test section, and an after-condenser. An additional component was the accumulator. The boiler consisted of a 1/2 in. (12.7 mm) O.D. by 8.64 ft (2.63 m) long stainless steel 304 tube that was direct electrical heated. A twisted tape with twist of 5 diameters to a 180° turn was inserted into the boiler tube so as to enhance the critical heat flux. The superheater consisted of a 1/2 in. (12.7 mm) O.D. copper tube with a 60 ft (18.3 m) long Nichrome wire wound around it. As mentioned
earlier, the test section was essentially a counterflow, shell-and-tube heat exchanger with refrigerant flowing through the inner tube and the cooling/heating water flowing in the annulus. The test fluid exiting from the test section was condensed and/or cooled in the after-condenser by an ethylene glycol and water mixture.

Rubber hoses provided at each end of the boiler and the superheater electrically isolated these components from the rest of the test apparatus. Additionally, sight glasses were provided at the inlet and outlet of the test section for observing the flow patterns. Considerable emphasis was also put on providing a smooth transition between the sight glasses and the test section.

Seven 30 gage copper constantan thermocouples were used to monitor the test fluid temperatures at various locations in the flow loop. Figure 6.3 shows their locations on the refrigerant flow loop. A carefully calibrated piston type flow meter manufactured by Connometr was used to measure the refrigerant mass flow rate through the loop.

Test section The test section consisted of a 3/8 in. (9.525 mm) O.D. by 13 ft (3.96 m) long copper test tube enclosed in a 3/4 in. (19.0 mm) I.D. by 12 ft (3.66 m) long annulus tube, thus, resulting in a shell-and-tube heat exchanger as shown in Fig. 6.5. Two custom-made pressure fittings that were silver-soldered 12.1 ft (3.68 m) apart onto the test tube wall provided pressure drop measurements.

To maintain concentricity of the test tube inside the annulus, five sets of spacers were placed at equal intervals along the test section. A schematic of a typical spacer assembly is shown in Fig. 6.6. Three 1/4 in. (6.35 mm) NPT female brass adapters were welded 120° apart
Figure 6.5. Details of the test section
Figure 6.6. Spacer details
circumferentially on the annulus tube. Each of these three adapters were provided with 1/16 in. (1.58 mm) size Conax fittings. Bakelite rods that were approximately 2 in. (5.08 mm) long and 1/16 in. (1.58 mm) O.D. were then inserted through these Conax fittings so as to position the test tube in the center of the test section.

Figure 6.5 shows locations of thermocouples along the test section. Two 30 gage copper-constantan thermocouples, one at each end of the test section, measured the test fluid bulk temperatures. Water side bulk temperatures were measured using two 36-gage copper-constantan thermocouples inserted in a tee-section welded to each end of the annulus tube. The inlet, outlet, and differential pressures were measured using Bourdon type pressure gages.

**Water loop** The water flow loop acted as either a heat sink or source depending on whether evaporation or condensation experiments were being performed. The water flow loop consisted of a pump, a filter, a heater, a flow meter, an accumulator, and a water cooled heat exchanger. The water was circulated through the test section via a filter and a flow meter using a centrifugal pump. An accumulator was installed at the exit of the pump to dampen fluctuations in the flow. The heat exchanger installed in the flow loop served as a heat sink during condensing experiments.

Seven thermocouples measured water temperatures at various locations in the flow loop. As mentioned earlier, waterside bulk temperatures were measured using two 36-gage copper-constantan thermocouples inserted in a Tee-section welded to each end of the test
section. The water flow rate was measured using a calibrated Water-Mag flow meter.

Annulus Calibration Test

Introduction

As mentioned earlier, annulus calibration tests using a Wilson plot approach were necessary for determining the heat transfer coefficient on the outer wall of the test tube. Unfortunately, correlations available in the literature could not be used because of flow disturbances caused by spacers and possible sagging. To obtain realistic values for the heat transfer coefficient on the annulus side using the Wilson plot technique, equal fluid thermal resistances on both sides of the test tube wall are desirable. Therefore, water was selected as a test fluid for both the shell and the tube side of the test section. It should be noted, however, that R-22 was circulated through the test tube during actual condensation/evaporation tests.

Apparatus

The test apparatus used for calibration tests was the same as described in the earlier section. However, the refrigerant flow loop had to be modified to accommodate water as a test fluid due to the noncompatibility of moisture with some of the permanent components, such as the flow meter, dryer and filter unit, and accumulator. Figure 6.7 shows the modified test apparatus that was used specifically for the calibration tests. The annulus side flow loop was identical to that described in an earlier section.
Figure 6.7. A schematic of test apparatus for annulus calibration tests
The tube side flow loop included a positive displacement gear pump, a rotameter, a boiler, a superheater, a test section, an expansion tank, and a heat exchanger. The water was circulated to the tube side of the test section via the boiler and superheater using the pump. It should be noted that even though water was passed through the superheater, there was no power in the superheater during the calibration tests. After receiving heat from the annulus-side water in the test section, the tube-side water was then cooled in the heat exchanger.

Fourteen thermocouples measured temperatures at different locations in the test apparatus. The inlet and outlet bulk temperatures for both fluids were measured using 36 gage copper-constantan thermocouples inserted in the flow streams. A calibrated rotameter measured the tube side flow rate. Details of the test procedure and data analysis are discussed in the following section.

Calibration procedure

The calibration tests were performed at 3, 4, and 5 gpm of water flowing through the annulus. Additionally, for each of these flow rates, calibration tests were conducted at three different average bulk temperatures: 80°F (26.7°C), 100°F (37.7°C), and 110°F (43.2°C). These bulk temperatures were selected on the basis of the selected R-22 condensation/evaporation experimental conditions (Table 6.1).

Calibration tests were performed by setting the flow rate on the shell side to the desired value using a flow regulating valve. At the same time, the tube side water flow rate was set to approximately 2.0 gpm using another flow regulating valve. The average tube-side
temperature was then set by adjusting the electrical (dc) power supplied
to the boiler. It should be noted that this average tube temperature,
in turn, governed the shell side average fluid temperatures.

The data-acquisition system was initiated and the experimental data
were analyzed when a quasi-steady state condition was reached. The tube
side mass flow rate was then increased until measurable bulk temperature
differences were obtained. During these tests the shell side water flow
rate and the average shell side fluid temperature were kept constant.
In general, at a particular water flow rate and temperature on the shell
side, five data points were taken by varying the mass flow rate on the
tube side. The same procedure was repeated for different mass flow
rates and different temperature conditions on the shell side. Details
of the data reduction are discussed in the following section.

Data reduction

The following data reduction procedure was adapted for calculating
heat transfer coefficients on the shell side.

The rate of heat transfer on the tube side is

\[ Q_{\text{tube}} = m_{\text{tube}} C_{\text{tube}} \left( T_{\text{tube}}^{\text{in}} - T_{\text{tube}}^{\text{out}} \right) \]  

(6.5)

while the rate of heat transfer on the shell side is

\[ Q_{\text{sh}} = m_{\text{sh}} C_{\text{sh}} \left( T_{\text{sh}}^{\text{out}} - T_{\text{sh}}^{\text{in}} \right) \]  

(6.6)
It should be noted that a heat balance between the shell-and-tube side of the test section was within \( \pm 5\% \) for most of the data sets.

The overall heat transfer coefficient is defined by

\[
U = \frac{Q_{av}}{A_{sur}(\text{LMTD})} \tag{6.7}
\]

where

\[
Q_{av} = \frac{Q_{\text{tube}} + Q_{\text{sh}}}{2} \tag{6.8}
\]

The log mean temperature difference (LMTD) across the test section is

\[
\text{LMTD} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} \tag{6.9}
\]

where

\[
\Delta T_1 = T_{V_{\text{tube, in}}} - T_{V_{\text{sh, out}}} \tag{6.10}
\]

and

\[
\Delta T_2 = T_{V_{\text{tube, out}}} - T_{V_{\text{sh, in}}} \tag{6.11}
\]

The Wilson plot technique required a plot of \(1/U\) versus \((1/V_{\text{tube}})^{0.8}\)

where \(V_{\text{tube}}\) is the velocity in the test tube. The tube side velocity was computed using the continuity equation.
The experimental data points were then correlated using a least-square fit computer program. Extrapolation of this line yields an ordinate intercept which is the outside (shell side) heat transfer coefficient.

Results

The annulus side heat transfer calibration was obtained at three different mass flow rates and at three different shell side temperatures. This was necessary so as to incorporate Prandtl number effects in the heat transfer coefficients. Extrapolation of each of the curves shown in Figs. 6.8 through 6.10 resulted in a different value for outside heat transfer coefficient depending on the shell side flow rate and average temperature. In general, for a given mass flow rate, increases in the average water temperature on the shell side resulted in increases in the heat transfer coefficient. A summary of the experimental data is reported in Table 6.2.

A systematic comparison of the experimental data against predictions by the McAdams (1942) and Kays and Leung (1963) correlations was performed (Fig. 6.11). Both of these correlations are for turbulent flow in an annular geometry. In general, experimental data were in excellent agreement with the Kays and Leung predictions. However, McAdams' correlation underpredicted the experimental data by approximately 20%. Underprediction of the data would be expected since the spacers that were used in the present experimental setup tended to
ANNULUS SIDE MASS FLOW RATE = 3.0 GPM
REYNOLDS NUMBER = 10,930 - 15,698
PRANDTL NUMBER = 3.82 - 5.51
AVERAGE SHELL SIDE BULK TEMPERATURE = 82.7, 101.4, AND 112.3°F

Figure 6.8. Calibration of annulus at low mass flow rate
Figure 6.9. Calibration of annulus at medium mass flow rate

ANNULUS SIDE MASS FLOW RATE = 4.0 GPM
REYNOLDS NUMBER = 14,444 - 20,187
PRANDTL NUMBER = 3.81 - 5.49
AVERAGE SHELL SIDE
BULK TEMPERATURE = 82.0, 101.4, AND 112.6°F
ANNULUS SIDE MASS FLOW RATE = 5.0 GPM
REYNOLD'S NUMBER = 19,601 - 28,081
PRANDTL NUMBER = 3.73 - 5.41
AVERAGE SHELL SIDE BULK TEMPERATURE = 85.1 AND 114.5°F

Figure 6.10. Calibration of annulus at high mass flow rate
Table 6.2. Annulus calibration test results

<table>
<thead>
<tr>
<th>No.</th>
<th>Shell Side Reynolds Number</th>
<th>Shell Side Average Prandtl Number</th>
<th>Average Shell Side Bulk Fluid Temperature $\nu_{\text{Fr}}$</th>
<th>Annulus $\text{Nu}/\text{Fr}^{0.4}$ Parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>16297</td>
<td>3.75</td>
<td>112.00</td>
<td>56.19</td>
</tr>
<tr>
<td>2</td>
<td>10930</td>
<td>5.51</td>
<td>82.72</td>
<td>48.25</td>
</tr>
<tr>
<td>3</td>
<td>13868</td>
<td>4.33</td>
<td>101.41</td>
<td>55.73</td>
</tr>
<tr>
<td>4</td>
<td>15698*</td>
<td>3.82</td>
<td>112.25</td>
<td>53.89</td>
</tr>
<tr>
<td>5</td>
<td>20187</td>
<td>3.81</td>
<td>112.63</td>
<td>73.16</td>
</tr>
<tr>
<td>6</td>
<td>17956</td>
<td>4.34</td>
<td>101.35</td>
<td>66.58</td>
</tr>
<tr>
<td>7</td>
<td>14444</td>
<td>5.49</td>
<td>83.09</td>
<td>52.79</td>
</tr>
<tr>
<td>8</td>
<td>19602</td>
<td>5.41</td>
<td>85.11</td>
<td>75.25</td>
</tr>
<tr>
<td>9</td>
<td>28081</td>
<td>3.73</td>
<td>114.52</td>
<td>94.15</td>
</tr>
</tbody>
</table>

* Repeatability check run.
Figure 6.11. Comparison of experimental annulus side heat transfer coefficient data against predictions of McAdams (1942) and Kays and Leung (1963)
increase the turbulence and mixing of the fluid, thus enhancing the heat transfer.

The present experimental data were correlated with the Nusselt number, Prandtl number, and Reynolds number (based on hydraulic diameter) as variables. The resulting equation is

\[
Nu = 0.023 \, Re^{0.813} \, Pr^{0.4} \tag{6.13}
\]

The slightly higher Reynolds number exponent could be due to the fact that the spacers induced more turbulence in the flow. Equation 6.13 was used throughout this study for reducing both the single-phase and two-phase experimental data.

Experimental Procedure

**Single-phase**

Single-phase heat transfer coefficients are useful in predicting the performance of the superheat region of evaporators and the subcooled region of condensers. The heat transfer tests were conducted at five different Reynolds numbers in the range of 20,000 to 40,000. Additionally, the tests were performed at approximately 180-220 psi so as to maintain the liquid phase in the test flow loop for the entire experimentation.

First, the test fluid flow rate was set to attain the desired Reynolds number using the pump bypass valve and the flow regulating valve. However, the fluid throttle valve was left fully open. The test fluid was then heated in the boiler using dc power to attain a bulk
temperature rise of approximately $40^\circ F$. The annulus side flow rate was then set to achieve a bulk temperature rise of approximately $5-8^\circ F$ ($3-4^\circ C$). At a steady state condition the heat balance between the tube side and the shell side of the test section was checked and data were acquired using the data acquisition system. For most of the experimentation, the heat balance was within $\pm 5\%$. The details of the data reduction procedure are described in the following section.

Evaporation

Evaporation tests were conducted at three different mass velocities. For most of the experimentation, a quality of 0.1 at the inlet of the test section and an outlet quality in the range of 0.95 to 1.0 was maintained. First, the R-12 refrigeration unit was started so as to lower the storage tank fluid (water-glycol mixture) temperature. The test fluid mass flow rate and system pressure were then set using the pump bypass valve, the fluid flow control valve, and the fluid throttle valve. The test fluid system pressure was maintained at approximately 100 psia (689 kPa) with a pump back pressure of approximately 200 psia (1.37 MPa). Next, the liquid flowing through the boiler was heated using dc power to attain the test section inlet quality of approximately 0.1. The superheater was not used during the evaporation experimentation.

The annulus side fluid flow rate was then set to obtain a bulk temperature change of approximately $5-10^\circ F$ ($3-6^\circ C$). The flow rate of the cooling water for the heat exchanger and the heat input to the heater installed in the water flow loop were then adjusted to attain a
quality change of approximately 0.85 in the test section. Finally, the cooling fluid flow rate for the condenser was adjusted to attain a subcooling of approximately 20-30°F (11-17°C) at the inlet of the boiler.

Repeated checks of the temperature, pressures, and the mass flow rates were used to judge the steady state condition of the refrigerant flow loop. At a steady state condition, the experimental data were acquired either manually or automatically using the data acquisition system. The detailed data reduction procedure is discussed in the following section.

**Condensation**

Similar to evaporation tests discussed in the earlier section, condensation tests were also performed at three different mass velocities. However, the test section inlet quality for most of the tests was in the range of 0.9 to 1.0. Additionally, the test fluid exiting the test section was either saturated liquid or at a quality of 0.1 for most of the condensation tests. These tests were performed at saturation pressures of approximately 250 psia (1.72 MPa).

The desired test fluid flow rate and the system pressure of approximately 250 psia (1.72 MPa) were maintained by controlling the fluid flow valve, the pump bypass valve, and the fluid throttle valve. The test fluid was then heated in the boiler tube using dc power to a quality of approximately 0.6 to 0.7. The maximum quality obtained at the outlet of the boiler was governed by the dryout condition. The liquid-vapor mixture of the test fluid exiting the boiler was further
heated to a quality of 0.9 to 1.0 in the superheater using indirect resistance heating. An autotransformer controlled the heat input to the superheater. The annulus side water flow rate was then set so that there was complete condensation and the bulk temperature rise was approximately 6–10°F (3–5°C). The mass flow rate of the ethylene-glycol mixture was then set to attain approximately 20°F (11°C) subcooling of the test fluid at the inlet of the boiler.

At the steady-state or quasi-steady state condition, as determined by the repeated checks of the temperatures, pressures, and mass flow rates, the data acquisition system was initiated. The details of the data reduction used for evaluating the heat transfer coefficients are described in the following section.

Data Reduction

The tube-side heat transfer coefficients were determined for single-phase, evaporation, and condensation heat transfer. The single-phase heat transfer coefficients could serve as a basis for correlating two-phase heat transfer data. Additionally, the heat balance between the shell and tube side of the test section obtained during the single-phase experimentation checked out the validity of the test facility. The following section describes the data reduction procedure for both single-phase and two-phase heat transfer.
Single-phase heat transfer

The heat transfer on the tube side is

\[ Q_{\text{tube}} = m_F C_F \left( T_{F_{\text{in}}} - T_{F_{\text{out}}} \right) \tag{6.14} \]

whereas the shell side heat transfer is

\[ Q_{\text{sh}} = m_w C_w \left( T_{W_{\text{out}}} - T_{W_{\text{in}}} \right) \tag{6.15} \]

It should be noted that the properties of both the fluids were evaluated at the linear average of the inlet and outlet bulk temperatures.

For most of the data points, the heat balance between the shell and tube side of the test section was within ±5%. The overall heat transfer coefficient is defined by

\[ U_i = \frac{Q_{\text{av}}}{A_{\text{surf}} (\text{LMTD})} \tag{6.16} \]

where

\[ Q_{\text{av}} = \frac{Q_{\text{tube}} + Q_{\text{sh}}}{2} \tag{6.17} \]

The log mean temperature difference (LMTD) is defined as

\[ \text{LMTD} = \frac{\Delta T_{\text{in}} - \Delta T_{\text{out}}}{\ln\left(\frac{\Delta T_{\text{in}}}{\Delta T_{\text{out}}}\right)} \tag{6.18} \]
where,

\[
\Delta T_{\text{in}} = T_{F_{\text{in}}} - T_{W_{\text{in}}}
\]  

(6.19)

and

\[
\Delta T_{\text{out}} = T_{F_{\text{out}}} - T_{W_{\text{out}}}
\]  

(6.20)

The inside heat transfer coefficient is

\[
h_i = \frac{1}{\frac{1}{U_i} - \left(\frac{A_i}{A_o}\right) \frac{1}{h_o}}
\]  

(6.21)

where

\[
h_o = 0.023 \, \text{Re}^{0.813} \, \text{Pr}^{0.4} \, (K/D_h)
\]  

(6.22)

The surface areas \(A_i\) and \(A_o\) are the inside and outside heat transfer areas of the annulus section, respectively. The Nusselt number was then calculated using the inside diameter of the test tube as a characteristic dimension.

**Two-phase heat transfer**

**Test-section inlet quality** The heat transfer in the boiler can be expressed as

\[
Q_{\text{boiler}} = (0.99 \, V_T \, I_S) \times 3.412
\]  

(6.23)
where the 0.99 accounts for the approximately 1% loss of heat from the boiler. Also,

\[ Q_{\text{boiler}} = Q_{\text{sens}} + Q_{\text{latent}} \]  

(6.24)

where

\[ Q_{\text{sens}} = m_P C_P (T_{\text{sat}} - T_{\text{boiler,in}}) \]  

(6.25)

Thus,

\[ Q_{\text{latent}} = Q_{\text{boiler}} - Q_{\text{sens}} \]  

(6.26)

The quality of the fluid exiting from the preheater is

\[ X_{\text{boiler, out}} = \frac{Q_{\text{latent}}}{m_P i_f} \]  

(6.27)

The quality at which the test fluid is exiting from the boiler is the same as the inlet quality of the superheater; hence,

\[ X_{\text{sup, in}} = X_{\text{boiler, out}} \]  

(6.28)

The amount of heat supplied to superheater is

\[ Q_{\text{sup}} = 0.98 V_T I_S \]  

(6.29)
where the 0.98 accounts for the estimated 2% heat loss from the preheater. It should be noted that a calibration determined that the power factor was near unity. The change in quality of the test fluid across the superheater is

\[
\Delta x_{\text{sup}}^{\text{out}} = \frac{Q_{\text{sup}}}{m_{\text{F}} \cdot \Delta f_{\text{ig}}} \tag{6.30}
\]

Hence, the inlet quality of refrigerant entering the test section is

\[
x_{\text{test}}^{\text{in}} = x_{\text{boiler}}^{\text{out}} + \Delta x_{\text{sup}}^{\text{out}} \tag{6.31}
\]

In general, a test section inlet quality of 0.1 to 0.15 during evaporation and 0.80 to 1.0 during condensation was maintained for most of the experimentation.

Test-section exit quality The waterside heat transfer can be stated as

\[
Q_{\text{w}} = m_{\text{w}} \cdot C_{\text{w}} \cdot (\Delta T_{\text{w}}) \tag{6.32}
\]

where

\[
\Delta T_{\text{w}}^{\text{evaporation}} = T_{\text{w}}^{\text{in}} - T_{\text{w}}^{\text{out}} \tag{6.33}
\]

\[
\Delta T_{\text{w}}^{\text{condensation}} = T_{\text{w}}^{\text{out}} - T_{\text{w}}^{\text{in}} \tag{6.34}
\]
Considering no heat loss from the annulus side of the test section,

\[ \Delta X = \frac{Q_w}{m_F} i_{fg} \]  

(6.35)

Hence,

\[ X_{\text{test out}} = X_{\text{in}} + \Delta X \text{ (evaporation)} \]  

(6.36)

and

\[ X_{\text{test out}} = X_{\text{in}} - \Delta X \text{ (condensation)} \]  

(6.37)

In general, an exit quality of 0.80-1.0 for evaporation and an exit quality of 0.1-0 fluid for condensation were attained at the exit of the test section. It should also be noted that all the thermodynamic and thermophysical properties were computed at the average pressure of the test fluid in the test section.

