2004

Altered Bi-Phase Flow Regime in Evaporative Coils

David A. Wightman
XDX Innovative Refrigeration LLC

Bernard Wendrow
Consultant

Richard S. Sweetser
Exergy Partners Corp.

William M. Worek
Energy Resources Center (UIC)

Altered Bi-phase Flow Regime in Evaporative Coils

David A. Wightman[1], Bernard Wendrow, PhD, P.E[2], Richard S. Sweetser[3], William M. Worek, PhD, MS. [4]

[1]XDX Innovative Refrigeration LLC, CEO
Arlington Heights, IL USA
847.398.0250 Tel, 847.398.1365 Fax, dawightman@xdxusa.com

[2]XDX LLC, Consultant
Arlington Heights, IL USA
847.398.0250 Tel, 847.398.1365 Fax, bwendrow@xdxusa.com

[3] EXERGY Partners Corp., President
Herndon, VA USA
703.707.0293 Tel, 703.707.0138 Fax, rsweetser@exergypartners.com

[4] Director of the Energy Resources Center of Illinois
Professor and Head of Department of Mechanical and Industrial Engineering, UIC
Chicago, IL USA
312.996.5610 Tel, 312.996.5620 Fax, wworek@uic.edu

ABSTRACT

Field tests of several refrigeration systems operating with thermal expansion valves have recently been documented and compared in systems using a High Vapor Faction and Turbulent (HVFT) flow in an Altered Bi-phase Flow regime (ABF). The results demonstrated improved heat transfer coefficients with redistribution of frost on the evaporator coils permitting longer periods between defrost, and reduction in system-wide energy usage. It was theorized that a HVFT (High Vapor Fraction and Turbulent) entering the evaporator coil could have the effect of more evaporator work through improvement in the flow regime than if entering with a greater liquid percentage, despite the associated enthalpic capacity. Since this theory was from different perspective than the last fifty years of industry work, and due to the counter intuitive nature of the theoretical principles, visualization of Altered Bi-phase Flow (ABF) regime was studied through the use of a glass tube evaporator model.

1. INTRODUCTION

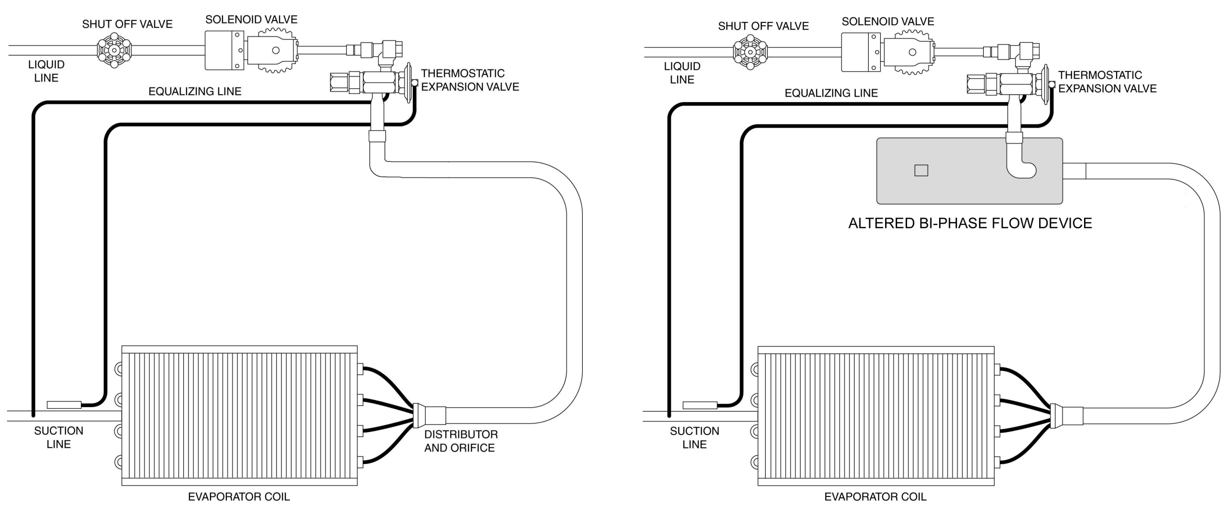
Research over the past few years in bi-phase fluid flow visualization entering refrigerant-to-air evaporator coils fostered interest in testing the affects of radically different liquid/vapor fractions in combination fundamentally changing refrigerant flow regimes. Experiments, some of which were the focus of previous ASHRAE and IIR work, studying and documenting this alteration of a vapor fraction and flow regime of entering refrigerant to the evaporators have been completed and reported over the past two and a half years, caused the authors to take the experiment toward visualization.