**Heat transfer coefficient**  
The overall heat transfer coefficient is

\[ U_i = \frac{Q_w}{A_{sui}} \text{ (LMTD)} \]  

(6.38)

where

\[ \text{LMTD} = \frac{\Delta T_{in} - \Delta T_{out}}{\ln(\Delta T_{in}/\Delta T_{out})} \]  

(6.39)

Also, for evaporation heat transfer
\[ \Delta T_{\text{in}} = T_{\text{win}} - T_{\text{Fout}} \quad (6.40) \]

and

\[ \Delta T_{\text{out}} = T_{\text{wout}} - T_{\text{Fout}} \quad (6.41) \]

However, for condensation heat transfer

\[ \Delta T_{\text{in}} = T_{\text{Fin}} - T_{\text{win}} \quad (6.42) \]

and

\[ \Delta T_{\text{out}} = T_{\text{Fout}} - T_{\text{wout}} \quad (6.43) \]

Finally, using Eqs. (6.21) and (6.22), the inside heat transfer coefficient can be computed.

**Results for Smooth Tube**

The results of the heat transfer experiments for single-phase and two-phase flows using the smooth tube are reported in this section. The smooth tube test was used to verify the experimental facility by comparing the experimental data against the predictions of several correlations. It also served as a basis for determining the enhancement factors for the augmented tubes.
Heat transfer

The single-phase heat transfer tests were conducted in the Reynolds number range of 14,000 to 24,000. The experimental heat transfer coefficients were compared with the classical Dittus-Boelter/McAdams equation (Eq. 2.3) and the Petukhov-Popov correlation (Eq. 2.5). The data are within ±10% of the predicted heat transfer coefficients (Figs. 6.12 and 6.13).

In addition to single-phase heat transfer tests, the evaporation tests were conducted at four different mass velocities. In general, the test fluid entered the test section at an inlet quality of 0.10. For most of the data points, particularly those at low and medium mass velocity, the maximum quality of the refrigerant at the exit of the test section was 0.80. It should be noted that the lowest mass velocity in the evaporation data set is lower than that utilized in earlier experiments (Chapters III through V). Figure 6.14 indicates that an increase in mass velocity increases the heat transfer coefficient. Most of the data points reported in Fig. 6.14 were obtained at a test section average quality of approximately 0.50. The range of average system pressure and quality change in the test section are indicated in Fig. 6.14.

A comparison of the experimental heat transfer coefficients against predictions of Pujol and Stenning (1969), Kandlikar (1983), and Shah (1982) was carried out and is reported in Fig. 6.15. Most of the data points are within ±20% of the predicted values.

The condensation tests were also performed at three different mass velocities and at an average quality of 0.45 in the test section. The
Figure 6.12. Comparison of single-phase heat transfer coefficients data with predictions of Dittus-Boelter/McAdams correlation

\documentclass{article}
\usepackage{amsmath}
\usepackage{siunitx}
\begin{document}
\begin{align}
\text{Nu} &= \frac{1}{0.3} \\
\text{Re} &= \frac{1}{10,000}
\end{align}
\end{document}
Figure 6.13. Comparison of single-phase Nusselt numbers with predictions of Petukhov-Popov correlation
Figure 6.14. Average evaporation heat transfer coefficients for the smooth tube.
Figure 6.15. Comparison of average condensation heat transfer coefficients with predictions of Kandlikar (1983), Shah (1982), and Pujol and Stenning (1969)
average system pressures and change in quality in the test section are indicated on Fig. 6.16. Similar to the evaporation study, the condensation heat transfer coefficients increased with mass velocity. For most of the tests, the refrigerant entered the test section as a saturated vapor. The test fluid was cooled to the saturated liquid condition using water in the annulus. The condensation heat transfer coefficients were compared against the correlations of Cavallini and Zecchin (1974), Traviss et al. (1972), and Shah (1979). Figure 6.17 indicates a good agreement (+30%) between the predicted values using these correlations and the experimental data.

**Pressure drop**

The pressure drop data for evaporation and condensation tests using the 12 ft (3.65 m) long smooth tube were correlated against the predictions of Lockhart and Martinelli (1949). The data agreed well, being within ±40% of the predicted values (Fig. 6.18). Due to the limited accuracy of the measurements, analysis for the pressure drop enhancement factors is not reported.

**Results for the Micro-fin Tube**

**Heat transfer**

The single-phase heat transfer tests were conducted in the Reynolds number range of 12,000 to 25,000 by cooling the test fluid in the test section. Single-phase enhancement factors of 1.8-2.0 were recorded using micro-fin Tube 10 (Fig. 6.19).
SYSTEM PRESSURE = 1.39-1.60 Mpa
\[ \chi_{AV} = 0.36-0.52 \]
\[ \Delta X = 0.64-0.96 \]

- LOW MASS VELOCITY = 316 kg/(m\(^2\)-s)
- MEDIUM MASS VELOCITY = 415-487 kg/(m\(^2\)-s)
- HIGH MASS VELOCITY = 534-590 kg/(m\(^2\)-s)

Figure 6.16. Average condensation heat transfer coefficients for the smooth tube
Figure 6.17. Comparison of average condensation heat transfer coefficients with predictions of Shah (1979), Traviss et al. (1972), and Cavallini et al. (1974)
Figure 6.18. Comparison of average evaporation and condensation pressure drop with predictions of Lockhart-Martinelli (1949) equation
Figure 6.19. Single-phase heat transfer coefficients for micro-fin Tube 10
The average evaporation heat transfer coefficients were obtained at three different mass velocities and at an average quality range of 0.43 to 0.55 (Fig. 6.20). This was accomplished with the quality change of 0.63 to 0.88. As mentioned earlier, the lowest mass velocity in the present data set is a step lower than the low mass velocity reported in the earlier chapters. However, the comparison of enhancement factors is made at corresponding mass velocities. Enhancement factors of approximately 1.31, 1.24, and 1.17 were recorded for low, medium, and high mass velocities, in that order.

The local evaporation enhancement factors reported earlier (Chapter V) were comparable to the average enhancement factors at high mass velocities. However, much lower enhancement was recorded at medium mass velocities using the water-jacketed test section. This can probably be explained by the fact that local evaporation data were taken with considerably lower exit qualities.

The condensation tests were also performed at three different mass velocities (Fig. 6.21). However, these mass velocities are comparable to those reported in earlier chapters. In general, average qualities in the range of 0.37-0.44 was attained with saturated fluid exiting the test section. For three different mass velocities, enhancement factors in the range of approximately 1.6-1.75 were recorded. These enhancement factors are much lower than those attained during short test section tests with R-113 as a test fluid (Chapter III). An analysis of the system pressure to density ratio \( \rho_g/\rho_1 \) base scaling required R-113 condensation tests at approximately 100 psia (Fig. 5.19), whereas the
Figure 6.20. Average evaporation heat transfer coefficients for micro-fin Tube 10.
SYSTEM PRESSURE = 152-170 MPa
X_{av} = 0.49
\Delta X = 0.71-0.96

<table>
<thead>
<tr>
<th>TUBE</th>
<th>MASS VELOCITY</th>
<th>kg/(m^2·s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>△ SMOOTH</td>
<td>281 408 524</td>
<td></td>
</tr>
<tr>
<td>△ TUBE 10</td>
<td>284 426 503</td>
<td></td>
</tr>
</tbody>
</table>

Figure 6.21. Average condensation heat transfer coefficients for micro-fin Tube 10
tests were carried out at 50 psia. This is speculated to be a major factor contributing to the differences in the two different data sets.

A qualitative comparison of the present condensation heat transfer data indicates that the enhancement factors of Tojo et al. (1984) and Tatsumi et al. (1982) are within $\pm 20\%$ at low and medium mass velocities.

Conclusions

Increases in mass velocity increases heat transfer for both evaporation and condensation. The heat transfer coefficient was noted to be a strong function of mass velocity for micro-fin Tube 10 as compared to the smooth tube, specifically during evaporation heat transfer. The specific conclusions are:

1. Single-phase enhancement factors of 1.8-2.0 were recorded. The enhancement factors were comparable to those attained earlier (Chapters III through V) using micro-fin Tube 10.

2. Increases in the mass velocity decreases the evaporation enhancement factors. However, an insignificant effect of mass velocity on heat transfer enhancement factors was recorded for condensation.

3. The average evaporation heat transfer coefficients were comparable to the local values for the smooth tube. However, approximately 20-25\% lower average heat transfer coefficients were recorded for micro-fin Tube 10 when compared with the local values, specifically at low mass velocity. In general,
the average enhancement factors were lower than that of local values.

4. A qualitative comparison of average condensation enhancement factors using two different refrigerants (i.e., R-113 and R-22) indicated that the values for R-22 are approximately 40% lower. This could be mainly due to the different operating parameters of the two different test facilities.
CHAPTER VII. CONCLUSIONS AND RECOMMENDATIONS

Conclusions

The present study first investigated the effects of various internal geometrical parameters on single-phase, evaporation, and condensation heat transfer using R-113 as a test fluid. A smooth tube and nine micro-fin tubes of 3/8 in. (9.525 mm) O.D. were tested in a short water heated/cooled test section. A long smooth tube and one micro-fin tube were electrically heated and the local evaporation coefficients were compared with the short section average values. The experiments were extended to R-22 with an entirely new apparatus developed for this program. Single-phase and evaporation tests were carried out with electrically heated smooth and micro-fin tubes. These data were compared with the R-113 data. The final tests with R-22 involved the determination of average coefficients for essentially complete evaporation or condensation in a smooth tube and a micro-fin tube using a water heated or cooled test section.

**Single-phase study**

Heat transfer enhancement factors for the micro-fin tubes tested in the present investigation ranged from 1.30 to 2.0. The increase in heat transfer area and the flow separation over the spiral fins are considered to be major factors enhancing the heat transfer.
Evaporation study

In general, an increase in the mass velocity resulted in an increase in the heat transfer coefficient and pressure drop for both smooth and augmented tubes. Increases in quality resulted in higher average heat transfer coefficients using the short length test sections. However, the local evaporation heat transfer coefficients were essentially independent of quality. Heat flux increases the heat transfer coefficients in the nucleate boiling region, specifically at low mass velocities. However, at high mass velocity, increases in heat flux increases heat transfer coefficients for both nucleate boiling and forced convection region.

The smooth tube evaporation heat transfer data were within ± 30% of the predictions of Pujol and Stenning (1969), Shah (1982), and Kandlikar (1983). The best agreement was obtained with the prediction of Kandlikar (i.e., ± 20%).

Evaporation heat transfer enhancement factors were in the range of 1.3 to 2.6 for all nine augmented tubes and the maximum enhancement factor for pressure drop was 1.8. The increase in surface area of the micro-fin tubes is a major factor in enhancing the heat transfer coefficients and pressure drops. Several major conclusions are:

1. A geometrical parametric analysis indicates that micro-fin tubes having lower (0.0061 in. or 0.15 mm) but numerous (i.e., 70) fins, a flat or sharp peak, a round valley, and spiral angles greater than 10° should result in greater heat transfer performance. However, the effects of most of the geometrical parameters on pressure drop are insignificant. It should,
however, be noted that Tube 10 having a sharp peak, a flat valley with 60 fins that are 0.008 in. (0.20 mm) high, and a spiral angle of 16.5° resulted in the best heat transfer performance. This was accompanied by the maximum pressure drop increase.

2. Local and average heat transfer enhancement factors were comparable for micro-fin Tube 10 using R-113. Hence, reliable screening tests could be attained using short length test sections. However, approximately 20-25% higher pressure gradients were recorded using the longer test section.

3. In general, an increase in mass velocity resulted in a decrease in the enhancement factor for both R-113 and R-22 using micro-fin Tube 10. The local enhancement factors attained from R-22 for Tube 10 were approximately 30% lower than those of R-113, with the best agreement (approximate difference of 10%) occurring at high mass velocities. Considering the experimental uncertainties and the operating conditions of the two different refrigerants, it can be stated that the evaporation enhancement factors with R-113 as a test fluid could satisfactorily lead to first-hand information on the performance of the micro-fin tubes with R-22.

4. A qualitative comparison of the local and average smooth tube evaporation heat transfer coefficients using R-22 results in approximately 12-20% lower average values using the water-jacketed test section. For micro-fin Tube 10, approximately 25% lower average heat transfer coefficients were recorded.
Condensation study

For all cases, an increase in the mass velocity and quality results in an increase in the heat transfer and pressure drop. Heat transfer enhancement factors up to 3.83 were recorded and the maximum pressure drop enhancement factor was 2.0. The increase in surface area referenced to the smooth tube, the surface tension driven forces, and the liquid film disturbances are considered to be the important factors in enhancing the heat transfer.

Micro-fin tubes having greater fin height (0.007 in. or 0.18 mm), fewer fins (~60), and spiral angles in the range of 10-20° seem to result in the best performance. The conclusions derived from the geometrical parametric study coincided with most of the geometrical parameters of micro-fin Tube 10, which resulted in the best heat transfer performance. For example, Tube 10 has a flat valley, higher fins, fewer fins, and a spiral angle of 16.5°. In general, insignificant effects of most of the geometrical parameter on pressure drop were noted.

The condensation heat transfer enhancement factors for micro-fin Tube 10 using R-22 as a test fluid were lower than those for R-113. Based on a density-ratio-pressure curve for the two different refrigerants, it is speculated that condensation tests with R-113 as a test fluid at a higher system pressure (approximately 100 psia) would probably result in comparable enhancement factors.
Recommendations

1. Further study of the micro-fin tubes having recommended peak and valley shapes will provide an interesting and valuable extension that could yield optimum enhancement factors. Also, a systematic study of various spiral angles for tubes having the recommended fin profiles should be conducted.

2. It is evident from the present analysis that the fin geometry which results in good performance varies with mass velocity and quality. Hence, a new generation of micro-fin tubes with fin geometry as a function of tube length (or quality) could result in even greater overall heat transfer performance. It can also be speculated that these micro-fin tubes might not result in a significant increase in the pressure drop enhancement factor.

3. The present experimental data should be useful in developing a semi-empirical correlation of heat transfer coefficients for micro-fin tubes.

4. In reality, condensers and evaporators operate with approximately 2-10% oil mixed with the refrigerant. Therefore, evaporation and condensation enhancement factors for micro-fin tubes using oil-refrigerant mixtures should be studied to provide more realistic enhancement factors.

5. Commercial evaporators and condensers are normally built in the form of a serpentine coil; hence, the effects of bends and fittings should be investigated.

6. A generalized performance evaluation criteria (PEC) for ranking augmented tubes in two-phase flow has not been reported in the
literature. A generalized PEC would aid the efficient selection of an appropriate micro-fin tube. Using this PEC, tubes having relatively lower increases in the pressure drop with moderate heat transfer enhancement factors should be critically evaluated.

7. With minor modifications, the presently designed test apparatus would be capable of handling refrigerants such as R-11, R-12, and R-502. Therefore, the apparatus could be used to perform a systematic fluid-to-fluid modeling study.
REFERENCES


Altman, M.; Staub, F. W.; and Norris, R. H. 1960. "Local heat transfer and pressure drop for refrigerant 22, condensing in horizontal tubes." Chemical Engineering Progress Symposium Series 56: 151-159.


ACKNOWLEDGMENTS

I take this opportunity to express my sincere appreciation and deep sense of gratitude to my major professors Arthur E. Bergles and Michael B. Pate for their constant assistance, without which this study could not have been undertaken. To both of them, I owe much more than thanks. Their profound understanding and constant encouragement have been immensely responsible for bringing this work to fruition. A meeting with them in a trying moment has always delightfully resulted in not only clearing me out of confusion but also a renewed interest and enthusiasm at every stage. I am indebted to them for letting me resolve the difficulties of the experimentation; the well-known "Murphy's Law" was my true companion throughout the entire research work.

I also wish to thank Professor William J. Cook, Professor George H. Junkhan, Professor Bruce R. Munson, and Professor Dean L. Ulrichson, the members of my dissertation committee, for their assistance throughout my Ph.D. program.

For their work in the preparation of the dissertation manuscript, I would like to thank Miss Carla Holbrook for her endless hours of patient typing and the technical illustrators of the Engineering Research Institute for preparation of figures.

Special thanks are due John Heise, Paul Day, and Kyle Van Meter, student technicians, whose enormous amount of help, patience, and imagination made the extensive experimental program workable, especially in such a limited amount of time. I would like to acknowledge Mr.
Robert "Hap" Steed for providing equipment and technical assistance, and Mr. Gay Scandrett for teaching me the art of welding.

Very special thanks go to my parents, who have with great devotion, apart from sustaining me throughout many difficult times with care and affection, worked ceaselessly all these years to keep me out of the business of earning a living, and to provide me an opportunity for higher education.

I also wish to thank Arco Metals and American Brass for funding this research program. Their financial support and the helpful suggestions of Dr. Lyle MacAulay and Mr. Granville Ashley are appreciated.

In this short section, I could only touch on very few of the people who helped me throughout this research program; hence, many must go unnamed.
APPENDIX A. PARAMETRIC STUDY OF LOCAL EVAPORATION HEAT TRANSFER COEFFICIENTS

The effects of various parameters such as heat flux, quality, system pressure, and mass velocity on the evaporation heat transfer coefficient is discussed in this section. To achieve this objective, the correlations considered were: Pujol and Stenning (1969), Kandlikar (1983), Shah (1979), Dembi et al. (1978), and Lavin and Young (1965). Two different mass velocities, medium [475 kg/(m² s) or 350,000 lbm/(hr ft²)] and high [678 kg/(m² s) or lbm/(hr ft²)], and heat fluxes of 3517 Btu/(hr ft²) (12,000 W/m²) and 4689 Btu/(hr ft²) (16,000 W/m²) were considered for this analysis. In addition, two different system pressures were considered, specifically 100 psia (689 kPa) and 200 psia (1.38 kPa). Heat transfer coefficients at specific mass velocities, system pressures, and heat fluxes were computed using different correlations.

It is evident from the heat transfer coefficients plotted in Figs. A.1 through A.6 that the various correlations differ considerably. This is probably due to the fact that most of the correlations are valid only for a narrow range of experimental conditions and for specific fluids. Overall, it can be concluded that the heat transfer coefficient increases with mass velocity (Figs. A.1 and A.2). However, the dependence of heat transfer coefficient on quality is debatable. For example, sharp increases in heat transfer coefficient with quality are predicted for the Lavin and Young, and Dembi et al. correlations while the Kandlikar and Shah correlations suggest that heat transfer coefficient is not a strong function of quality.
Figure A.1. Effect of mass velocity on local evaporation heat transfer at constant (low) system pressure and constant heat flux.
Figure A.2. Effect of mass velocity on local evaporation heat transfer at constant (high) system pressure and constant heat flux

AVERAGE SYSTEM PRESSURE = 1.28 MPa
HEAT FLUX = 12000 W/m$^2$

- △ MASS VELOCITY = 475 kg/(m$^2$.s)
- ▲ MASS VELOCITY = 678 kg/(m$^2$.s)
- ▲ PUJOL AND STENNING CORRELATION
- □ KANDLIKAR CORRELATION (1983)
- ▼ DEMBI ET AL. CORRELATION (1978)
Figure A.3. Effect of heat flux on local evaporation heat transfer at a constant (low) mass velocity and constant average system pressure.
Figure A.4. Effect of heat flux on local evaporation heat transfer at a constant (high) mass velocity and constant average system pressure.
Figure A.5. Effect of system pressure on evaporation heat transfer at constant (low) mass velocity and constant heat flux.
Figure A.6. Effect of average system pressure on local evaporation heat transfer at constant (high) mass velocity and constant heat flux.
A close observation of Figs. A.1 and A.2 indicates that increases in heat transfer with mass velocity are somewhat greater at higher heat fluxes. For example, at higher heat fluxes an additional increase of approximately 10% in heat transfer coefficients was predicted by Shah for the same increase in mass velocity. A comparison of heat transfer coefficients using Figs. A.5 and A.6 reveals that heat transfer coefficients are relatively independent of system pressure. For example, doubling the system pressure (~ 200 psia or 1.38 MPa) results in the heat transfer coefficients changing by less than 20%.

In conclusion, an increase in the mass velocity results in an increase in the heat transfer coefficient. In addition, an increase in the heat flux results in an increase in the effect of mass velocity on heat transfer. Finally, the effect of system pressure on heat transfer is small.
APPENDIX B. DETAILS OF EXPERIMENTAL LOOP (R-113) COMPONENTS

The experimental facility for testing tubes using short and long test sections was described earlier in Chapters III and IV, respectively. A detailed description of the equipment used in the apparatus is contained in this section. Reasons for selecting R-113 for experimentation were described in detail by Luu (1979).

Pump

A positive displacement type gear pump with a mechanical seal and carbon bearings was used for pumping the test fluid. The pump relied on lubrication from the refrigerant so as to avoid oil contamination of the apparatus. The lip seal of the pump had to be replaced after about 100 hours operation of the pump because of the poor lubration of the R-113. A 1/4 hp motor running at 1140 rpm pumped the test fluid at a differential pressure of approximately 100 psia (6.89 kPa) and a maximum flow rate of 1.5 gpm. A 1/2 in. (12.7 mm) Henry relief valve having a relief pressure setting of 150 psi was installed near the pump outlet. A Sporlan model C-414 filter-dryer unit installed downstream of the pump was used to remove any contamination particles and moisture present in the test fluid.

Degassing Tank

The presence of noncondensable gases in the R-113 refrigerant can lead to experimental errors (Luu, 1979). Hence, a standard model UR-66 liquid receiver was modified and used as the degassing tank.
The tank was mounted at a height of 4.9 ft (1.5 m) from the ground level. This suppressed cavitation at the pump inlet. The tank was equipped with a 587 Btu/hr (2 kW) immersion heater for heating the contents during the degassing procedure. At the top of the tank, a small condenser was installed to condense the vapors evaporated in the tank during the same degassing process. A sight glass, made up of Tygon tube, was installed on the side of the tank for the purpose of indicating the liquid level in the tank.

Pre-heater/Pre-evaporator Tube

A 0.44 in. (11.18 mm) O.D. by 8.2 ft (2.5 mm) long stainless steel 304 tube with a wall thickness of 0.054 in. (1.37 mm) was used as a preheater. This tube was electrically heated using direct current. A 29981 Btu/hr (75 kW) capacity American Rectifier Corporation Model-SIMSASF611225E Rectifier/Transformer unit supplied the required power to the preheater tube. The maximum output for the transformer unit was attained at 1225 amps and 61 volts. The input power to the preheater was controlled by a remote control box. Two 3/0 cables having a maximum current capacity of 200 amps conducted the current from the rectifier unit to the preheater tube.

After-condenser/Condenser

An after-condenser was installed to condense and cool the refrigerant exiting from the test section before it entered the pump. The condenser was oversized for the present application, hence, it was difficult to obtain a reasonably good heat balance for the entire loop.
The condenser was a conventional four pass, shell-and-tube heat exchanger operated with the test fluid on the tube side. The tubes were of admiralty metal and provided $12.5 \text{ ft}^2 (181.50 \text{ m}^2)$ of heat transfer surface. Thermocouples were placed at the inlet and outlet of the condenser on both the tube (test fluid) and shell (coolant) sides.

**Annulus Side Heat Exchanger**

The heated water exiting from the annulus part of the test section was cooled by a shell-and-tube type heat exchanger (Model BCF-BCC11G3) having a surface area of approximately $4.3 \text{ ft}^2 (62.4 \text{ m}^2)$. Cooling water from the building mains was supplied on the shell side of the heat exchanger.

**Recirculating Pump for Water Loop**

A centrifugal pump having a differential pressure of 10 psi and a maximum flow rate of 1 gpm was used for recirculating the water in the water flow loop. A March pump (Model #809 HS) was selected due to its capability of performing well at higher fluid temperatures. Specifically, the special plastic impeller performed well at higher pressures (~ 50 psia or 344.9 kPa) and higher temperatures (~ 220°F or 104.3°C). A Filterite filter unit (Model LM04B-3/8) was installed at the outlet of the pump for the removal of foreign particles from the water.

**Accumulator**

A Greerolator accumulator (Model #20-25OTMR-S3/4) with a 1 gallon capacity and a Neoprene bladder were installed at the outlet of the pump.
in the water circuit. The accumulator maintained system pressure at a level sufficient to avoid boiling of water at 210°F (99°C) and also dampened flow fluctuations.

Power Control Unit for Boiler

A 3282 Btu/hr (11.2 kW) capacity boiler installed in the test fluid loop was used. To attain the desired degree of subcooling of the fluid entering the test section, the power supplied to the boiler was controlled by a four-gang General Electric auto-transformer unit. Six heaters were installed in the boiler so as to form three heater groups having two heaters each. The boiler control panel was capable of eliminating the power supplied to any of these three groups. The auto-transformer, having a maximum current capacity of 44 amps (ac) at 220 V (ac), controlled the degree of subcooling at the inlet of the test section. It should be noted that one single autotransformer controlled the electrical power supplied to the boiler. Panel meters installed in the control panel indicated the voltage and current supplied to the unit.

Instrumentation

Data acquisition system

The data acquisition system consisted of a Hewlett-Packard Model 9825A computer, a Hewlett-Packard Model 3421A scanner plus voltmeter unit, a Hewlett-Packard Model 3455A voltmeter, two Hewlett-Packard Model 3425A digital multimeters, and a Kaye instruments Model K170-36C ice-point reference. A selector switch was installed in between the
ice-point reference and scanner for local evaporation tests. This was necessary due to a limited availability of scanner channels in the existing scanner unit.

The operation of the data acquisition system was previously described in detail by Jensen (1976) and Luu (1979), who successfully used the system for data collection and analysis. Computer software for single-phase, condensation, evaporation, local single-phase, and local evaporation heat transfer was developed and reported in Appendix F. The refrigerant property subroutines used in these programs were described by Jensen (1976).

**Temperature**

A total of 23 thermocouples for the short test section and 54 thermocouples for the long test section was used for collecting the experimental data. All thermocouples from the test rig were directed to a central switch board. These thermocouple wires were then connected to a selector switch having a total of 56 channels. Finally, these thermocouples were read using a voltmeter via a reference ice-junction unit and a scanner unit.

Temperature measurements were generally carried out using Duplex TT-T-30 copper-constantan thermocouple wires from Omega. However, 36-gage copper-constantan wires were used to measure the water side bulk temperature. Limitations of different gage wires in terms of accuracy are reported in the Omega Handbook (1985).
Detailed subroutines for converting the millivolts to temperature were reported by Luu (1979). In the present data acquisition program, this subroutine is designated as "TEMP".

**Pressure**

The absolute (static) pressure measurements were taken using two Heise Bourdon type pressure gages having \( \pm 1/4\% \) of full scale accuracy. It was necessary to use two gages in parallel due to the sensitivity of the pressure measurements. While reducing the experimental data, it was noted that a pressure difference of 0.725 psia (5.0 kPa) resulted in 1\(^\circ\)F (0.6\(^\circ\)C) difference in the saturation temperature at a system pressure of 35 psia (241 kPa). Since a very small temperature difference (~ 3-5\(^\circ\)F or 1.7-2.8\(^\circ\)C) existed between the test tube wall and the fluid for most of the two-phase experimentation, an accurate pressure measurement was necessary. Both pressure gages were calibrated using a dead weight pressure gage tester manufactured by Amther. Calibration equations were then obtained for both pressure gages using a "A CALIBRATION" program developed for a Hewlett-Packard 9845A computer.

A Meriam differential manometer (Model #A-203) having a resolution of 0.25 in. (6.35 mm) of Hg was used to measure differential pressures. It is important to note that only the static pressure entering the test section and the differential pressure across the test section were measured. The test fluid pressures exiting the test section were thus evaluated by subtracting the pressure drop from the inlet pressure. In order to remove air from pressure lines and other parts of the system, a vacuum of about 27 in. (6.86 m) of mercury was pulled for at
least 3 hours. This procedure reduced the possibility of sustaining trapped air in the refrigerant lines leading to the pressure measuring instruments. A bleed valve and a nonreturn valve provided at the top of the differential manometer helped in bleeding air bubbles in the lines connecting the differential manometer to the test section. The procedure for air removal was tedious but efficient.

Measuring pressure drops at low mass flow rates was difficult due to the resolution of the differential manometer relative to the pressure drop occurring over the rather short length test section. Also, it was not possible to replace the manometer fluid of mercury with another having a smaller specific gravity since most of these fluids were not compatible with R-113.

The above factors plus fluctuations in the pressure measurements caused a wide scatter in the pressure drop data at low mass flow rates. Using a manometer with better a resolution or else an inclined manometer is recommended for improved accuracy in the pressure drop measurements. A pressure transducer with good accuracy and response characteristics would be even better.

Flow measurements

Rotameters were used for measuring the flow rates of the R-113 test fluid, water in the annulus, and the water flowing through the after-condenser. These flowmeters were calibrated using a tank, a scale, and a stop watch. Calibration equations were then fitted using software developed on a Hewlett-Packard 9845A computer. The details of these flowmeters along with their calibration curves are as follows:
1. Test fluid (R-113) flowmeter
   Brooks Rotameter, Type 1114
   Tube number: R-8M-25-2
   Float number: 8-RV-3, stainless steel
   Range: 0-0.52 GPM
   \[ m = 2.79423 \, M - 20.35313 \]
   \( m \) in lbm/hr, \( M \) in percent

2. Water (annulus-side) flowmeter
   Brooks Rotameter, Type 1110
   Tube number: R-8M-25-4
   Float number: 8-RV-3
   Range: 0.14-1.4 GPM
   \[ m = (0.9899 \, M - 0.0002) \]
   \( m \) in GPM, \( M \) in GPM

3. After-condenser coolant (water) flowmeter
   Brooks Rotameter, Type 1110
   Tube number: R-10M-25-2
   Float number: 10-RS-64
   Range: 0-6.40 GPM
   \[ m = 12.76958 \, m - 10.730 \]
   \( m \) in lbm/hr, \( M \) in mm

The calibration was carried out at a constant temperature of 70°F
(21°C), hence, a temperature correction factor had to be incorporated
into the above equations. As suggested in the Brooks Catalog (1985),
the factor can be written as
\[ C_1 = \eta \left( \frac{8.04 - \eta}{7.04 \eta} \right)^{1/2} \]  

(B.1)

where \( \eta = \frac{\rho''}{\rho'} \)

\( \rho' \) = density of the calibrated fluid

\( \rho'' \) = density of the metered fluid

All of these equations were incorporated into the data acquisition program.
APPENDIX C. EXPERIMENTAL PROCEDURE

The initial testing, consistency, repeatability, and stability of the test apparatus described in Chapters III and IV are discussed in this section. In addition, the procedure for the removal of noncondensable gases from R-113 is also described.

Initial Testing of R-113 Test Rig

The test fluid flow loop was pressure tested for leaks using nitrogen gas at approximately 100 psia (689 kPa) prior to its initial operation. These leaks were identified using a soapy water solution. Similar pressure tests at 50 psia (345 kPa) were also conducted for the water flow loop. After fixing leaks, both rigs were thoroughly flushed using fluids. The freon loop was then evacuated for at least 5 hours for the removal of air.

The test rig was designed so that the test section and preheater tube could be isolated from the rest of the system. Hence, only the portion of the rig involving these two components was pressure tested and evacuated whenever an installation of new test tube was executed. It should be noted that leaks in the test fluid loop primarily occurred at the pressure tap connections; they were fixed by tightening or resoldering. In contrast, leaks in the water flow loop occurred through the Teflon insulation surrounding the thermocouples installed on the tube wall. The removal of the outermost layer of Teflon insulation from the thermocouples (Omega TT-T-30) eliminated these leaks. Minor leaks
in the water flow loop were then fixed by applying Devcon 5-minute epoxy or Dow Corning Silastic 732 RTV sealant at the appropriate locations.

Removal of Noncondensable Gases

Luu (1979) reported in detail the importance of the removal of noncondensable gases from R-113. An experiment conducted by Luu resulted in 0.36 cc of air being dissolved in each cc of the test fluid at 86°F (30°C) and atmospheric pressure. This results in potential difficulties in using pure refrigerant properties for heat transfer data evaluation. The air could also seriously affect condensation, single-phase, and, to a lesser extent, the evaporation tests.

Air removal from the test fluid was accomplished by boiling the fluid in the degassing tank at atmospheric pressure. The R-113 vapor, being heavier than air, stratified in the vapor phase. The gas mixture (i.e., test fluid vapor and air) exiting from the degassing tank then passed through the degassing condenser. Most of the test fluid was condensed and thus recovered in this degassing condenser while the air was discharged from the system. The detailed procedure for the removal of air was as follows:

1) Cooling water was circulated through the after-condenser and the degassing condenser.

2) The valve at the top of the degassing tank was fully opened.

3) Both the test fluid throttle valve and the inlet valve were fully opened.
4) The valve bypassing the degassing tank was fully closed whereas the valve connecting the tank was fully opened.

5) The pump was then turned on. The test fluid was circulated through the drier and filter unit.

6) An autotransformer controlled the power input to the degassing tank. It should be noted that the tank had a maximum heating capacity of 586.2 Btu/hr (2 kW).

7) During the initial hour of the degassing, the temperature of the test fluid in the degassing tank was maintained several degrees below the saturation temperature at atmospheric pressure. Thereafter, the tank temperature was maintained at or near the saturation temperature. This process was continued for two to three hours.

Flow Stability

The stability of the system was attained by controlling two flow regulating valves and one fluid throttle valve provided in the refrigerant flow loop. Variations of the coolant and refrigerant mass flow rates, heat transfer rates, and the dc power supply to the preheater tube sometimes resulted in sudden transients in the system. However, the original system pressures were restored in a relatively short period of time.

Pressure, inlet quality, and mass flow rate fluctuations were noted during normal operation due to the nature of two-phase flow. However, because of their short time duration these variations did not have a significant effect on heat transfer measurements. In addition, the
fluctuations in the inlet pressure were not significant. For example, a maximum variation of 0.2 psia (1.37 kPa) in the absolute pressure was observed. Pressure drop fluctuations due primarily to variations in mass flow rates were also insignificant. Specifically, the installation of the accumulator in the water flow loop resulted in a steady mass flow rate of water on the annulus side of the test section. A typical fluctuation in the pressure drop was approximately 0.01 in. (0.25 mm) of Hg. In summary, instabilities were controlled so that the measurement of reliable data was possible.

Consistency and Repeatability

The characteristic of two-phase flow, the test fluid flow fluctuations, and the system pressure fluctuations, etc., collectively resulted in a great deal of concern about consistency and repeatability of the experimental data. Evaluating the repeatability of data was difficult because it is impossible to duplicate fluid mass flow rates, system pressures, and inlet/exit qualities from a previous test run. For example, at a constant test fluid mass flow rate and a steady system pressure, the exit quality of freon varied with fluctuations in the electrical (dc) power supplied to the preheater. This, in turn, varied the quality of the test fluid entering the test section. In addition, the change in the quality along the test section depended upon the temperature and the mass flow rates of the annulus side water which were also difficult to control exactly. Therefore, even if conditions on the freon (test fluid) side were duplicated, the average quality and change in quality over the test section might not have been truly repeated.
Figure C.1. Comparison of single-phase heat transfer coefficients with Dittus-Boelter/McAdams correlation
Figure C.2. Wilson plot for calculating refrigerant side heat transfer coefficient
AVERAGE SYSTEM PRESSURE = 315-330 kN/m²

MASS VELOCITY kg/(m²·s)

△LOW ○MEDIUM □HIGH

SMOOTH TUBE

(Repeat) 218 367 526

SMOOTH TUBE

237 354 559

Figure C.3. Repeatability test for evaporation heat transfer coefficients with the smooth tube.
Figure C.4. Repeatability test for condensation heat transfer coefficients with the smooth tube.
AVERAGE SYSTEM PRESSURE = 315-330 kN/m²

MASS VELOCITY kg/(m²·s)

- LOW
- MEDIUM
- HIGH

SMOOTH TUBE
218 367 526

(REPEAT)
237 354 559

Figure C.5. Repeatability test for evaporation pressure drop with the smooth tube
AVERAGE SYSTEM PRESSURE = 310-352 kN/m²

MASS VELOCITY kg/(m².s)

- SMOOTH TUBE LOW MEDIUM HIGH
- SMOOTH TUBE 197 390 565
- (REPEAT) 242 355 561

Figure C.6. Repeatability test for condensation pressure drop with the smooth tube
Figure C.7. Repeatability test for single-phase heat transfer with micro-fin Tube 9
Figure C.8. Repeatability test for condensation heat transfer coefficients with micro-fin Tube 9
Figure C.9. Repeatability test for evaporation heat transfer coefficients with micro-fin Tube 9
Figure C.10. Repeatability test for evaporation pressure drop with micro-fin Tube 9
AVERAGE SYSTEM PRESSURE = 321-337 kN/m²
MASS VELOCITY kg/(m²·s)

LOW MEDIUM HIGH

TUBE 9 220 361 576
TUBE 9 224 357 575
(REPEAT)

Figure C.11. Repeatability test for condensation pressure drop using micro-fin Tube 9
Hence, a complete replication of a test run was a very difficult task. Nevertheless, heat transfer and pressure drops were compared for a smooth tube and micro-finned Tube 2 at quite similar test fluid mass flow rates, pressures, and qualities. The two-phase data runs were carried out at three different mass velocities. Repeatability test runs for single-phase flow were also performed over a Reynolds numbers range of 4,000 to 11,000.

A smooth tube repeatability test indicated excellent agreement between two data sets (Fig. C.1). An additional repeatability test was performed at a Reynolds number of 7,025 using the Wilson plot technique (McAdams, 1942). The single-phase heat transfer coefficient results obtained by the conventional method and the Wilson plot technique were within 2.5% (Fig. C.2).

Figures C.3 and C.4 show good agreement between two different data sets for two-phase heat transfer. However, repeat evaporation pressure drop data shown in Fig. C.5 were lower by approximately 25% at medium mass velocities and 10% at high mass velocities. However, excellent agreement of the evaporation pressure drop data was observed at low mass velocities. During condensation tests, a maximum disagreement of 10% was observed for the pressure drop data (Fig. C.6). In addition to the smooth tube, a repeatability test was also conducted using micro-finned Tube 9. Figures C.7 through C.11 confirm that the experimental data were consistent for both heat transfer and pressure drop.

In conclusion, the repeatability test results were within about ±10%. Most of the variations in the data could possibly be due to differences in the system pressures, mass velocities, and heat fluxes.
APPENDIX D. SAMPLE CALCULATIONS AND ERROR ANALYSIS FOR TEST APPARATUS USING R-113

This section describes the sample calculations and the propagation of error for determining the average and the local heat transfer coefficients using the test apparatus described in Chapters III and IV. As mentioned earlier, the experimental data are recorded in the Heat Transfer Laboratory of the Department of Mechanical Engineering at Iowa State University.

Test Apparatus for Average Heat Transfer Coefficients

Sample calculation

To illustrate the data reduction procedure, a sample calculation for the smooth tube is given. The data run 1.E.4 at medium mass velocity is reported herein.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube I.D., $D_i$</td>
<td>0.343 in.</td>
</tr>
<tr>
<td>Tube O.D., $D_o$</td>
<td>0.375 in.</td>
</tr>
<tr>
<td>Tube length, $L$</td>
<td>39.0 in.</td>
</tr>
<tr>
<td>Preheater shunt voltage</td>
<td>58.0 mV</td>
</tr>
<tr>
<td>Preheater terminal voltage</td>
<td>17.84 V</td>
</tr>
<tr>
<td>Test fluid flow rate</td>
<td>62.2%</td>
</tr>
<tr>
<td>Shell side water flow rate</td>
<td>0.85 gpm</td>
</tr>
<tr>
<td>After-condenser coolant flow rate</td>
<td>22.0 mm</td>
</tr>
<tr>
<td>Ambient pressure</td>
<td>14.38 psig</td>
</tr>
<tr>
<td>System pressure</td>
<td>36.50 psig</td>
</tr>
</tbody>
</table>
Pressure drop across the test section 2.60 in. Hg
Test fluid preheater inlet temperature 92.13°F
Saturation temperature of test fluid at average test-section pressure 190.78°F
Test fluid temperature at the flow meter 85.33°F
Annulus side water average inlet temperature 209.01°F
Annulus side water average outlet temperature 204.97°F

Wall temperatures
\[ T_1 = 200.50°F \]
\[ T_2 = 198.15°F \]
\[ T_3 = 198.78°F \]
\[ T_4 = 198.58°F \]
\[ T_5 = 199.22°F \]
\[ T_6 = 200.16°F \]
\[ T_7 = 199.90°F \]

Test fluid mass flow rate:

From the calibration equation, \( m_F = 154.19 \text{ lbm/hr} \), and applying the temperature correction

\[
C_1 = \eta \left( \frac{8.04 - \eta}{7.04 \eta} \right)^{0.5} = 1.15
\]

\[
m_F = 1.15 \times 154.19
\]

\[ = 177.47 \text{ lbm/hr} \]
Test-section average pressure:

Inlet pressure = 36.50 + 14.38 = 50.88 psia

\[ P_{av} = P_{in} + \frac{\Delta P}{2} \]

\[ = 50.88 - \frac{2.60}{2} \]

\[ = 49.58 \text{ psia} \]

Input power to preheater:

\[ Q_{pre} = 0.98 \times V_T \times I_s \]

where

\[ I_s = \text{shunt voltage (mv)} \times 2.4 \]

and 0.98 accounts for the estimated 2% heat loss in the preheater.

\[ Q_{pre} = 0.98 \times 17.84 \times (58.0 \times 2.4) \times 3.412 \]

\[ = 8303.65 \text{ Btu/hr} \]

Preheater outlet quality:

\[ Q_{pre} = Q_{sens} + Q_{latent} \]
where

\[ Q_{\text{sens}} = m_F \, C_F \, (T_{\text{sat}} - T_{\text{pre, in}}) \]

\[ = 177.47 \times 0.25 \times (190.78 - 92.13) \]

\[ = 4376.86 \text{ Btu/hr} \]

\[ Q_{\text{latent}} = Q_{\text{pre}} - Q_{\text{sens}} \]

\[ = 8210.56 - 4376.86 \]

\[ = 3926.79 \text{ Btu/hr} \]

\[ Q_{\text{latent}} = m_F \, i_{fg} \, \Delta X \]

\[ i_{fg} = 56.58 \text{ Btu/lbm} \]

\[ \Delta X = \frac{Q_{\text{latent}}}{m_F \, i_{fg}} = \frac{X_{\text{pre, out}}}{56.58 \times 177.47} \]

\[ = 0.39 \]

Also

\[ X_{\text{pre, out}} = X_{\text{test, in}} \]
Annulus side water mass flow rate:

Using the calibration equation,

\[ m_w = 420.98 \text{ lbm/hr} \]

**Water side heat transfer:**

\[ Q_w = m_w C_w (T_{w_{in}} - T_{w_{out}}) = 420.98 \times 1.0 \times (209.01 - 204.97) \]

\[ = 1700.75 \text{ Btu/hr} \]

**Change of quality in the test section:**

\[ \Delta X = \frac{Q_w}{m_p i_{fg}} \]

\[ = \frac{1700.75}{177.47 \times 56.58} = 0.17 \]

For evaporation

\[ X_{test_{out}} = X_{test_{in}} + \Delta X \]

\[ = 0.37 + 0.17 = 0.58 \]

Also
\[ X_{av} = X_{in} + \Delta X/2 \]
\[ = 0.39 + 0.17/2 \]
\[ = 0.48 \]

Heat transfer coefficient calculations:

\[ h_i = \frac{q_w}{A_{sur}} (T_{wav} - T_{sat}) \] (3.11)

where

\[ T_{wav} = \left[ T_1 + T_2 + \left( \frac{T_3 + T_4 + T_5}{3} \right) + T_6 + T_7 \right] / 5 \]
\[ = \left( 200.50 + 198.15 + \frac{198.78 + 198.58 + 199.22}{3} \right) \]
\[ + 200.16 + 199.00 \right] / 5 \]
\[ = 199.25^\circ F \]

\[ h_i = \frac{1700.75}{(199.35 - 190.78) \times 0.292} \]
\[ = 674.41 \text{ Btu/(hr ft}^2 \text{ °F)} \]

**Propagation of error**

The following section follows the procedure recommended by Kline and McClintock (1953) to evaluate the experimental uncertainty for a single-sample experiment. This procedure is termed propagation of error.
and has been followed for a typical run (Run 1.E.4) involving the smooth tube. It should be noted that the error reported in this section is the absolute value of the maximum expected deviation.

From Eq. (3.11), the average heat transfer coefficient is

\[
h = \frac{Q_w}{A_{\text{sur}}(T_{W_{av}} - T_{\text{sat}})} \tag{3.11}
\]

The uncertainty in evaluation \( h \) is estimated as follows:

\[
V_h = \left[ \left( \frac{\partial h}{\partial Q} \right)^2 + \left( \frac{\partial h}{\partial A} \right)^2 + \left( \frac{\partial h}{\partial T_{W_{av}}} \right)^2 + \left( \frac{\partial h}{\partial T_{S}} \right)^2 \right]^{1/2} \tag{D.1}
\]

where

\[
\frac{\partial h}{\partial Q} = \frac{1}{A(T_{W_{av}} - T_{S})} \tag{D.2}
\]

\[
\frac{\partial h}{\partial A} = -\frac{Q}{A^2(T_{W_{av}} - T_{S})} \tag{D.3}
\]

\[
\frac{\partial h}{\partial T_{S}} = \frac{Q}{A(T_{W_{av}} - T_{S})^2} \tag{D.4}
\]

\[
\frac{\partial h}{\partial T_{W_{av}}} = -\frac{Q}{A(T_{W_{av}} - T_{S})^2} \tag{D.5}
\]
For convenience, $Q_y$ and $A_{sur}$ are referred to as $Q$ and $A$, respectively. Using the experimental conditions for a particular data run

$$A = nD_1L = 0.2918 \text{ ft}^2$$

$$T_s = 190.81^\circ F$$

$$T_{av} = 199.23^\circ F$$

$$Q = 1700.72 \text{ Btu/hr}$$

Thus,

$$\frac{\partial h}{\partial Q} = \frac{1}{(0.2918)(8.64)} = 0.3966 \frac{1}{(\text{ft}^2 \circ F)}$$

$$\frac{\partial h}{\partial A} = \frac{1700.72}{(0.2918)^2(8.64)} = 2311.79 \frac{\text{Btu}}{(\text{hr ft}^4 \circ F)}$$

$$\frac{\partial h}{\partial T_s} = \frac{1700.72}{(0.2918)^2(8.64)^2} = 78.88 \frac{\text{Btu}}{(\text{hr ft}^2 \circ F^2)}$$

$$\frac{\partial h}{\partial T_{av}} = -\frac{1700.72}{(0.2918)^2(8.64)^2} = -73.08 \frac{\text{Btu}}{(\text{hr ft}^2 \circ F^2)}$$

The uncertainties $W_Q$, $W_A$, $W_{T_s}$ and $W_{T_{av}}$ are estimated as in Eq. (D.1).
Estimate of $q_w$

\[ Q_w = m_w C_w (T_{w, in} - T_{w, out}) \]  \hspace{1cm} (D.6)

Thus,

\[ q_w = \left[ \left( \frac{\partial Q}{\partial m_w} \right) m_w \right]^2 + \left( \frac{\partial Q}{\partial C_w} C_w \right)^2 + \left( \frac{\partial Q}{\partial T_{w, in}} T_{w, in} \right)^2 \]
\[ + \left( \frac{\partial Q}{\partial T_{w, out}} T_{w, out} \right)^2 \]^{1/2} + heat exchange with environment \hspace{1cm} (D.7)

with

\[ m_w = 420.98 \text{ lbm/hr} \]
\[ C_w = 1.0 \text{ Btu/lbm °F} \]
\[ T_{w, in} = 209.01 \text{ °F} \]
\[ T_{w, out} = 204.97 \text{ °F} \]
\[ \Delta T_w = 4.04 \text{ °F} \]

Also,

\[ \frac{\partial Q}{\partial m_w} = C \Delta T_w = 4.04 \text{ Btu/lbm} \]  \hspace{1cm} (D.8)
The calibrated rotameter results in an uncertainty of 0.025 gpm, i.e., 12.5 lbm/hr.

The uncertainty \( W_c = 0.004 \text{ Btu/(lbm }^\circ\text{F}) \)

Also the uncertainty \( W_T = 0.15^\circ\text{F} \)

Hence,

\[
W_q = [(4.04 \times 12.5)^2 + (1700.76 \times 0.004)^2 + (420.98 \times 0.15)^2 \\
+ (420.98 \times 0.15)^2]^{1/2} + 0.02 \times 1700.75 = 145.10 \text{ Btu/hr}
\]

The last term in the above-mentioned equation represents 2% heat loss from the shell side heat transfer fluid. It is expected that this overestimates the heat loss since the test section was very well insulated using Armaflex insulation.

Estimate of \( W_A \)

\[
A = \pi D_L
\]  
(D.11)

Thus,
\[ W_A = \left( \frac{2A}{2D_i} W_{D_i} \right)^2 + \left( \frac{2A}{2L} W_L \right)^2 \right]^{1/2} \\
= \left( (\pi L V_{D_i})^2 + (\pi D_i V_{L})^2 \right)^{1/2} \quad (D.12) \\

\text{with} \\
D_i = 0.0285 \text{ ft} \\
L = 3.25 \text{ ft} \\
W_{D_i} = 0.001 \text{ ft} \\
W_L = 0.01 \text{ ft} \\
W_A = \left( (\pi \times 3.25 \times 0.001)^2 + (\pi \times 0.0285 \times 0.01)^2 \right)^{1/2} \\
= 0.0102 \text{ ft}^2 \\

\text{Estimate of} \ W_T^{T_s} \\

The fluid saturation pressure was obtained from the linear average of inlet and outlet static pressures of the test section.

\[ T_s = T(P_s) \quad (D.13) \]

Hence,
The uncertainty, $W_p$, can be obtained from the following equations:

\[
W_{p_s} = \left[ \left( \frac{\partial T}{\partial P} W_{p_s} \right)^2 \right]^{1/2} + W_{p-T}
\]  
\text{(D.14)}

\[
W_p = \left[ \left( \frac{P_s}{P_{in}} W_{p_{in}} \right)^2 + \left( \frac{P_s}{\Delta P} W_{\Delta P} \right)^2 \right]^{1/2} = \left( W_p^2 + \frac{1}{4} W_{\Delta P^2} \right)^{1/2}
\]  
\text{(D.16)}

Hence,

\[
W_{p_s} = \frac{(P_{in} + P_{out})}{2} = \frac{[P_{in} + (P_{in} + \Delta P)]}{2} = P_{in} + \Delta P/2
\]  
\text{(D.15)}

The partial derivative of $P_{in}$ or $\Delta P$ with respect to $P_s$ is assumed to be unity. Considering

- uncertainty in the pressure gage ± 0.25 psi
- uncertainty in the manometer ± 0.1 psi

\[
W_{p_s} = (0.25^2 + 1/4 (0.1)^2)^{1/2}
\]

= 0.255 psia

Using Luu (1979), $W_{p-T}$ for R-113 is 0.006°F with $dT/dP_s = 0.9$ from the thermodynamic property formula; thus,
\[ W_{T_s} = 0.9 \times 0.255 + 0.06 \]

\[ = 0.3^\circ F \]

Estimate of \( W_{Tw} \) and \( W_{T_{av}} \)

The uncertainty due to irregularities in the thermocouple wire was estimated to be 0.15°F. Using Eq. (D.1)

\[ W_h = [(0.3966 \times 145.10)^2 + (78.88 \times 0.15)^2] + (2311.99 \times 0.0102)^2 + (78.88 \times 0.3)^2 \]

\[ = 674.41 \pm 67.58 \text{ Btu/(hr ft}^2\text{ °F) \]

Thus, the uncertainty for a typical experimental run is approximately ± 10%.

Test Apparatus for Local Heat Transfer Coefficients

Sample calculation

The detailed data reduction procedure was outlined earlier in Chapter IV, however, the sample calculation for Run 1.EL.2 is reported here. The following experimental data were noted for the smooth tube at medium mass velocity.
<table>
<thead>
<tr>
<th>Location</th>
<th>Wall Temperature, °R</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>658.25</td>
</tr>
<tr>
<td>2</td>
<td>659.15</td>
</tr>
<tr>
<td>3</td>
<td>660.65</td>
</tr>
<tr>
<td>4</td>
<td>663.90</td>
</tr>
<tr>
<td>5</td>
<td>663.10</td>
</tr>
<tr>
<td>6</td>
<td>662.68</td>
</tr>
<tr>
<td>7</td>
<td>664.37</td>
</tr>
<tr>
<td>8</td>
<td>662.82</td>
</tr>
<tr>
<td>9</td>
<td>662.03</td>
</tr>
<tr>
<td>10</td>
<td>660.79</td>
</tr>
</tbody>
</table>
Test-section location pressure

Inlet pressure = 34.18 + 14.28 = 48.46 psia considering inlet pressure and sectional differential pressure, the pressures at locations 1, 4, 7, 9, and 13 (Fig. 4.3) can be stated as

\[ P_1 = \text{inlet pressure} = 48.46 \text{ psia} \]

\[ P_4 = P_1 - \Delta P_1 = 48.29 \text{ psia} \]

\[ P_7 = P_4 - \Delta P_2 = 48.02 \text{ psia} \]

\[ P_{10} = P_7 - \Delta P_3 = 47.12 \text{ psia} \]

\[ P_{13} = P_{10} - \Delta P_4 = 45.59 \text{ psia} \]

Using a linear distribution of the pressure between two consecutive pressure taps, the local pressures at all thirteen locations can be evaluated and are reported as
Saturation Change in Wall Temperature

<table>
<thead>
<tr>
<th>Location</th>
<th>Pressure psia</th>
<th>Saturation Temperature $T_R$</th>
<th>Change in Wall Temperature $T_R$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>48.46</td>
<td>651.01</td>
<td>7.23</td>
</tr>
<tr>
<td>2</td>
<td>48.41</td>
<td>650.93</td>
<td>8.21</td>
</tr>
<tr>
<td>3</td>
<td>48.35</td>
<td>650.85</td>
<td>9.81</td>
</tr>
<tr>
<td>4</td>
<td>48.29</td>
<td>650.76</td>
<td>13.13</td>
</tr>
<tr>
<td>5</td>
<td>48.20</td>
<td>650.63</td>
<td>12.47</td>
</tr>
<tr>
<td>6</td>
<td>48.11</td>
<td>650.49</td>
<td>12.19</td>
</tr>
<tr>
<td>7</td>
<td>48.02</td>
<td>650.36</td>
<td>14.00</td>
</tr>
<tr>
<td>8</td>
<td>47.72</td>
<td>649.91</td>
<td>12.92</td>
</tr>
<tr>
<td>9</td>
<td>47.42</td>
<td>649.45</td>
<td>12.58</td>
</tr>
<tr>
<td>10</td>
<td>47.12</td>
<td>648.99</td>
<td>11.80</td>
</tr>
<tr>
<td>11</td>
<td>46.61</td>
<td>648.21</td>
<td>11.30</td>
</tr>
<tr>
<td>12</td>
<td>46.10</td>
<td>647.43</td>
<td>10.88</td>
</tr>
<tr>
<td>13</td>
<td>45.59</td>
<td>646.64</td>
<td>8.60</td>
</tr>
</tbody>
</table>

Quality and heat flux calculations:

The heat transfer to the test section is

$$Q_{test} = V_T I_S$$  \hspace{1cm} (D.17)

where

$$I_S = 30 \times \text{shunt voltage (mV)}$$

$$Q_{test} = 4.68 \times (18.11 \times 30) \times 3.412 = 8675.50 \text{ Btu/hr}$$
The sensible heat required to attain the saturated temperature of fluid is,

\[ Q_{\text{sens}} = m_F C_F (T_{\text{sat}} - T_{\text{in}}) \]

\[ = 184.82 \times 0.25 (32.08) \]

\[ = 1482.25 \text{ Btu/hr} \]

\[ Q_{\text{latent}} = Q_{\text{test}} - Q_{\text{sens}} = 7193.25 \text{ Btu/hr} \]

heat flux \[ q_{\text{test}} = \frac{Q_{\text{test}}}{m D_i L_{\text{test}}} \quad \text{(D.18)} \]

\[ q_{\text{test}} = 7790.29 \text{ Btu/(hr ft}^2) \]

Considering Eq. (4.10) and the enthalpy of vaporization at inlet pressure,

\[ X_{\text{in}} = 0.013 \text{ at location 3 on the test tube} \]

Also,

\[ \Delta X_{\text{sectional}} = \frac{q_{\text{test}}}{m_F i_{fg}} = 0.07 \quad \text{(D.19)} \]
Hence, the quality and the saturation pressures at different locations can be calculated as

<table>
<thead>
<tr>
<th>No.</th>
<th>Location</th>
<th>Quality</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>3</td>
<td>0.01</td>
</tr>
<tr>
<td>2</td>
<td>4</td>
<td>0.08</td>
</tr>
<tr>
<td>3</td>
<td>5</td>
<td>0.15</td>
</tr>
<tr>
<td>4</td>
<td>6</td>
<td>0.21</td>
</tr>
<tr>
<td>5</td>
<td>7</td>
<td>0.28</td>
</tr>
<tr>
<td>6</td>
<td>8</td>
<td>0.34</td>
</tr>
<tr>
<td>7</td>
<td>9</td>
<td>0.41</td>
</tr>
<tr>
<td>8</td>
<td>10</td>
<td>0.48</td>
</tr>
<tr>
<td>9</td>
<td>11</td>
<td>0.54</td>
</tr>
<tr>
<td>10</td>
<td>12</td>
<td>0.61</td>
</tr>
<tr>
<td>11</td>
<td>13</td>
<td>0.68</td>
</tr>
</tbody>
</table>

The local heat transfer coefficients can be calculated using the local saturation temperature, the local wall temperature, and the heat flux. For example, at location 5,

\[ h_5 = \frac{q_{\text{test}}}{(T_{w5} - T_{\text{sat}5})} \]  \hspace{1cm} (D.20)

\[ = \frac{7790.29}{12.47} = 624.72 \text{ Btu/}(\text{hr ft}^2) \]

\[ = 3544.5 \text{ W/}(\text{m}^2 \text{ K}) \]
The repetitive calculations can be performed for all other locations to attain the heat transfer coefficients.

**Propagation of error**

The error propagation analysis described in this section is similar to that of average heat transfer coefficient. It should be noted that the error reported in this section is the absolute value of the maximum expected deviation for test run 1.EL.2. The detailed procedure to evaluate the experimental uncertainty is reported by Kline and McClintock (1953).

From Eq. (4.13), the local heat transfer is calculated as

$$h_i = \frac{Q_{test}}{A_{sur}(T_{v_L} - T_{sat_L})}$$

As stated by Eq. (D.1)

$$v_h = \left[ \left( \frac{\partial h}{\partial Q} \frac{\partial Q}{\partial T_{v_L}} \right)^2 + \left( \frac{\partial h}{\partial A} \frac{\partial A}{\partial T_{v_L}} \right)^2 + \left( \frac{\partial h}{\partial T_{v_L}} \right)^2 + \left( \frac{\partial h}{\partial T_{sat_L}} \right)^2 \right]^{1/2}$$

where

$$\frac{\partial h}{\partial Q}, \frac{\partial h}{\partial A}, \frac{\partial h}{\partial T_{v_L}} \text{ and } \frac{\partial h}{\partial T_{sat_L}}$$

are described by Eqs. (D.2) through (D.5).

For data point 5, and at $X = 0.28$,

$$T_{v_L} = 663.10^\circ_R$$
\[ T_{\text{sat}_L} = 650.63^\circ R \]

\[ Q_{\text{test}} = 8675.50 \text{ Btu/hr} \]

\[
\frac{\partial h}{\partial Q} = \frac{1}{A_{\text{sur}}(T_{v_L} - T_{\text{sat}_L})} = \frac{1}{(1.113)(12.47)} = 0.072 \frac{1}{(\text{ft}^2 \cdot ^\circ F)}
\]

\[
\frac{\partial h}{\partial A_{\text{sur}}} = - \frac{Q_{\text{test}}}{A_{\text{sur}}^2(T_{v_L} - T_{\text{sat}_L})} = - \frac{8675.50}{(1.11)^2(12.47)} = -564.65 \frac{\text{Btu}}{(\text{hr} \cdot \text{ft}^4 \cdot ^\circ F)}
\]

\[
\frac{\partial h}{\partial A_{v_L}} = \frac{Q}{A_{\text{sur}}(T_{v_L} - T_{\text{sat}_L})^2} = \frac{8675.50}{(1.11)^2(12.47)^2} = 50.26 \frac{\text{Btu}}{(\text{hr} \cdot \text{ft}^2 \cdot ^\circ F^2)}
\]

\[
\frac{\partial h}{\partial T_{\text{sat}_L}} = - \frac{Q}{A_{\text{sur}}(T_{v_L} - T_{\text{sat}_L})^2} = - \frac{8675.50}{(1.11)^2(12.47)^2} = -564.65 \frac{\text{Btu}}{(\text{hr} \cdot \text{ft}^2 \cdot ^\circ F^2)}
\]

The uncertainties \( \bar{W}_Q, \; \bar{W}_A_{\text{sur}}, \; \bar{W}_T_{\text{sat}} \) and \( \bar{W}_{T_{\text{wall}}} \) are estimated as in Eq. (D.1).
Estimate $W_q$

$$Q_{test} = V_T I_S 3.412$$  \hspace{1cm} (D.21)

Thus,

$$W_q = \left( \frac{\partial Q}{\partial V_T} W_{I_S} \right)^2 + \left( \frac{\partial Q}{\partial V_T} W_{V_T} \right)^2 \right)^{1/2}$$  \hspace{1cm} (D.22)

$$= \left[ (3.412 I_S W_{I_S})^2 + (3.412 V_T W_{V_T})^2 \right]^{1/2}$$

$W_{I_S} = 0.01 \text{ mV}$

$W_{V_T} = 0.10 \text{ V}$

Hence,

$$W_q = \left[ (3.412)(543.3)(0.01) \right]^2 + \left[ (3.412)(4.68)(0.10) \right]^2$$

$$= 18.60$$

Estimate $W_A$

$D_i = 0.0285 \text{ ft}$

$L = 12.4 \text{ ft}$

$W_{D_i} = 0.001 \text{ ft}$
Using Eq. (D.12)

\[
W_A = [(\pi \times 12.4 \times 0.001)^2 + (\pi \times 0.0286 \times 0.01)^2]^{1/2} = 0.0389
\]

Estimate \( W_{T_{\text{sat}L}} \) and \( W_{T_{\text{WL}}} \)

The estimated values of \( W_{T_{\text{sat}L}} \) and \( W_{T_{\text{WL}}} \) are described in the earlier part of this Appendix (Eq. D.16):

\[
W_{T_{\text{sat}L}} = 0.3^\circ\text{F}
\]

\[
W_{T_{\text{WL}}} = 0.3^\circ\text{F}
\]

The uncertainty for heat transfer coefficient is then given by Eq. (D.1)

\[
W_h = [(0.072 \times 18.60)^2 + (-564.65 \times 0.0389)^2 + (50.26 \times 0.3)^2
\]

\[
+ (50.26 \times 0.30)^2]^{1/2}
\]

An uncertainty of approximately \( \pm 5\% \) is calculated for a typical experimental run.
APPENDIX E. DETAILS OF EXPERIMENTAL LOOP (R-22) COMPONENTS

The experimental arrangement for determining the local and the average heat transfer coefficient using R-22 as a refrigerant were described earlier in Chapters V and VI, respectively. A detailed description of the equipment used is described in this section.

Pump

A Wanner Engineering diaphragm pump (Model #D-10) was used to circulate the R-22 in the test flow loop. The pump shaft was coupled to a 1/4 hp motor whose speed was reduced from 1750 to 125 rpm using a belt-and-pulley mechanism. The neoprene diaphragms of the pump were reciprocated to pressurize the test fluid using a cam and plunger assembly connected to the pump shaft. Except for the diaphragms, the rest of the moving parts of the pump were immersed in an oil bath for lubrication. The pump was capable of circulating 0-2 gpm of test fluid at a maximum system pressure of 3000 psi. A Sporlan model C-414 filter-dryer unit installed downstream of the pump was used to remove the contamination particles and the moisture from the test fluid.

After-condenser/Condenser

An after-condenser/condenser was installed at the exit of the test section to condense and cool the refrigerant exiting from the test section. An American Standard (Model HCF #02036) shell-and-tube-type heat exchanger with refrigerant circulating in the tube side was used as a condenser. The shell side of the condenser was supplied with the
chilled water-glycol mixture of the condenser flow loop. The thermo-couples placed at the inlet and outlet of the shell-and-tube side of the condenser were useful in monitoring the test fluid system pressure.

Boiler

The test fluid was heated and boiled using the boiler before entering the annulus test section (Chapter VI). The subcooled refrigerant entering into the boiler was electrically heated (dc power) using a 0.434 in. (11.0 mm) O.D., 8.64 ft (2.63 m) long, and 0.054 in. (1.37 mm) thick stainless steel 304 tube. Additionally, a 8.9 ft (2.71 m) long twisted tape having 5 tube diameters per 180° turn was inserted inside the boiler tube to augment the dryout heat flux.

R-12 Refrigeration Unit

The evaporation tests required system operating temperatures of approximately 30-50°F. Hence, a 5 ton capacity (Lennox) refrigeration unit was installed as a secondary system to the after-condenser/condenser flow loop. The evaporator coil of this refrigeration unit installed inside the storage tank of the after-condenser/condenser flow loop exchanged heat with the water-glycol mixture. This unit is characterized by an expansion valve having 10°F superheat and a thermostat with a temperature differential of 5°F. The specifications of the refrigeration unit are as follows:

Make: Lennox make HS6-651V-1C unit
Refrigerant: R-12
Condenser coil: Finned (13 fpi), 1/2 in. (12.7 mm) O.D., 38 ft (11.58 m) long (3 rows)

Evaporator coil: Smooth, 1/2 in. (12.7 mm) O.D.; 25 ft (7.62 m) long

Compressor: 5 ton, 230 V, 60 c/s

An untimely breakdown of this R-12 unit occurred at the end of local evaporation tests. Subsequently, a new refrigeration unit was installed to conduct the average evaporation heat transfer and pressure drop tests (Chapter VI). The usage of a refrigeration unit was necessary since the evaporation tests were to be performed at low temperatures (~30-50°F or -1-10°C). It should be noted that the thermostat control unit, the expansion valve, and the evaporator coil for the new unit were the same as described earlier. However, the new refrigeration unit had a water-cooled condenser. The specifications of the new refrigeration unit are as follows:

Make: Climate Control BW-0500-E5 Snyder General Corporation

Refrigerant: R-12

Condenser: Water cooled

Evaporator coil: Smooth, 1/2 in. (12.7 mm) O.D.; 25 ft (7.62 m) long

Compressor: Model MRB-0500, Semi-Hermatic 230/460 volts, 60 c/s
Superheater

A 0.5 in. (12.7 mm) O.D. by 6.2 ft (1.89 m) long copper tube was used as a superheater. This tube was heated using a 66 ft (20.11 m) long nichrome wire wound around it. This nichrome wire was electrically isolated from the superheater (copper tube) using ceramic beads. To reduce the heat loss, a layer of 1/4 in. (6.35 mm) thick asbestos tape was wrapped around the nichrome wire. Finally, the entire assembly was enveloped inside a 1/2 in. (12.7 mm) thick fiberglass insulation to further reduce the heat loss.

The nichrome wire was electrically heated using 110 V (ac) power supply. A Variac installed in the power line controlled the heat input to the test fluid. The supply voltage and current were measured using a Hewlett-Packard Digital Multimeter Model 3435A and a Fluke current transformer (Model 60I-600), respectively.

Recirculating Pump

A March (Model #TE-55C-MD) centrifugal type recirculating pump with a maximum capacity of 9 gpm was used to circulate the after-condenser flow loop fluid. The pump was connected to a 110 V (ac) motor rotating at 3450 rpm. The specific feature of the pump was its sound operation at very low (30-50°F or -1.1 to 10°C) temperatures.

A similar pump was also installed in the water flow loop for circulating water in the annulus side of the test section (Chapter VI).
Accumulator

The low boiling temperatures of the test fluid at atmospheric pressure required installation of an accumulator in the test apparatus in order to maintain the test fluid in the liquid phase at room temperature. This was achieved by pressurizing the test fluid to approximately 170 psi using the accumulator. Additionally, the high system pressures required for condensation tests could be easily attained using the accumulator. Most importantly, it served as an expansion tank which was necessary during the two-phase flow tests. An Oil Air (Model #1-1002) one gallon accumulator with EPDM bladder was installed at the exit of the pump. The bladder was compatible with the R-22 at temperatures as low as $-15^\circ$F ($-28^\circ$C).

Instrumentation

Data acquisition system

The data acquisition system consisted of a Hewlett-Packard Model 9825A computer, a Hewlett-Packard Model 3495A scanner, a Hewlett-Packard Model 3455A voltmeter, two Hewlett-Packard Model 3425A digital multimeters, and an electronic ice junction manufactured by Omega. The details of the operation of the data acquisition system were described by Luu (1979) and Jensen (1976). Computer software for single-phase, condensation, evaporation, local single-phase, local evaporation, and annulus calibration heat transfer tests were developed and are reported in Appendix F.
Temperature measurements

The temperatures across the test facility were measured using TT-T-30 copper-constantan thermocouple wires from Omega. However, the water side bulk temperatures were measured using 36 gage copper-constantan thermocouple wires. A total of 44 thermocouples for the local heat transfer and 18 thermocouples for the average heat transfer tests was used for the temperature measurement. The thermocouples from various locations on the test apparatus were directed to a central switchboard. They were then connected to a 40 channel scanner. Finally, the software developed was used to trigger the desired scanner channels so that the temperature could be read using a digital voltmeter.

Pressure measurements

The absolute and differential pressures were measured using two different Bourdan type pressure gages having \(\pm 1/4\%\) of the full scale accuracy. It should be noted that the precise pressure measurements were not essential since the system was operated at higher pressures (i.e., \(~300\) psi or \(2.06\) MPa). Additionally, a large change in the absolute pressure resulted in a relatively small change in the saturation temperature. Nonetheless, both gages were periodically calibrated using a deadweight pressure gage tester manufactured by Amther. The calibration curves are reported in the data reduction program.
Flow measurements

Two positive displacement flowmeters and a rotameter measured the flow rates of the fluids flowing in the test facility. Specifically, a piston type Connometer measured the test fluid (R-22) flow rate, the Water-Mag meter measured the water side flow rates, and the Brooks rotameter measured the water/glycol mixture flow rate. These flow meters were calibrated using a tank, an electronic balance, and a stopwatch. The calibration equations were then fitted using software developed on 9845A Hewlett-Packard computer. The details of these flow meters along with their calibration curves are as follows:

1. Test fluid (R-22) flow meter

   Connometer Model no.: B13-AAS
   Range = 0.05 - 2.0 gpm
   Accuracy = Better than 1.0% of instantaneous rate
   \[ m = \frac{2}{1000} M \]
   \( m \) in gpm, \( M \) in mv

2. Water (annulus-side) flow meter

   Water-Mag Model no.: 7485-1W1A6AA
   Range = 0 - 37 gpm
   Accuracy = ± 2%
   \[ m = 281.25 \frac{M}{1000} - 1125 \]
   \( m \) in lb/hr, \( M \) in percent

3. After-condenser fluid (ethylene-glycol) flow meter

   Brooks rotameter
   Tube number = R-10M-25-3
   Float number = 10-RV-138
Range = 0.3 - 8.5 gpm

\[ m = 17.0239 \, M + 5.1315 \]

\( m \) in lbm/hr, \( M \) in mm

The method of applying the correction factors for the density variations between the calibrated and the metered fluid is reported in Appendix F.
APPENDIX F: DATA REDUCTION COMPUTER PROGRAMS LISTINGS
Water Heated/Cooled Short Test Section

(R-113 as a refrigerant)

**Single-phase**

```plaintext
0:   wrt 0:"COMPUTER PROGRAM FOR CALCULATING"
1:   wrt 0:"SINGLE-PHASE HEAT TRANSFER COEFFICIENTS"
2:   dim QC[20],LC[18],MC[10],HC[15],XC[13],YC[18],XC[14]
3:   dim CT[20],TI[34],VC[4],FC[10],FC[83],RC[9],EC[23],EC[63],DC[53],AC[19]
4:   dim S[10],DC[10],R[3]
5:   wtb 6,32,32,32,32,32,32,32,32
6:   wtb 6,27,77
7:   fmt 3:"F07.3"
8:   fmt 2:"F16.9"
9:   ent ' NO OF STEPS?? ',N
10:  for I=1 to N
11:   K=1
12:   for J=1 to 2
13:   wrt 7,99:"CLS",K;wrt 722.3;red 722,E
14:   'TEMP'(1000E)>YC123
15:   K+1=K
16:   dsp ' channel',K;"TEMP",Y[123];wait 1000
17:   next I
18:   dsp ' SET PRINTER AT THE TOP OF PAGE';stp
19:   wtb 6,27,84
20:  wtb 6,27,87,int(12*120/64),int(12*120)
21:  wtb 6,27,76,int(11*96/64),int(11*96)
22:  fmt 1:"F07.3"
23:  fmt 2:"F16.9"
24:  ent ' TIME?? ',A$
25:  wtb 6,"TIME: ",A$
26:  wtb 6,10,10,10
27:  wait 1000
28:  wrt 6," DATA INPUT BEGINS!"
29:  wtb 6,10,10,10
30:  wrt 0:"D.C.POWER INPUT BEGINS"
31:  ent "SHUNT VOLTAGE",V[1];"V11"
33:  wait 1000
34:  dsp ' TO TAKE DATA:STOP AND CONT DATA'
35:  wait 1000
36:  wrt 6," DATA BEGINS"
37:  wrt 6,"SHUNT VOLTAGE",V[1]"
39:  dsp 'To take data:STOP and CONT data'
40:  "data"
41:  wrt 7,99,"F07.3"
42:  wait 1000
43:  wtb 6,10,10,10
44:  wrt 0:"TEMPERATURE MEASUREMENT BEGINS"
45:  dsp 'SCANNER TAKING DATA.WAIT'
46:  for I=1 to 2000;C[N];next I
47:  for I=1 to 10
48:  int M=1 to 20;wrt 7,99,"CLS",N-1;wrt 722.3;red 722,E
49:  C[N]>C[N];next M
50:  for I=1 to 2000;C[N];next I
51:  'TEMP'(10000C[N])"TIJ"
53:  fmt 7,"*F04.0;f12.9;f10.4"
54:  wtb 6,7,"F07.3"TIJ
55:  next I
56:  wtb 6,10,10,10
57:  wrt 0:"SYSTEM PRESSURES AND FLOW RATES INPUT BEGINS"
58:  ent 'FREON PRESSURE ',P[7]
59:  wtb 6,"FREON INLET PRESSURE,PSIA"
```
PC73-.315>PC13
wrt 6,"CORRECTED FREON PRESSURE,Psia",P[1]

ent "FREON PRESSURE DROP,in mm of Ha",P[3]

wrt 6»"PRESSURE DROP,in of Ha :",PC3

ent "ATMOSPHERIC PRESSURE?",PC10

25.4F(103.01934>PC10)

TC163>TC263

TC173>TC273

TC183>TC283

ent "FREON FLOW RATE?, %",FC6

wrt 6,"FLOU RATE FREON,% :",F[6]

2.81297FC6>20.21323>FC13

wrt 6,"MASS FLOW RATE OF FREON,Ibm/hr ":F[1]

ent "WATER FLOW RATE,GPM",F[7]

wrt 6,"FLOW RATE WATER,GPM :",FC7


wrt 6,"MASS FLOW RATE ON CONDENSATE SIDE,GPM ":F[3]

wrt 6,"AFTER COND FLOW RATE?,in mm",F[8]

12.7695776FC8>10.4708347>FC3

wrt 6,"AFTER CONDENSER FLOW RATE,in Ibm/hr ":F[3]

wrt 0,"TUBE DIMENSIONS"

.375>LC23

40.99/12>LC33

.343>LC13

.506>Lc53

.506>LC63

2233-RCl

wrt 6,10,10,10

wrt 6,10,10,10

wrt 6,10,10

INSIDE DIA OF TEST SECTION :

OUTSIDE DIA OF TEST SECTION,IN :

LENGTH OF TEST SECTION,in :

THERMAL CONDUCTIVITY OF TUBE,Btu/hrft :

DETAILS OF HEATER PORITON

wrt 6,10,10

wrt 6,10,10

INSIDE DIA OF HEATED SECTION,in :

LENGTH OF HEATED SECTION,ft :

ANNUlus PORTION DETAILS :

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

CALCULATION OF ALL PROPERTIES OF R-113 

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

wrt 6,10,10

DENSITY OF VAP AT TSAT,lbm/ft3 :
ENTHALPY OF VAPOUR, \( h_{VP} \), Btu/lbm

LIQ' (TC16)

DENSITY OF LIQUID, lbm/ft³

LIQT' (TC17a)

DENSITY OF LIQUID AT SAT TEMP., lbm/ft³

LATENT HEAT OF VAPORIZATION, \( h_{FD} \), Btu/lbm

HVAP' (HC11, HC31)

HVAP

TC16DTC17

OP' (TC16) > BC13

SPECIFIC HEAT OF LIQUID, Btu/lbm F

CPVP' (TC16)

CP VAPOR

MUL' (TC16)

VISCOSITY OF LIQUID, centipoise

MUV' (TC16)

VISCOSITY OF VAPOR, centipoise

KL' (TC16)

fat 9.9 K FREN' > 3Bx, f10.3

wrt 6.9, BC53

LIQR' (TC113)

DENSITY OF FREN AT COND OUTLET TEMP

PRE EVAPORATOR CALCULATION BEGINS

Y13JVC22Q11

CURRENT IN AMPS

wrt 6 * HEAT FLUX IN PREHEATER SECTION, Watts

3.142L[53L63/12]Y15J

Q113.412Y16J

3B5J/62.4XME53

(8.04-ME53)/7.04ME53.ME63

ME63 > 5SME53.ME73

VISCOSITY FACTOR FOR ROTAMETER

F113MF73HC11

F1350.49HF22

FC33ME33

.7856LJ13L13LC22

AC23/144AC22

ME13/AC23MC43

wrt 6 * FREON MASS FLOW RATE, lbm/hr ft²

wrt 6 * FREON SIDE MASS FLOW RATE, lbm/hr

wrt 6 * MASS FLOW RATE OF WATER, lbm/hr

wrt 6 * MASS FLOW RATE AFTER COND, lbm/hr

wrt 6 > DT FREON TEST SECTION

wrt 6 > DT WATER TEST SECTION
CALCULATION OF FRICTION FACTOR

**FRICTION FACTOR f**

VISCOSITY OF WATER, lbm/hr ft²

**VELOCITY IN ANNULUS OF CONDENSER**, ft/sec

DATA FILE STORAGE STARTED??

SUBROUTINES FOR CALCULATING THE PROPERTIES OF R-113 AND WATER

'TSAT':

if **PC13**<=4.374 then
  disp "P<3.174"; stop
endif

if 7.14<**PC13** and **PC13**<13.174 then
  disp "**PC13**<13.174"; stop
endif

if **PC13**<=10.07 then
  return 482.038164+10.17409**PC13**-.265311**PC13****PC13**-459.6
endif

if **PC13**<=14.84 then
  return 496.903183+7.293994**PC13**-.125058**PC13****PC13**-459.6
endif
300: if PC13<=21.19; ret 511.178705+5.424238*PC13-.06357*PC13*PC13-459.6
301: if PC13<=29.48; ret 526.236224+4.064817*PC13-.03277*PC13*PC13-459.6
302: if PE<=108.2; ret 578.075017+1.740811*PC13-.01423*PC13*PC13-459.6
303: "DVISCO":
304: if TC253->459.6 TC253
305: if TC253->50)/50 TC253
306: 5.6036-.76097 TC253+459.6 TC253
307: YC43-DC43-3+YC33-DC43-2+YC23-DC43+YC13-4
308: 3DC43-2YC43+2DC43YC33+YC23>P5
309: if abs(P6)<=.001 Jmp 3
310: ret BC63
311: "LIQ":
312: .0000ST[73]=-0.0214Y[4]
313: .002618TC173-4.035Y[3]
315: -PC13Y[1]
316: if TC173<=559.6;2>DC23;Jmp 5
317: if TC173<=591.6;3>DC23;Jmp 4
318: if TC173<=629.6;4>DC23;Jmp 3
319: if TC173<=709.6;5>DC23;Jmp 2
320: if TC173<=809.6;6>DC23
321: if TC173<=809.6;7>DC23
322: YC113-DC113-3+YC123-DC113-2+YC113-DC113+YC73-4
323: YC113-2YC113+2DC113YC123-4
324: p/5+5
325: if abs(p6)<=.001 Jmp 3
326: if abs(p6)<=.001 Jmp 5
327: ret DC43
328: ret DC43
329: "ENTV":
330: .07963TC173=1.59e-4TC173-2/2+.185053(4.035DC43+.0214DC43-2/2)>HC13
331: HC13+25.1983-HC13
332: ret HC13
333: "LIQ":
334: if TC163-459.6; TC163
335: 103.55-.0712TC163-6.36e-5TC163;2>DC13
336: TC163+459.6; TC163
337: ret DC13
338: "HFG":
350: "HVAP":
351: if TC163-459.6; TC163
352: if TC163-459.6; TC163
353: "CPLD":
354: if TC163-459.6; TC163
355: if TC163-459.6; TC163
356: if TC163-459.6; TC163
357: if TC163-459.6; TC163
358: if TC163-459.6; TC163
359: if TC163-459.6; TC163
360: if TC163-459.6; TC163
361: if TC163-459.6; TC163
361: ret BC[2J
362: 'KL':
366: ret BC53
367: 'MUL':
370: 'MVU':
374: ret BC43
375: 'LIQ':
379: ret DC53
380: 'TEMP':
381: if P1 <= 1.494; ret 31.99925+46.80117P1-1.40739P1 "2+.07802P1 "3-.00739P1 "4
382: if P1 <= 3.941; ret 33.42956+44.48835P1-.07422P1 "2-.25389P1 "3+.02879P1 "4
383: if P1 <= 6.621; ret 33.82222+45.39092P1-1.01507P1 "2+.03592P1 "3-.00064P1 "4
Evaporation

0: wrt 0,"COMPUTER PROGRAM FOR CALCULATING"
1: wrt 0,"EVAPORATION HEAT TRANSFER COEFFICIENT"
2: dim C(20),L[8],MC10],H[15],X[4],Y[18],KC4]
3: dim B[10],A[10],A(10),A(10),A(10),A(10),A(10),A(10),A(10),A(10)
4: wtb 6,32,32,32,32,32,32,32,32
5: wtb 6,27,77
6: fmt 3,"FIRA1IZ1MOPO"*
7: fmt 2,"FIRA1IZ1ST1"*
8: ent "NO of Steps??",N
9: for I=1 to N;for J=1 to 4;wrt 709,"CLS",J+6;wrt 722.3
10: red 722,E;"TEMP'(1000E);Y[12]
11: dsp "Channel",J+6,"TEMP",Y[12];wait 1000
12: next J;next I
13: dsp "SET PRINTER AT THE TOP OF PAGE";fmt
14: wtb 6,27,84
15: wtb 6,27,87,int(12*120/64),int(12*120)
16: wtb 6,27,88,int(11*96/64),int(11*96)
17: fmt 2,"F1R1H1A1TI«3"
18: fmt 2,"F1RA1Z1N5T1"
19: fx=4
20: ent "TIME??",A$
21: wtb 6;"TIME: ";A$
22: wait 1000
23: wtb 6,10,10,10
24: wait 1000
25: wtb 6,* DATA INPUT BEGINS:*
26: wtb 6,10,10,10
27: wtb 0,"DC-POWER INPUT BEGINS"
28: ent "SHUNT VOLTAGE, mV",VC13
29: ent "Terminal voltage, volts",VC2]
30: wait 1000
31: dsp "TO TAKE DATA:STP AND CONT DATA"
32: wait 1000
33: wrt 6,* DATA BEGINS*
34: wrt 6,"SHUNT VOLTAGE,mV "
35: wtb 6,"TERMINAL VOLTAGE,volts"
36: wtb 6,"TO take data:STP and CONT data"
37: wtb 6,"data:"
38: wtb 6,10,10
39: wtb 709,"FIRA1IZ1ST1"
40: wait 1000
41: dsp "SCANER READY TO TAKE DATA"
42: wait 1000
43: wtb 0,*TEMPERATURE READOUT BEGINS NOW*
44: for I=1 to 10;for N=1 to 20
45: wtb 709,"CLS",N-1;wrt 722.3;red 722,E
46: C(N)="i";C(N)="j";i=10;N=20;N=1;N=2;N=3;N=4;N=5;N=6;
47: for I=1 to 20;CE=i/100;i=10;CE=j;C(i);i=10;N=1;N=2;N=3;N=4;
48: wtb 6,10,10
49: wtb 6,*"MILLIVOLTS",*
50: wtb 6,10,10
51: for I=1 to 20
52: wtb 0,"TEMP'(1000DII))TII"
53: TII=459.6;TII=TII/1000;CII
54: fmt 7,*"f4.0",f12.9;10*x;f10.4
55: wtb 6.7;i,CII,TII
56: CII=1000000000
57: next I
58: TII=777;TC273;TC183;TC283;TC193;TC293;TC203;TC303
59: wtb 6,10,10
362

40: \texttt{wrt 6, "TEMPERATURES AT DIFFERENT LOCATIONS"}
41: \texttt{wrt 6, 10}
42: for I=1 to 16; \texttt{T[I]-459.6-T[I];next I}
43: for I=1 to 7; \texttt{wrt 6, "TUBE WALL TEMPERATURE OF \#T[I];next I}
44: \texttt{wrt 6, "WATER INLET TEMP (TEST SECTION) OF \#T[8];}
45: \texttt{wrt 6, "WATER OUTLET TEMP (TEST SECTION) OF \#T[9];}
46: \texttt{wrt 6, "FREON INLET TEMP (PREHEATER SECTION) OF \#T[10];}
47: \texttt{wrt 6, "FREON OUTLET TEMP (AFT COND) OF \#T[13];}
48: \texttt{wrt 6, "FREON INLET (AFT COND) OF \#T[14];}
49: \texttt{wrt 6, "FREON OUTLET TEMP (AFT COND) OF \#T[15];}
50: \texttt{wrt 6, "OUTLET WATER TEMP (CHECK) OF \#T[16];}
51: \texttt{wrt 6, "INLET WATER TEMP (CHECK) OF \#T[27];}
52: \texttt{wrt 6, "INLET WATER TEMP (CHECK) OF \#T[28];}
53: \texttt{wrt 6, "HEAT EXCH INLET TEMP OF \#T[29];}
54: \texttt{wrt 6, "HEAT EXCH OUTLET TEMP OF \#T[30];}
55: \texttt{wrt 6, 10, 10}
56: for I=1 to 16; \texttt{T[I]+459.6-T[I];next I}
57: \texttt{wrt 6, 0, "SYSTEM PRESSURES AND MASS FLOW RATES INPUT BEGINS"}
58: \texttt{ent "FREON PRESSURE\#1", P[9]}
59: \texttt{wrt 6, "FREON INLET PRESSURE, PSIA: \#P[9];}
60: \texttt{PC93-.3142>PC13}
61: \texttt{wrt 6, "CORRECTED INLET PRESSURE, PSIA: \#P[13];}
62: \texttt{P[13].01934PC32.4-PC22}
63: \texttt{wrt 6, "CHECK INLET PRESSURE, PSIA: \#P[2];}
64: \texttt{PC22/14.5013-PC22}
65: \texttt{wrt 6, "FREON PRESSURE DROP, PSIA: \#P[3];}
66: \texttt{P[3/>.01934PC32.4-PC22}
67: \texttt{wrt 6, "PRESSURE DROP IN PSIA: \#P[3];}
68: \texttt{PC22/14.5013-PC22}
69: \texttt{wrt 6, "PRESSURE DROP IN Bar: \#P[2];}
70: \texttt{P[2]/14.5013-PC22}
71: \texttt{wrt 6, "ATMOSPHERIC PRESSURE, PSIA: \#P[4];}
72: \texttt{P[4].01934PC32.4-PC22}
73: \texttt{wrt 6, "ATM PRESSURE, PSIA: \#P[4];}
74: \texttt{P[4].01934PC32.4-PC22}
75: \texttt{wrt 6, "FREON FLOW RATE?, \#F[6];}
76: \texttt{ent "FREON FLOW RATE?, \%", \#F[6];}
77: \texttt{2.81297F6J-20.21328F6J}
78: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
79: \texttt{F[6].01934PC32.4-PC22}
80: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
81: \texttt{F[6].01934PC32.4-PC22}
82: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
83: \texttt{F[6].01934PC32.4-PC22}
84: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
85: \texttt{F[6].01934PC32.4-PC22}
86: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
87: \texttt{F[6].01934PC32.4-PC22}
88: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
89: \texttt{F[6].01934PC32.4-PC22}
90: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
91: \texttt{F[6].01934PC32.4-PC22}
92: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
93: \texttt{F[6].01934PC32.4-PC22}
94: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
95: \texttt{F[6].01934PC32.4-PC22}
96: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
97: \texttt{F[6].01934PC32.4-PC22}
98: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
99: \texttt{F[6].01934PC32.4-PC22}
100: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
101: \texttt{F[6].01934PC32.4-PC22}
102: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
103: \texttt{F[6].01934PC32.4-PC22}
104: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
105: \texttt{F[6].01934PC32.4-PC22}
106: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
107: \texttt{F[6].01934PC32.4-PC22}
108: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
109: \texttt{F[6].01934PC32.4-PC22}
110: \texttt{wrt 6, "FLOW RATE FREON, \%: \#F[6];}
111: \texttt{wrt 6, "TUBE DIMENSIONS"}
112: \texttt{.375LE22f, .343LE11}
113: \texttt{40.99/12LE11}
114: \texttt{.75LE11}
115: \texttt{9.98LE63}
116: \texttt{2230RE11}
117: \texttt{wrt 6, 10, 10, 10}
118: \texttt{1C}
119: \texttt{wrt 6, 10, 10}
HEAT BALANCE CALCULATION BEGINS

FREON SIDE HEAT TRANSFER (SENS. HEAT)

WATER SIDE HEAT TRANSFER

HEAT BALANCE AFTER CONDENSER

FREON AFTER CONDENSER (TOTAL)

FREON HEAT TRANS (AFT. COND.) LATENT

WATER AFTER COND

% DIFF AFTER COND

LOOP HEAT BALANCE (WATER BASE)

TOTAL LOOP BALANCE, FREON BASE

CALCULATION OF HEAT TRANSFER COEFF, Btu/hr ft²

AVERAGE WALL TEMP, °F

AV HT COEFF

AV HT COEFF U/ft²°C

TEMP DIFF FREON SIDE

TEMPERATURE DIFF WATER SIDE

TEMP DIFF FREON SIDE AFT. COND

TEMPERATURE DIFF WATER AFT. COND

EXIT FREON QUALITY

CALCULATION OF NON DIMENSIONAL PARAMETERS

REYNOLDS NUMBER, LIQUID BASE

REYNOLDS NO, VAPOR PHASE

PRANDTL NUMBER, LIQUID

PRANDTL NUMBER, VAPOR

NUSELT NUMBER

FRICTION FACTOR "f"

FRICTION FACTOR, "CF"

CALCULATION for WATER SIDE REYNOLDS No.

CALCULATION for WATER SIDE REYNOLDS No.

FRICTION FACTOR for WATER SIDE REYNOLDS No.

VELOCITY IN ANNULUS OF CONDENSER, ft/sec
\[366\]

300: \text{L[4]}-\text{LC[2]}\text{R[6]}\]
302: wtb 6:9REYNOLDS No. WATER SIDE 
303: wtb 6:9FLOW TYPE:ANNUAL FLOW
304: wtb 6:9IMPORTANT PARAMETERS
305: utb 6:10:10:10
306: wtb 6:Mass flow rate :\text{MC[4]}
307: wtb 6:InLET PRESURE :\text{P[13]}
308: wtb 6:Inlet quality :\text{X[11]}
309: wtb 6:change in quality :\text{X[23]}
311: wtb 6:10:10:10
312: wtb 6:average quality :\text{X[4]}
313: wtb 6:HEAT TRANSFER COEFF :\text{H[12]}
314: wtb 6:wall average temp :\text{T[18]}
315: wtb 6:Saturation temperature :\text{T[17]}
316: wtb 6:DT wall :\text{T[19]}
317: wtb 6:in heat in water temp :\text{T[22]}
319: wtb 6:heat flux in preheater section :\text{Y[16]}
320: dsbp *DATA FILE STORAGE STARTS NOW*
321: ent *FILE NAME?????*G$
322: G$
323: open G$,J
324: asdn G$+1.0
325: spvt 1[X+1,][L[3]*MC[3]*HC[3]*X[3]*Y[3]*K[3]]
326: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
327: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
328: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
329: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
330: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
331: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
332: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
333: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
334: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
335: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
336: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
337: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
338: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
339: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
340: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
341: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
342: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
343: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
344: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
345: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
346: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
347: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
348: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
349: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
350: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
351: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
352: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
353: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
354: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
355: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
356: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
357: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
358: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
359: spvt 1[X+1,][MC[3]*HC[3]*X[3]*Y[3]*K[3]]
360: \( Y^2 - Y [123] + 2.0539\times 10^{-3} < Y [7] \)
363: \( \text{ret } Y [9] \)
364: \( \text{"HVAP" } \)
366: \( \text{ret } HC[2] \)
367: \( \text{"CPLQ" } \)
369: \( \text{wrt } 0, \text{"CPLQ" } \)
371: \( Y [63] - 23846 \times Y [63] \)
372: \( \text{ret } Y [63] \)
373: \( \text{"CPUP" } \)
375: \( B [23] - 23846 \times B [23] \)
376: \( \text{ret } B [23] \)
377: \( \text{"KL" } \)
379: \( .57789 ( .0802 - 0.00205 Y [23] ) Y [8] \)
381: \( \text{ret } Y [8] \)
382: \( \text{"MUL" } \)
383: \( \text{if } Y [23] \leq 609.6 \text{ret } 10.48364 - .03139 Y [23] + 2.443 e^{-5} Y [23] Y [23] \)
385: \( \text{"MUL" } \)
389: \( \text{ret } T [8] \)
390: \( \text{"LIQQ" } \)
391: \( Y [23] - 459.6 Y [23] \)
392: \( 103.55 - 0.0712 Y [23] - 6.34 e^{-5} Y [23] T [163] \)
393: \( Y [23] + 459.6 Y [23] \)
394: \( \text{ret } Y [3] \)
395: \( \text{"TEMP" } \)
396: \( \text{if } P [1] \leq 1.494 \text{ret } 31.9928 + 46.80117 e^{-1} + 1.40739 e^{-3} + 0.00394 e^{-4} \)
397: \( \text{if } P [1] > 3.944 \text{ret } 33.4295 + 44.48835 e^{-1} + 0.7422 e^{-3} - 2.53895 e^{-3} + 0.2087 e^{-4} \)
398: \( \text{if } P [1] \leq 6.621 \text{ret } 33.8222 + 45.3909 e^{-1} + 1.01507 e^{-3} + 0.0359 e^{-4} \)
Condensation

The data reduction program reported earlier for computing the average evaporation heat transfer coefficients was also used for calculating condensation heat transfer coefficients with few modifications. The details of the data reduction procedure are reported in Chapter III.
Electrically Heated Long Test Section

(R-113 as a refrigerant)

Single-phase

0: wrt 0,*"COMPUTER PROGRAM FOR CALCULATING"
1: wrt 0,*"SINGLE-PHASE HEAT TRANSFER COEFFICIENT"
2: dim ALS[1:320],CC[401:1200],HC[601:943],GC[25]
3: dim LC[41:MC73],PF[341:101123],TC[951:1439:333],X[15]
4: dim RC1001:SC401:KC601
5: dim UKC201
6: wrt 0,*"SET THE PRINTER AT THE TOP OF THE PAGE"
7: 600
8: wb 0,27,84
9: wb 0,27,87,int(12*120/64),int(12*120)
10: wb 0,27,76,int(11*96/64),int(11*96)
11: fmt 1,"F1R1H1A1M3"
12: fmt 2,"F1R1A1Z1N5T1"
13: fmt 3,"F1R1T1Z1M0P0*
14: fix 4
15: dsp *"TRIAL CHECK ON WALL TEMP"
16: *"CH1":
17: ent *"NO OF STEPS??",Z
18: 0K
19: for I=1 to Z;K=1;for J=1 to 5
20: if J=5;K=2K
21: wrt 730,*"CLS",Kwrt 722.3;red 722,E;K+5;K
22: 'TEMP'(1000E)>Y[1]
24: wait 1000;next J;next I
25: dsp *"TRIAL CHECK OVER?":wait 500
26: ent *"want to run again?":if yes for 0",Z
27: if Z=1;goto *"CH1"
28: dsp *"BULK TEMP CHECK BEGINS NOW"
29: *"CM2":ent *"NO OF TRIALS FOR BULK TEMP",N
30: for I=1 to N;for J=1 to 2;K=26;K=J
31: while 722.3;red 722,E;TEMP'(1000E)>Y[1]
33: wait 1000;next J;next I
34: ent *"want to run again?":if yes for 0",Z
35: if Z=1;goto *"CM2"
36: *"CM3":ent *"NO OF TRIALS FOR GUARD HEATER TEMP",N
37: for I=1 to N;for J=1 to 3;K=17;K=J
38: J+17;Kwrt 729,*"CLS",K-1wrt 722.3;red 722,E
39: 'TEMP'(1000E)>Y[1]
40: *"TEMP'(1000E)>Y[1]
41: dsp *"WALL TEMP",Y[1];wait 1000;next J;next I
42: ent *"want to run again?":if yes for 0",Z
43: if Z=1;goto *"CM3"
44: dsp *"TRIAL RUN IS OVER"
45: wrt 0,*"DATA RUN BEGINS NOW"
46: wb 0,10,10
47: dsp *"DATA READOUT STARTS NOW"
48: wrt 0,*"POSITION THE SWITCH TO 1":stp
49: wrt 0,*"D.C.POWER INPUT BEGINS"
50: ent *"SHUNT VOLTAGE??",VC23
51: ent *"TERMINAL VOLTAGE??",VC1
52: wrt 0,*"TEMPERATURE READOUT BEGINS NOW"
53: for I=1 to 10;for N=1 to 30
54: wrt 739,*"CLS",N-1wrt 722.3;red 722,E
55: CCN+2;CNJnext N;next I
56: wrt 0,*"MILLIVOLTS","RANKINE"
57: wb 0,10
58: dsp *"CHANGE THE SCANNER CHANNEL POSITION to 2":stp
59: for I=1 to 10;for N=1 to 7;N+13K
60: `CLS'*K-1;wr 722.3;red 722*E
61: C1+13;+E+J;CCK+18
62: for I=1 to 38
63: IF I=1 then 38
64: for I=1 to 38
65: C1+13;+E+J;CCK+18
66: for I=1 to 38
67: for I=1 to 38
68: for I=1 to 38
69: fmt 7,* ',f4.0,4x,fl2.9,10x,fl0.4
70: wrt 6.7*I;CCI3>T;CCI3;fl000;CCI3;CCI3
71: for I=1 to 5;0}P[I];next I
72: wtb 0,10
73: wrt 0;"SYSTEM PRESSURES AND MASS FLOW RATES INPUT BEGINS";
74: ent "FREON INLET PRESSURE",PC13
75: wrt 0;"FREON INLET PRESSURE",PC13
76: PC13-.3142>PC53
77: wrt 0;"CORRECTED FREON PRESSURE",PC53
78: ent "INLET PRESSURE(CHECK) USING NASA",PC23
79: wrt 0;"INLET PRESSURE USING NASA",PC23
80: 1.0100886PC23-.130828>PC63
81: wrt 0;"CORRECTED INLET PRESSURE",PC63
82: ent "ATM PRESSURE:in of Ha",PC33
83: wrt 0;"ATM PRESSURE IN Ha",PC33
84: .01934PC3325.4>PC33
85: wrt 0;"ATMOSPHERIC PRSSURE,Psia",PC33
86: PC53>PC153;PC153-PC83>PC183;PC183-PC103>PC213
87: PC213-PC123>PC243;PC243-PC143>PC273
88: wrt 0;"Pressure at five Locations"
89: wtb 0,10
90: fmt 4,2x,f4.1,3x,5x,f8.4,5%,f8«4
91: wtb 0,10
92: wrt 0;"SECTION","PRESSURE","DP"
93: wtb 0,10
94: wrt 6.4;I,PC153,PC83
95: 2>I
96: wrt 6.4;I,PC183,PC103
97: 3>i
98: wrt 6.4;I,PC213,PC123
99: 4>I
100: wrt 6.4;I,PC243,PC143
CALCULATION FOR THE HEAT FLUX IN COPPER

HEAT TRANSFER IN WATTS

HEAT TRANSFER IN BTU/HR

CALCULATION OF PROPERTIES AT INTERPOLATED TEMP

CALCULATION OF LOCAL TEMP USING HEAT FLUX

CALCULATION OF ALL THE PROPERTIES

COMPARISON OF THE TEMPERATURES FOR LIQUID R13
CPLG'((V111J)BC11)

fat 9.5x+9.4x+5.4x+5.1x+5.4x+10.5x+3.4x+10.4

wrt 6.9*(DC111+DC114+DC1101+BC11

0:Y11J

next I

wrt 0"CALCULATION OF ALL THE MASS FLOW RATE BEGINS NOW"

wrt 0"FREON SIDE MASS FLOW RATE "M[1]

wrt 0,'MASS FLOW RATE IN AFTER CONDENSER "MC3J

wrt 0"DIMENSIONS FOR THE TUBE BEGINS NOW"

.375>LC23î.343>LC13;i2.5>LC33;3>LC43

"OUTSIDE DIAMETER OF THE TUBE "*LC23

"INSIDE DIAMETER OF THE TUBE "*LC13

"LENGTH OF THE TEST SECTION "*LC33

"EACH SECTION LENGTH "*LC43

"AREA OF CROSS SECTION (TUBE) "*AC33

"SURFACE AREA OF THE TUBE [O.D] "*AC3J

wrt 0,"CALCULATION FOR THE HEAT FLUX IN COPPER"

wrt 0,"SHUNT VOLTAGE "*VC2J

wrt 0,"TERMINAL VOLTAGE "*VC11

wrt 0,"HEAT FLUX IN BTU/HR "*QG2J

wrt 0,"HEAT FLUX IN BTU/HR FT**2 "*OJ3J

wrt 0,"HEAT TRANSFER IN EACH SECTION "*OJ4J

wrt 0,"CALCULATION FOR HEAT BALANCE BEGINS NOW "

MC111BC(12)TC(12)+(12)TC(12)

wrt 0,"TOTAL HEAT TRANSFER "*OJ6J

wrt 0,"HEAT FLUX DUE TO ELECTIC HEAT "*OJ2J

wrt 0,"AFTER CONDENSER HTF FREON SIDE] "*OJ7J

wrt 0,"AFTER CONDENSER HTF WATER SIDE] "*OJ9J

wrt 0,"HEAT TRANSFER BASE "*OJ8J

wrt 0,"HEAT BALANCE FREON BASE "*OJ8J

wrt 0,"HEAT BALANCE WATER BASE "*OJ8J

wrt 0,"HEAT BALANCE AFT COND[FEON] "*OJ10J

wrt 0,"HEAT BALANCE AFT COND[WATER] "*OJ10J

wrt 0,"HEAT BALANCE AFT COND[WATER] "*OJ10J

for 1=1 to 5

next I

wrt 0"CALCULATION OF HEAT TRANSFER COEFF BEGINS NOW".
for I=1 to 13
   0,YCI+133, next I
   for 1=1 to 13
      0,I,YCI3, YCI10+I3
      next I
   for 1=1 to 13
      MC13/AC23>MC43
      for 1=1 to 13
         MC43/AC13+403>RCI3
         RCI3/12>REI3
      next I
      for 1=1 to 13
         HEI3/MCI3+203>RCI+403
         RCI+403/12>RCI+403
      next I
      0, "NUMBER"*, "REYNOLDS"*, "PR NO"*, "NU NO"
      fmt 6,3x, f8.4, 3x, f10.4, 3x, f10.4
      for 1=1 to 13
         6.9, I, REI+603, SEI+203
      next I
   0, "PETUKHOV-POPOV'S CORRELATION OUTPUT*"
   0, "SECT", *Nusselt No.*", "NU/Pr.4", "NuE"
   fmt 7, 3x, f6.2, 3x, f8.4, 3x, f8.4
   for 1=1 to 13
      6.7, I, KI+203, KI+403, SEI+203
   next I
   0, "CHECKING PR NO EXPONENT *
   for I=1 to 13
      0.023RII+8RII+203*.4>RII+603
      RII+603/REI+203*.4>RII3
      (1.82103CKCI3)-1.62)''2>KLI3
      1/KCI3>KCI3
      RCI3RCI+203KCI3/8>KEI+203
      <KEI3/8)-'.5<REI+203''.66-l>12.7>KEI3
      KCI3+1.07>KEI3; KEI+203/KEI3>KEI+203
      KEI+203/REI+203*.4>KEI+403
   next I
next I
ent "FILE NAME??????";US
open US;J
assn US;1;0
sprt 1,AC*];BC*];CC*];DC*];E[*];FC*];GC*]
sprt 1,LC*];MC*];PC*];QC*];TC*];UC*];VC*];X[*]
sprt 1,R[*];SC*];KC*];"end"
prt "file name",US
end

dsp "SUBROUTINES FOR CALCULATING PROPERTIES OF R-113"

YL3-459.6,YL13
103.55-.0712YL13-6.36e-5YL13YL13>3
YL13+459.6,YL13
0,YL13
ret P3

*CLQ*:
YL13-459.6,YL13;YL13-32>YL13;YL13/1.3>YL13;YL13+273>YL13
ret YL23.23884
*KL*:
YL13-459.6,YL13;YL13-32>YL13;YL13/1.8>YL13
ret YL23.23884

MUL:
if YL13<=609.6;ret 10.48364-.031393YL13+2.443e-5YL13YL13
if YL13>=609.6;ret 4.13253-9.97482e-3YL13+6.35e-6YL13YL13

LIQR:
T113-459.6,T113
103.55-.0712T113-6.36e-5T113T113>5
T113+459.6,T113
ret D15

*HUL*:
if YL13<=509.6;ret 10.48364-.031393YL13+2.443e-5YL13YL13
if YL13>=509.6;ret 4.13253-9.97482e-3YL13+6.35e-6YL13YL13

*LIQR*:
T113-459.6,T113
103.55-.0712T113-6.36e-5T113T113>5
T113+459.6,T113
ret D15

*TEMP*:
if P1<=1.494;ret 31.9992544.80117e-1-1.407396e1^2+0.07802e1^3-.007394e1^4
if P1<=3.941;ret 33.4295644.48635e-1-.07422e1^2-.253895e1^3+.02878e1^4
if P1<=6.621;ret 33.8282245.39092e-1+1.05078e1^2+.03592e1^3-.00642e1^4
Evaporation

0: wrt 0,"COMPUTER PROGRAM FOR CALCULATING"
1: wrt 0,"EVAPORATION HEAT TRANSFER COEFFICIENTS"
2: dim A[5], B[3], C[48], D[50], H[60], F[4], G[25]
3: dim LC4[], M[7], P[27], 0:6[], TI:73[], VC4[], Y[23], X[40]
4: dim GSC[0]
5: dsp "SET THE PRINTER AT THE TOP OF THE PAGE"
6: 6>0
7: wtb 0,27,84
8: wtb 0,27,87,int(12*120/64),int(12*120)
9: wtb 0,27,76,int(11*96/64),int(11*96)
10: fmt 1,"F1R1H1A1T1M3"
11: fmt 2,'F1RA1Z1N5T1"
12: fmt 3,"F1R1T1Z1M0P0"
13: fnd 4
14: dsp 'TRIAL CHECK ON BULK TEMP'
15: 'BULK':
16: ent 'NO OF STEPS?', Z
17: for I=1 to Z; for J=1 to 2; 26+J; K; wrt 709;'CLS', K
18: wrt 722.3; red 722.3; E
19: 'TEMP'(1000E) Y[I]
20: dsp 'BULK TEMP'; J; Y[I]; I
21: ent 'WANT TO RUN AGAIN?'; yes 0
22: if N=1; goto 'BULK'
23: dsp 'TRIAL CHECK ON WALL TEMP'
24: 'CH1':
25: ent 'NO OF STEPS?', Z
26: 0; K
27: for I=1 to Z; for J=1 to 5; K
28: if K=21; 23; K
29: wrt 709;'CLS'; K; for J=1 to 2; 26+J; K; I=K+5; K
30: 'TEMP'(1000E) Y[I]
31: dsp 'CHL'; K-4; 'TEMP'; Y[I]
32: wait 1000; next I; wait 500
33: ent 'want to run again?'; yes 1; or 0; Z
35: if Z=1; goto 'CH1'
36: dsp 'SUPERHEAT CHECK BEGINS NOW'
39: ent 'ATMOSPHERIC PRESSURE IN MM OF Hg'; P[3]
40: .01934; PI[3]; 43; PI[3]
41: PI[S3]; PI[43]; PI[S3]
42: 'TSAT'(PI[S3]); PI[S3]
43: 'CH2':ent 'NO OF TRIALS FOR SUPERHEAT'; Z
44: for I=1 to Z; wrt 709;'CLS'; 27
45: wrt 722.3; red 722.3; 'E' 'TEMP'(1000E) Y[I]
46: dsp 'Channel'; 27; 'temp'; Y[I]; Y[I]; Y[I]-Y[E]; Y[I]
47: dsp 'superheat'; YE33; wait 1000; next I
48: ent 'WANT TO RUN AGAIN?'; yes 1; or 0; Z
49: if Z=1; goto 'CH2'
50: 'CH3':ent 'NO OF TRIALS FOR GUARD HEATER'; N
51: dsp 'CHANGE THE SWITCH POSITION'; ist
52: for I=1 to N; for J=1 to 3
53: J+17; K; wrt 709;'CLS'; K; I; K; for J=1 to 3
54: 'TEMP'(1000E) Y[I]
55: dsp 'WALL TEMP'; J; YE13; wait 1000; next I; wait 500
56: ent 'WANT TO RUN AGAIN?'; if yes 1; Z
57: if Z=1; goto 'CH3'
58: dsp 'TRIAL RUN IS OVER'
59: wrt 0,"DATA RUN BEGINS NOW"
39: wrt 0,"DATA RUN BEGINS NOW"
60: ent "TIME":G$
61: wrt 0,"TIME":G$
62: wtb 0,10,10
63: dsp "DATA READOUT STARTS NOW"
64: dsp "POSITION THE SWITCH TO 1; setp
65: wrt 0,"D.C.POWER INPUT BEGINS"
66: ent "enter the shunt voltage":VC2
67: ent "TERMINAL VOLTAGE":V11
68: wrt 0,"TEMPERATURE READ OUT BEGINS"
69: for I=1 to 10; for N=1 to 20
70: wrt 709,"CLS",N-1; wrt 722,3; red 722,E
71: CCN2+3 next N; next I
72: 202Z; for I=1 to 10; for N=1 to 8
73: Z+1Zi; wrt 709,"CLS",Z; red 722,3; next N; next I
74: CCZ+1]+E; CCZ+1]; next N; next I
75: wtb 0,10,10
76: wrt 0,*"MILLIVOLTS*",*"RANKINE"
77: wtb 0,10
78: dsp * CHANGE THE SCANNER CHANNEL POSITION to 2; * setp
79: for I=1 to 10; for N=1 to 7; IN+13K
80: wrt 709,"CLS",K-1; wrt 722,3; red 722,E
81: CCX+13+4; CCK+13; next N; next I
82: for I=1 to 39
83: CC11/10CC11/TEMP'(1000CC11)/TIIjTIIj+459.6); IIj
84: next I
85: TC23Jj; TC24Jj; TC25Jj; TC26Jj; TC27Jj; TC28Jj; TC29Jj; TC30Jj; TC31Jj; TC32Jj; TC33Jj; TC34Jj; TC35Jj; TC36Jj; TC37Jj; TC38Jj; TC39Jj;
86: for I=1 to 39
87: CC11000CC11; fmb 7; *"f4.0"; f12.9; f10.4
88: wrt 6.7,1,CC11j459.6j; next I
89: for I=1 to 23
90: wrt 0,"*WALL TEMPERATURES*TIIj
91: next I
92: wrt 0,"R-113 INLET BULK TEMP",T[24]
93: wrt 0,"R-113 OUTLET BULK TEMP",T[25]
94: wtb 0,10
95: wrt 0,"CHECK THERMOCOUPLES"
100: wtb 0,10
101: wtb 0,10
102: TC23Jj; TC24Jj; TC25Jj; TC26Jj; TC27Jj; TC28Jj; TC29Jj; TC30Jj; TC31Jj; TC32Jj; TC33Jj; TC34Jj; TC35Jj; TC36Jj; TC37Jj; TC38Jj; TC39Jj;
103: for I=1 to 9; T[32]
104: for I=0; "WALL TEMPERATURES [T at 10",T[33]
105: for I=0; "WALL TEMP [T at 11",T[34]
106: for I=1 to 5; 0,PS11j; next I
107: wtb 0,10
108: wrt 0,"SYSTEM PRESSURES AND MASS FLOW RATES INPUT BEGINS"
111: P[11]-3.1403PSI
112: wrt 0,"CORRECTED FREON PRESSURE",P[15]
113: ent "INLET PRESSURE(CHECK) USING NASA":P[22]
114: wrt 0,"INLET PRESSURE USING NASA",P[22]
117: ent "ATM PRESSURE":in of Ha",P[33]
118: .01934P[33]j
119: wrt 0,"ATMOSPHERIC PRESSURE",Psia"
120: PC53>PC33>PC53?PC63>PC33>PC63
121: wrt 0,"INLET PRESSURE","P[5]
122: wrt 0,"INLET PRESSURE(CHECK)",P[6]
123: dsp "MEASUREMENT OF DP BEGINS NOW"
124: ent "DP FOR SECTION 1",PC73
125: ent "DP FOR SECTION 2",PC93
126: ent "DP FOR SECTION 3",PC113
127: ent "DP FOR SECTION 4",PC133
128: ent "FREON SIDE MASS FLOW RATE",PC133
129: ent "AFTER CONDENSER WATER SIDE MASS FLOW RATE",PC133
130: dsp "DATA SET COMPLETE";wait 1000
131: dsp "CALCULATION OF DP BEGINS NOW"
132: wrt 0,"PRESSURE DROP CALCULATION BEGINS NOW"
133: .01934PC7325.4>PC83
134: .01934PC9325.4>PC103
135: .01934PC11325.4>PC123
136: .01934PC13325.4>PC143
137: dsp "CALCULATION OF LOCAL PRESSURE";wait 1000
140: wrt 0,"Pressure at five Locations"
141: wtb 0;10
142: fmt 4.2;f4.1;3.x;s8.4;4.x,s8.4
143: wtb 0;10
144: wrt 0;"SECTION","","PRESSURE","","DP"
145: wtb 0;10;10
146: 1:1
147: wrt 6.4;I;P[15];P[8]
148: 2;1
149: wrt 6.4;I;P[18];P[10]
150: 3;1
151: wrt 6.4;I;P[21];P[12]
152: 4;1
153: wrt 6.4;I;P[24];P[14]
154: 5;1
155: wrt 6.4;I;P[27]
156: wtb 0;10
157: dsp "Linear Interpolation Begins";wait 1000
158: wrt 0,"LINEAR INTERPOLATION OF THE PRESSURE BEGINS NOW"
159: wtb 0;10
160: for J=1 to 3;J+K
161: J=3;J
164: (P[27]+P[24]+P[27]+P[24])
165: next J
166: wtb 0;10
167: fmt 4.3;x;f4.1;5.x;s8.4
168: wtb 0;10
170: wrt 0,"CALCULATION OF TSAT FOR EACH LOCATION OF THERMOCOUPLE"
171: wtb 0;10
172: wrt 0,"LOCATION","","PRESSURE","","TEMPERATURE"
173: for J=1 to 13;J+4
174: J=4;J
176: T[40]+J+459.67;T[40]+J
177: fmt 4.3;x;f4.1;5.x;s8.4;3;5.x;s8.4
178: wrt 6.4;J;PC73;PC143;PC403
179: next J
For I=1 to 13

PC(I+14) = PC(I+14) + YC(I+14)

T(I) = T(I) + YC(I) + DC(I)

LI0D < YC(I) + DC(I) > YC(I) + YC(I) + DC(I)

VAPD < YC(I) + YC(I) + DC(I) > YC(I) + YC(I) + DC(I)

ENTV < YC(I) + YC(I) + DC(I) > YC(I) + YC(I) + DC(I)

HFG < YC(I) + YC(I) + YC(I) + YC(I) > YC(I) + YC(I) + YC(I) + YC(I)

TC(I) > YC(I) + YC(I) + YC(I) + YC(I)

CPLQ < YC(I) + YC(I) + YC(I) + YC(I) > YC(I) + YC(I) + YC(I) + YC(I)

KL < YC(I) + YC(I) + YC(I) + YC(I) > YC(I) + YC(I) + YC(I) + YC(I)

LIQD < YC(I) + YC(I) + YC(I) + YC(I) > YC(I) + YC(I) + YC(I) + YC(I)

next I

CALCULATION OF ALL THE FLOW RATES BEGINS

2.81297×10^13 - 20.21328×10^13

CORRECTED FREON SIDE MASS FLOW RATE

15.7725×10^13 - 17.4679×10^13

MASS FLOW RATE IN AFTER CONDENSER

DIMENSIONS FOR THE TUBE BEGINS NOW

INSIDE DIAMETER OF THE TUBE

LENGTH OF EACH SECTION

APPROX%: \text{APPROXIMATE HEAT INPUT IN BOILER}
```
240:  wrt 0;"HEAT TRANSFER CALCULATION BEGINS",
241:  wrt 0;"TERMINAL VOLTAGE "',V[1]
242:  wrt 0;"SHUNT VOLTAGE "',V[2]
243:  wrt 0;"CURRENT IN COPPER TUBE "',V[3]
244:  wrt 0;"HEAT TRANSFER IN WATTS "',Q[2]
245:  wrt 0;"HEAT TRANSFER IN EACH SECTION "',Q[4]
246:  wrt 0;"HEAT FLUX IN BTU/HR "',Q[3]
247:  QC23/AC33>QC33
248:  wrt 0;"HEAT FLUX IN BTU/HR FT**2 "',Q[3]
249:  QC33/12>QC43
250:  wrt 0;"CALCULATION FOR QUALITY BEGINS NOW"
251:  QC33/12>QC43
252:  wrt 0;10+10
253:  wrt 0;10
254:  wrt 0;"CALCULATION FOR QUALITY BEGINS NOW"
255:  QC33/12>QC43
256:  QC23-TC23>Y[3]
257:  wrt 0;"DEGREE OF SUBCOOLING "',Y[3]
258:  QC23/12>QC23
259:  for I=1 to 10;QC23>QC63
260:  if QC63>QC53>sto 263
261:  next I
262:  I-=1
263:  wrt 0;"SENSIBLE HEAT "',Q[5]
264:  wrt 0;"SECTIONAL HEAT "',QC[5]
265:  wrt 0;"NO OF SECTIONS USED FOR SUBCOOL "',I
266:  fix: 4
267:  fix: 4
268:  wrt 0;"SP HT",RC33
269:  for N=1 to I;fixXN=next N
270:  fix: 0
271:  12-I,J;fix: 4
272:  I+=J
273:  QC23>QC33>QC53
274:  QC53/MC1320+K3>Y[3]
275:  K=I,J
276:  for N=J to 13
277:  QC23/MC1320+J3>Y[N]
278:  next N
279:  I+=2 to 13
281:  next I
282:  fix: 8;3+3;f5.2+5x+8.3,5x+8.3
283:  wrt 0;"SECT","","*DX",","","*SECT X"
284:  for I=1 to 13
285:  fix: 6.8;I+193+X[I]+1+203
286:  next I
287:  12
288:  for I=1 to 5
289:  (I+13+I+1+23)/3>Y[1]+1+5;J=next I
291:  wrt 0;10
292:  wrt 0;"CALCULATION OF HEAT TRANSFER COEFF BEGINS NOW"
293:  QC13>QC13>QC13>QC13>QC13
294:  QC13+QC23+QC33+QC43+QC53+QC63+QC73+QC83+QC93+QC103
295:  QC13+QC13+QC13+QC13+QC13+QC13+QC13+QC13+QC13+QC13
296:  for I=1 to 13;fixX[I]+10;J=next I
297:  QC13/12>QC43
298:  wrt 0;"HEAT FLUX "',Q[4]
299:  for I=1 to 13
```
301: GC[1][2][1]+103H[1][2]
302: next I
303: wrt 0* "LOCATION","*"QUALITY","*"WALL T","*"DT WALL"
304: for I=1 to 13
305: for I=1 to 13
306: HC[1][2][2]=.6784H[1][2][2]
307: wrt 6.81XIXI+203GEI[1][2][1]+HI[1][2][1]+HI[1][3][2]
308: next I
309: end
310: dsp "DATA FILE STORAGE STARTS NOW"
311: ent 'FILE NAME??????*J
312: open G$j asan G$j
313: sprt 1,Q[*],L[*],MC[*],H[*],X[*],Y[*],K[*]
314: sprt l,C[*],T[*],V[*],P[*],F[*],RC[*],E[*],BC[*],D[*],A[*],"end"
315: prt "file name",G$j
316: dsp "SUBROUTINES FOR PROPERTIES OF R-113"
317: "TSAT":
318: if P[1]<=4.374;dsp •P<3.174"STOP
319: if P[1]<=10.07;ret 482.038164+10.17409P[1]-1.26531P[1]-459.6
320: if P[1]<=14.64;ret 496.903183+259.99974P[1]-1.125058P[1]-459.6
323: if P[1]<=59.49;ret 545.740363+2.861825P[1]-1.0423P[1]-459.6
324: if P[1]<=108.2;ret 578.073017+1.74081P[1]-1.00442P[1]-459.6
325: "DVISCO":
329: T[2]-459.6}T[2]
331: "VAPD":
335: if Y[2][2]<=29.6123Y[4][1]jmp 4
338: if Y[2][2]<=.269123Y[4][1]jmp 1
339: if Y[2][2]<=.1596123Y[4][1]jmp 0
340: if abs(P[6])<=.001jmp 2
342: r7/P[5]<>6
343: 0Y[7]j
344: if abs(P[6])<=.001jmp 2
345: Y[4][1]+6Y[4][1]jam &
346: ret Y[4]
347: "ENTV":
351: ret Y[4]
352: "HFG":
Electrically Heated Long Test Section
(R-22 as a refrigerant)

The data reduction programs for local single-phase and evaporation heat transfer coefficients were similar to that reported earlier (with R-113 as a refrigerant). However, the property subroutines were modified to accommodate R-22 as a test fluid. These subroutines are reported in the following section.
Water Heated/Cooled Long Test Section

(R-22 as a refrigerant)

Single-phase

0: \texttt{wrt 0,"COMPUTER PROGRAM FOR CALCULATING SINGLE-PHASE"}
1: \texttt{wrt 0,"HEAT TRANSFER COEFFICIENT"}
2: \texttt{dim A[5],BC7],C[60],D[20],H[40],F[4],G[5],N[5]
3: \texttt{dim L[8],MC7],P[10],QC8],R[5],T[80],U[10],V[18],W[20],Y[20],X[10]
4: \texttt{dim GSC103}
5: \texttt{dsp "SET THE PRINTER AT THE TOP OF THE PAGE"}
6: \texttt{>0}
7: \texttt{fmt l,'C*,fz2.0,*E*,fz2.0,"E*}
8: \texttt{fmt 3,"FIRITIZIMOPO*}
9: \texttt{fxd 4}
10: \texttt{dsp "TRIAL CHECK ON BULK TEMP"}
11: \texttt{"BULK":}
12: \texttt{ent "NO OF STEPS?for bulk temp",Z}
13: \texttt{for I=1 to Z;for J=1 to 2;J+3;K+19}
14: \texttt{wrt 722.3;red 722,E}
15: \texttt{"TEMP'(1000E);Y11J}
16: \texttt{dsp "BULK TEMP",J,Y11J;wait 1000;next J;next I}
17: \texttt{ent "WANT TO RUN AGAIN?yes 1,No 0",N}
18: \texttt{if N=1;sto "BULK"}
19: \texttt{dsp "CHECK ON SHELL SIDE BULK TEMP";wait 1000}
20: \texttt{"BULK"}
21: \texttt{ent "NUMBER OF STEPS?",Z}
22: \texttt{for I=1 to Z;for J=1 to 2;J+3;K+19}
23: \texttt{wrt 722.3;red 722,E;"TEMP'(1000E);Y11J}
24: \texttt{dsp "ANNULUS BULK",J,Y11J;wait 1000;next J;next I}
25: \texttt{ent "WANT TO RUN AGAIN?yes=1,No=0",N}
26: \texttt{if N=1;sto "BULA"}
27: \texttt{COOL:}
28: \texttt{ent "NO OF TRIALS FOR AFT COND?",Z}
29: \texttt{for J=1 to Z;for I=1 to 2}
30: \texttt{wrt 709.1;61;54+i;wrt 722.3;red 722,E}
31: \texttt{"TEMP'(1000E);Y11J}
32: \texttt{dsp "TEMP",34+1,Y11J;wait 1000;next J;next I}
33: \texttt{ent "WANT TO RUN AGAIN?yes=1,No=0",N}
34: \texttt{if N=1;sto "COOL"}
35: \texttt{HEAT:}
36: \texttt{ent "NO of trials for after cond bulk",N}
37: \texttt{for I=1 to N;for J=1 to 2}
38: \texttt{wrt 709.1;61;4+i;wrt 722.3;red 722,E}
39: \texttt{"TEMP'(1000E);Y11J}
40: \texttt{dsp "AFT COND",J,Y11J}
41: \texttt{wait 1000;next J;next I}
42: \texttt{ent "WANT TO CH AGAIN?yes=1,No=0",Z}
43: \texttt{if Z=1;sto "HEAT"}
44: \texttt{dsp "TRIAL RUN IS OVER"}
45: \texttt{wrt 0,"DATA RUN BEGINS NOW"}
46: \texttt{ent "TIME?",G$}
47: \texttt{wrt 0,"TIME",G$}
48: \texttt{wtb 0*10}
49: \texttt{dsp "DATA READOUT STARTS NOW"}
50: \texttt{wrt 0,"D.C.POWER INPUT BEGINS"}
51: \texttt{ent * enter the shunt voltage",VI2J}
52: \texttt{ent * TERMINAL VOLTAGE",VI1J}
53: \texttt{wrt 0,"TEMPERATURE READ OUT BEGINS"}
54: \texttt{for I=1 to 10;for N=1 to 24;NDC}
55: \texttt{wrt 709.1;61;4+i;wrt 722.3;red 722,E}
56: \texttt{CIN3=E;CIN3;next N;next I}
57: \texttt{wrt 0*","MILLIVOLTS","RANKINE"}
58: \texttt{wtb 0*10}
59: \texttt{for I=1 to 24}
60: CCI3/10>CCI3; 'TEMP'(1000CCI3)>CCI3; CCI3+459.6)>CCI3
61: next I
62: for I=1 to 24
63: CCI3/1000>CCI3
64: fmt 7": "f4.0;4x;r12.9;10x;r10.4
65: wrt 0. "INPUT FOR PRESSURES AND FLOW RATES BEGINS"
66: ent "FREON INLET PRESSURE", PC13
67: wrt 0. "FREON INLET PRESSURE" ",PC13
68: PC13=1505>PC53
69: wrt 0. "CORRECTED FREON PRESSURE", PC53
70: ent "OUTLET PRESSURE(CHECK) USING NASA", PC23
71: wrt 0. "OUTLET PRESSURE USING NASA", PC23
72: 1.0100886PC23-.130828>PC63; PC63+.3356>PC62
73: wrt 0. "CORRECTED OUTLET PRESSURE", PC63
74: ent "ATM PRESSURE(in of Ha", PC33
75: wrt 0. "ATMOSPHERIC PRESSURE,Psia", PC33
76: .01934PC33.4>PC33
77: wrt 0. "OUTLET PRESSURE,Psia", PC33
78: PC33=PC53; PC63+PC33>PC63
79: wrt 0. "INLET PRESSURE,Psia", PC63
80: wrt 0. "OUTLET PRESSURE,Psia", PC63
81: wrt 0. "FREON SIDE MASS FLOW RATE? in mv", FC13
82: wrt 0. "FREON SIDE MASS FLOW RATE, mv", FC13
83: ent "WATER SIDE MASS FLOW RATE, mA", FC23
84: wrt 0. "WATER SIDE MASS FLOW RATE, mA", FC23
85: ent "AFTER CONDENSER WATER SIDE MASS FLOW RATE, mm", FC3
86: wrt 0. "AFTER CONDENSER MASS FLOW RATE, mA", FC3
87: dsp "DATA SET COMPLETE"!wait 1000
88: wrt 0. "BULK TEMPERATURE CALCULATION BEGINS"
89: TC43=TC33; TC53=TC63
90: wrt 0. "Tube side bulk temperature", TC33
91: wrt 0. "Tube side inlet bulk temperature", TC43
92: wrt 0. "Tube outlet bulk temperature", TC53
93: wrt 0. "Tube outlet bulk temperature", TC63
94: wrt 0. "Shell inlet bulk temperature", TC11
95: wrt 0. "Shell inlet bulk temperature", TC12
96: wrt 0. "Shell outlet bulk temperature", TC13
97: wrt 0. "Shell outlet bulk temperature", TC14
98: dsp "CALCULATION OF DP BEGINS NOW"
99: wrt 0. "PRESSURE DROP CALCULATION BEGINS NOW"
100: PC53=PC63; PC83=PC63
101: PC53=PC63; PC83=PC63
102: wrt 0. "AVERAGE PRESSURE, Psia", PC83
103: wrt 0. "PRESSURE DROP, Psia", PC83
104: wtb 0;10
105: wtb 0;10
106: wtb 0;10
107: wtb 0;10
111: wrt 0. "FREON SIDE MASS FLOW RATE, lbm/hr", MC[13]
112: .0340478FC33+.01262799>MC33
113: MC33=MC33
114: wrt 0. "MASS FLOW RATE IN AFTER CONDENSER, lbm/hr", MC[13]
115: .5625FC23-.25>FC23
116: FC[23]=MC[23]
117: wrt 0. "Shell side mass flow rate, lbm/hr", MC[23]
118: wrt 0. "CALCULATION OF AVERAGE TEMPERATURES BEGINS"
124: wrt 0, "AVERAGE TUBE INLET TEMPERATURE", T[61]
125: wrt 0, "AVERAGE TUBE OUTLET BULK TEMPERATURE", T[62]
126: wrt 0, "AVERAGE INLET BULK TEMPERATURE (SHELL)", T[63]
127: wrt 0, "AVERAGE OUTLET BULK TEMPERATURE (SHELL)", T[64]
129: T[65] = T[65]
130: wrt 0, 10
131: wrt 0, "Tube side bulk temperature difference", T[65]
132: wrt 0, "Shell side bulk temperature difference", T[66]
134: wrt 0, "Average tube side bulk temperature", T[67]
135: wrt 0, "Average shell side bulk temperature", T[68]
136: T[67] = T[67]
137: wrt 0, "TUBE SIDE PROPERTY CALCULATION BEGINS NOW"
138: T[67] = Y[C2]
139: 'KL' (Y[C2]) = Y[C3] (Y[C3]) = Y[C4]
140: 'MUL' (Y[C2]) = B[C3] (B[C4]) = W[C1]
141: T[67] = Y[C2]
142: 'CPLQ' (Y[C2]) = Y[C3] (Y[C6]) = Y[C4]
143: wrt 0, "THERMAL COND", "VISCOSEITY", "SP. HT"
144: wrt 0, W[11], W[12], W[14]
145: wrt 0
146: wrt 0, "SHELL SIDE PROPERTY CALCULATION BEGINS NOW"
147: T[68] = Y[C1]
148: 'MUELW' (Y[C1]) = Y[C3]
149: Y[C3] = W[C1]
150: 'KLW' (Y[C1]) = Y[C3]
151: Y[C3] = W[C3]
152: wrt 0
153: wrt 0, "LIQUID VISCOSITY", "THERMAL CONDUCTIVITY"
154: wrt 0, W[C1], W[C2]
155: wrt 0
156: wrt 0, "DIMENSIONS FOR THE TUBE BEGINS NOW"
161: wrt 0, "INSIDE DIAMETER OF THE TEST TUBE", L[1]
162: wrt 0, "OUTSIDE DIAMETER OF THE TUBE", L[2]
164: wrt 0, "OUTSIDE DIAMETER OF THE ANNULUS TUBE", L[5]
165: wrt 0, "LENGTH OF THE TEST SECTION", L[3]
166: wrt 0, "AREA OF CROSS SECTION", A[1]
171: wrt 0, "Hydraulic diameter", L[6]
172: wrt 0, 10
173: wrt 0, "HEAT TRANSFER CALCULATION BEGINS"
174: wrt 0, "TERMINAL VOLTAGE", V[C1]
175: wrt 0, "SHUNT VOLTAGE", V[C2]
177: wrt 0, "CURRENT IN COPPER TUBE", A[C2]
178: wrt 0, "HEAT TRANSFER IN WATTS", Q[C2]
TUBE SIDE HEAT TRANSFER CALCULATION BEGINS

FREON SIDE HEAT TRANSFER

ANNUAL SIDE HEAT TRANSFER CALCULATIONS

WATER SIDE HEAT TRANSFER, BTU/hr

HEAT BALANCE CALCULATION BEGINS

HEAT BALANCE (FREON BASE) 0

HEAT BALANCE (WATER BASE) 0

LMTD OF THE TEST SECTION

OVERALL HEAT TRANSFER COEFFICIENT

SHELL SIDE HEAT TRANSFER COEFFICIENT CALCULATION BEGINS

SHELL SIDE REYNOLDS NUMBER

SHELL SIDE PRANDTL NUMBER

SHELL SIDE NUSSELT NUMBER

HEAT TRANSFER COEFFICIENT ON SHELL SIDE

HEAT TRANSFER COEFFICIENT ON TUBE SIDE

TUBE SIDE HEAT TRANSFER COEFFICIENT CALCULATION BEGINS

TUBE SIDE NUSSELT NUMBER

NU-PR PARAMETER

DITTUS BOELTER EQUATION CALCULATION BEGINS NOW
387

240: wrt 0,*"RE NO","*"NU-PR PARAMETER"*
241: wrt 0,"RC13,NC33"
242: dsp 'DATA STORAGE BEGINS NOW''
243: ent 'FILE NAME?????"G$'
244: 38>J
245: open G$;
246: assn G$,1,0
247: sprt 1,A[*],B[*],C[*],D[*],H[*],F[*],GC*,N[*]
248: sprt l,L[*],M[*],P[*],QC*,R[*],T[*],U[*],VC*,X[*],'end'
249: prt "file name"
250: end
251: dsp "SUBROUTINES FOR CALCULATING PROPERTIES OF"
252: dsp 'R-22 AND WATER''
253: 'TSAT':
254: if pl<=4.374rdsp "P<3.174'fstp
255: if pl<=109.02fret -36.847862+1.15571958p
256: if p1<=136.12;ret -23.17677+.8948589
257: if pl<=274.6;ret 3.772544+.56767401p
258: if pl<=497.26fret 38.4089+.3486329
259: if pl<=396.19;ret 22.74548+.4294088
260: if pl<=260;1.11782+1.34991e-4pl-8.0798e-6p2+3.03989e-8p3>YC63
261: "VAPF":
262: YC23-459.6+32.2
263: if YC23<=39;ret .74169319+,0131801657Y23+.000157388Y23^2
264: if YC23<=49;ret .778394+.0112198Y23+.0001837124Y23^2
265: if YC23<=39;ret .555261+.009028Y23+.000287197Y23^2
266: "HFG":
267: "HVAP":
268: "CPLQ":
269: YC23-459.6-32.2>YC23iYC23>1.8+459.6>YC23
270: ret YC83
271: "MUL":
272: YC23-459.6-32.2>YC23iYC23>1.8+459.6>YC23
300: 2.41928[C3]BC33
301: Y[C2]-273.3>Y[C2]+Y[C2]+1.8+32.2+459.6>Y[C2]
302: ret BC33
303: "LIQD":
304: 1-.001505Y[C2]>p1
305: p1-.3333>p2>p1-.6667>p3>p1-.3333>p4
306: 32.76+54.634409x2+36.74992x3-22.2925657x4+20.4732886x4>Y[C3]
307: ret Y[C3]
308: "MUEL":
309: YE13-32.2-459.6>YE13;YE13/1.8>YE13;YE13+273.3>YE13
310: YE13-32.2
313: exp(Y[C3]>Y[C3]
314: 2.419388[Y[C3]>Y[C3]
315: YE13-273.3>YE13;YE13/1.8>YE13;YE13+32.2+459.6>YE13
316: ret Y[C3]
317: "LIQW":
318: YE13-32.2
319: ret YE13
320: "KLW":
321: YE13-459.6>YE13;YE13-32.2>YE13;YE13/1.8>YE13
322: YE13+273.3>Y[C1]
323: -.616947.1851e-3Y[C1]-1.167e-5Y[C1]+2.70358e-9Y[C1]-3Y[C2]
324: .5774Y[C2]>Y[C2]
325: YE13-273.3>YE13;YE13/1.8>YE13;YE13+32.2+459.6>YE13
326: ret Y[C2]
327: "TEHP":
328: if P[1]<1.494fret 31.99925+46.80117p1-1.40739e-1+0.0780e-1=3-.00739e-1=4
329: if P[1]<3.94fret 31.42956+44.488e35-1.0742+2-.253895e1=3+.0287e1=4
330: if P[1]<6.621fret 33.82822+45.39092p1-1.015078p1-1.015078p1-1.015078p1-1-0.0642p1=4
Evaporation

01 DSP "COMPUTER PROGRAM FOR CALCULATING EVAPORATION HEAT"
1: DSP "TRANSFER COEFFICIENT USING R-22"
2: DIM AC53, BC73, CC603, DC403, H[50], F[6], G[5], RC103
3: DIM L[8], M[7], NC103, PC103, UC103, Q[10], T[80], V[12], W[20], Y[20], X[10]
4: DIM G[10]
5: DSP "SET THE PRINTER AT THE TOP OF THE PAGE"
6: 630
7: FMT 1, "C", F2.0, 'E', F2.0, 'E'
8: FMT 3, "F1R1T1Z1MOPO"
9: FXT 4
10: DSP "TRIAL CHECK FOR WALL TEMP"
11: "BOIL*"
12: FOR I=1 TO Z FOR J=1 TO 5; I=25+J; K=I+19
13: WRT 722.3; RED 722; E
15: DSP "WALL TEMP*" K; Y[11]; WRT 1000; NEXT J; NEXT I
16: ENT "WANT TO RUN AGAIN? YES=1, NO=0" N
17: IF N=1, "BOIL*"
18: DSP "TRIAL CHECK ON BULK TEMP"
19: "BULK*"
20: ENT "NO OF STEPS FOR BOILER TEMP", Z
21: FOR I=1 TO Z FOR J=1 TO 5; I=25+J; K=I+19
22: WRT 722.3; RED 722; E
24: DSP "BULK TEMP*" J; Y[11]; WRT 1000; NEXT J; NEXT I
25: ENT "WANT TO RUN AGAIN? YES=1, NO=0" N
26: IF N=1, "BULK*"
27: DSP "CHECK ON SHELL SIDE BULK TEMP"; WAIT 1000
28: "BULA*"
29: ENT "NUMBER OF STEPS?'" Z
30: FOR I=1 TO Z FOR J=1 TO 5; I=25+J; K=I+19
31: WRT 722.3; RED 722; E
33: DSP "ANNULUS BULK*" J; Y[11]; WRT 1000; NEXT J; NEXT I
34: ENT "WANT TO RUN AGAIN? YES=1, NO=0" N
35: IF N=1, "BULA*"
36: "COOL*"
37: ENT "NO OF TRIALS FOR AFT COND?" Z
38: FOR J=1 TO Z FOR I=1 TO 2
39: WRT 709.1; 61; 15+I; WRT 722.3; RED 722; E
41: DSP "TEMP*" J; Y[11]; WRT 1000; NEXT J
42: ENT "WANT TO RUN AGAIN? YES=1, NO=0" N
43: IF N=1, "COOL*"
44: "HEAT*"
45: ENT "NO OF TRIALS FOR AFTER COND BULK" N
46: FOR I=1 TO N FOR J=1 TO 2
47: WRT 709.1; 61; J*25; WRT 722.3; RED 722; E
50: WAIT 1000; NEXT J; NEXT I
51: ENT "WANT TO CHECK AGAIN? YES=1, NO=0" Z
52: IF Z=1, "HEAT*"
53: DSP "TRIAL RUN IS OVER"
54: WRT 0; "DATA RUN BEGINS NOW"
55: ENT "TIME?" G$
56: WRT 0; "TIME" R$
57: WTB 0; 10; 10
58: DSP "DATA READOUT STARTS NOW"
59: WRT 0; "D.C. POWER INPUT BEGINS"
•S3

ent • enter the shunt voltage".VC[2]

ent "TERMINAL VOLTAGE".VC[1]

wrt 0,"SUPER HEATER POWER INPUT BEGINS"

wrt 0,"SUPER HEATER VOLTAGE".VS[5]

wrt 0,"SUPER HEATER CURRENT + AMPS".VC[6]

wrt 0,"TEMPERATURE READOUT BEGINS"

for I=1 to 10
for N=1 to 24
N>C

PC[33]>PC53
PC53-tPC63>PE83?

wrt 0,"AVERAGE PRESSURE, Psia".PE83

for 1=1 to 1

PC83>YC[13]

dsp "CALCULATION OF DP BEGINS NOW"

wrt 0,"PRESSURE DROP CALCULATION BEGINS NOW"

wrt 0,"AVERAGE PRESSURE, Psia".P[8]

wrt 0,"PRESSURE DROP, Psia".P[7]

wrb 0+10

wrt 0,"CALCULATION OF FLUID PROPERTIES BEGINS"

for I=1 to 1

PC[8][3][I]+P[8][3][I]

for 1=1 to 24

CCN3/10>CCN3

wrt 0,* MILLIVOLTS*,"."RANKINE"
120: 'TSAT'(Y13)>Y12
121: Y12+459.6>Y12
122: Y12+5>Y12
123: 'LIQD'(Y12)>Y13;Y13>Y12
124: 0>Y13
125: TC13+5>Y13
126: 'VAPD'(Y12)>Y12;Y12+5>Y13
127: TC13+5>Y12
128: 'HFG'(Y12)>Y12;Y12+5>Y13
129: HI13+0>Y13
130: next I
131: 'KL'(Y12)>Y12;Y12+5>Y13
132: TC26+5>Y12;Y12+5>Y13
133: wtb 0,10,10
134: wtb 0,"LIST OF ALL THE PROPERTIES"
135: wtb 0,10,10
136: fmt 5,3x,f4.1,2%,f8.3,4x,f8.4
137: fmt 6,3x,5x,f8.3,4x,f8.4
138: for I=1 to 1
139: wtb 0,10
140: wtb 0,10
141: wtb 0,10
142: wtb 0,10
143: next I
144: wtb 0,10
145: wtb 0,"CALCULATION OF ALL THE FLOW RATES BEGINS"
146: TC26+5>Y12
147: 2FC13+5>FC13; 'LICiri'(Y12)>Y13;Y13>Y12
148: 8.01DC13+5>DC13+5>MC13+5
149: wtb 0,"CORRECTION FACTOR FOR THE ROTAMETER",MC13
150: wtb 0,"FREON SIDE MASS FLOW RATE, Ibm/hr",MC13
151: .0340478FC33+.01026279>MC33
152: MC33500MC33
153: wtb 0,"MASS FLOW RATE IN AFTER CONDENSER, Ibm/hr",MC33
155: FC23500>MC23
156: wtb 0,"Shell side mass flow rate, Ibm/hr",MC23
157: wtb 0,"CALCULATION OF AVERAGE TEMPERATURES BEGINS"
158: TC43+5>TC613;STC613+5>TC613
159: TC613+5>TC613+5>TC623
160: TC13+5>TC13+5>TC643+5
161: TC613+5>TC13+5>TC633+5
162: wtb 0,"AVERAGE TUBE INLET TEMPERATURE",TC613
163: wtb 0,"AVERAGE TUBE OUTLET BULK TEMPERATURE",TC613
164: wtb 0,"AVERAGE INLET BULK TEMPERATURE(SHELL)",TC613
165: wtb 0,"AVERAGE OUTLET BULK TEMPERATURE(SHELL)",TC613
166: TC613-TC623+5>TC653+5
167: TC623+5>TC623+5>TC663+5
168: wtb 0,10,10
169: wtb 0,"AVERAGE TUBE SIDE BULK TEMPERATURE DIFFERENCE",TC653+5
170: wtb 0,"AVERAGE SHELL SIDE BULK TEMPERATURE DIFFERENCE",TC663+5
171: wtb 0,"Average temperature",TC613+5
172: wtb 0,"Average Shell side bulk temperature",TC613+5
173: wtb 0,"SHELL SIDE PROPERTY CALCULATION BEGINS NOW"
174: TC682+5
175: 'MUEUW'(Y13)>Y13
176: Y13+5>Y13
177: 'KLU'(Y13)>Y13
YCI3>WC23
180: wrt 0
181: wrt 0
182: wrt 0
183: wrt 0
184: wrt 0
185: wrt 0
186: wrt 0
187: wrt 0
188: wrt 0
189: wrt 0
190: wrt 0
191: wrt 0
192: wrt 0
193: wrt 0
194: wrt 0
195: wrt 0
196: wrt 0
197: wrt 0
198: wrt 0
199: wrt 0
200: wrt 0
201: wrt 0
202: wrt 0
203: wrt 0
204: wrt 0
205: wrt 0
206: wrt 0
207: wrt 0
208: wrt 0
209: wrt 0
210: wrt 0
211: wrt 0
212: wrt 0
213: wrt 0
214: wrt 0
215: wrt 0
216: wrt 0
217: wrt 0
218: wrt 0
219: wrt 0
220: wrt 0
221: wrt 0
222: wrt 0
223: wrt 0
224: wrt 0
225: wrt 0
226: wrt 0
227: wrt 0
228: wrt 0
229: wrt 0
230: wrt 0
231: wrt 0
232: wrt 0
233: wrt 0
234: wrt 0
235: wrt 0
236: wrt 0
237: wrt 0
238: wrt 0
239: wrt 0
In the test section, the change in quality is monitored to determine the degree of subcooling. The LMTD (Logarithmic Mean Temperature Difference) calculation begins now.

The overall heat transfer coefficient calculation begins.

The shell side heat transfer coefficient calculation begins.

The tube side heat transfer coefficient calculation begins.

A summary of the experimental run is presented.

The data storage begins now.

The file name is saved.
394

300: end
301: wrt 0:*SUBROUTINES FOR PROPERTY CALCULATIONS OF R-22 AND WATER*
302: 'TSAT':
303: if p1<=4.374;dsp •P<3.174"îstp
304: if p1<=109.02;ret -36.847862+1.15571958p1-.0027931125p1p1
305: if p1<=136.12;ret -23.17677+.8948589p1-.001545557p1p1
306: if p1<=183.09;ret -12.12965+.7363052p1-.000975339p1p1
307: if p1<=274.6rret 3.772544+.56769401p1-.0005265296p1p1
308: if p1<=396.19rret 22.74548+.42940885p1-.0002733p1p1
309: if p1<=497.26?ret 38.4089+.3486329p1-.0001695p1p1
310: 'VAPD':
311: YC23-459.6>YC23
315: fxd 4
316: 'HFG':
317: YC23-459.6>YC23
318: fxd 4
319: Yi:i32.302585093>pi;686..1-YC23>p6f3.414/YC23>p7
320: loa(p6)/YC23YC2D>p8î686.1p8>p8
321: 1/YC43-1/YC33>P5
322: .434294/YC23+p3
323: .185053YC23>5YE99
324: YC23|Y23>p2
325: p1(384.193152/p2-p7+2.190939e-3-.445746703p3)P4
326: YC9j40/YC93
327: ret YC93
328: 'HUP':
329: HC13=HC33+HC23
330: ret HC23
331: 'CFLQ':
332: YC23-459.6-32.2>YC23;YC23/1.8>YC23+32.2>YC23
335: YC23-273.3|Y[2];|Y231.8+459.6+32.2|Y[2]
336: ret YC23
337: ret YC23
338: *CPUP':
339: YC23-459.6|YC23-32.2>YC23;YC23/1.83|YC23+273.3|YC23
342: YC23-273.3|Y[2];|Y231.8+459.6+32.2|Y[2]
343: ret YC23
344: *'KL'::
346: .57789(.1001-.000495|Y[2])|Y[2]
347: YC23.3|Y[2];|Y231.8+459.6+32.2|Y[2]
348: ret YC83
349: *'HUL':
350: YC23-459.6-32.2>YC23;YC23/1.83|YC23+273.3|YC23
354: YE23-273.3|YE23+32|YE23+459.63|YE23
355: ret YE83
356: *'LLOQ':
357: 1-.001505|Y[2]=1
358: p1+3.3333|p2+1.66667p3+p1+1.33333p4
360: ret Y[3]
361: *MUEL*:  
366: exp(Y[3])=Y[3]  
369: ret Y[3]  
370: *LIOW*:  
371: 62.4*Y[1]  
372: ret Y[1]  
373: *KLW*:  
375: Y[1]=327.3*Y[1]  
379: ret Y[2]  
380: *TEMP*:  
381: if P1<=1.494 ret 31.99925+46.80117*P1-1.407394*P1^2+0.07802*P1^3-0.007394*P1^4  
382: if P1<=3.941 ret 33.4295+44.48835*P1-0.07422*P1^2+0.253895*P1^3+0.02878*P1^4  
383: if P1<=6.621 ret 33.8222+45.39092*P1-1.01578*P1^2+0.03592*P1^3-0.00642*P1^4
Condensation

The data reduction program for computing the condensation heat transfer coefficients was similar to that reported earlier for evaluating evaporation heat transfer coefficients. Details of the data reduction procedure are reported in Chapter VI.