A laboratory project was designed to demonstrate a flow visualization using a glass tube evaporator to better understand the bi-phase refrigerant flow characteristics of conventional properly adjusted TXVs and the HVFT and ABF flow regime in a system using Dry expansion technology, or "direct expansion" by which it is commonly referred. Refrigerant feed through thermal expansion devices is widely recognized as providing limited control over coil performance. A thermal expansion valve modulates flow based upon the refrigerant temperature at the evaporative outlet line by translating this temperature into pressure (through the bulb and capillary tube) and then to a mechanical force that operates valve in a relationship with evaporative pressure. The difference in these pressures or forces is used to determine superheat at the outlet of an evaporative coil. This method modulates refrigerant flow in a "hunting manner" overfeeding refrigerant, underfeeding refrigerant and then overfeeding the evaporative coil again, which is more pronounced in transient modes of operation. Such operation provides limited control typically with a great deal of fluctuation in the refrigerant flow properties at the outlet of the evaporative coil. What seems to have been overlooked is the substantial impact this erratic flow pattern has on the heat transfer coefficient over the entire evaporator and more specifically that the quality of the refrigerant in the inlet portion of the coil can affect the heat transfer throughout the entire coil.

2. ABF/HVFT FLOW REGIME

Each test essentially consisted of the same setup. Comparative testing was performed between two systems designs. The first system setup uses a conventional thermal expansion valve (Baseline System) installed following all manufacturer specified recommendations. The second system combines thermal expansion valve in conjunction with the ABF/HVFT Flow, that varies the vapor fraction of the refrigerant and creates turbulent flow through a mechanically induced fluid process (ABF/HVFT System). Each test and verification project is monitored to measure the effect that this improved flow regime can have on cooling rates, compressor work, temperature differences, evaporator efficiency, control of superheat, and in other significant observations. (Figure 1)

Figure 1: Baseline System and ABF/HVFT System Test Schematics



Baseline **ABF/HVFT System**

The hypothesis being tested is that entering an evaporator coil with a high vapor faction enables the novel and highly efficient flow regime to be achieved throughout the evaporative coil. Furthermore, annular flow at the outlet of the evaporator coil in the ABF/HVFT System communicates very efficiently with the superheat sensing bulb, whereas the conventional vapor barrier of superheat in Baseline has extremely poor heat transfer and cannot communicate well.

The ABF/HVFT System has been operated throughout the study in multiple and single pass circuiting, air/ventilated and gravity feed coils, and high, medium and low temperature applications. Reduction in superheat exiting from the evaporator coil is accomplished with minimal liquid and is very tightly controlled with little fluctuation as verified on the glass tube evaporator test stand. Lower superheat can mean greater surface exposure to the refrigerant and result in higher evaporator pressures. Lower superheat allows for a denser refrigerant, boosting compressor capacity and lowering compressor outlet superheat. Energy is reduced in each case.

Evaporator temperature uniformity allows for frost to build more uniformly across the coil and can therefore reduce defrost frequency or duration by not causing a restriction in air-side velocity.

3. LABORATORY TEST & VERIFICATION OF EVAPORATOR FLOW REGIME

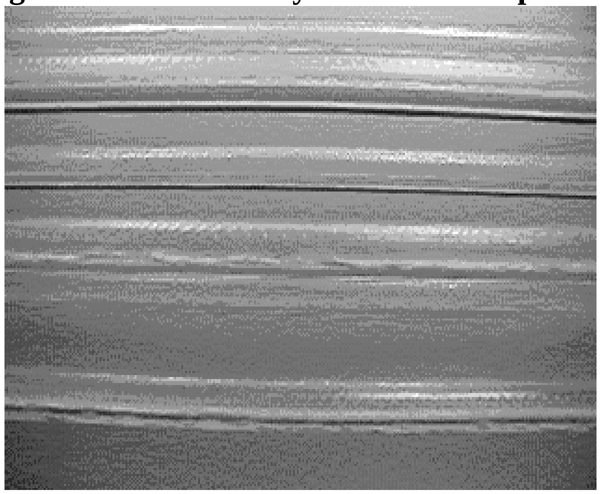
A glass tube evaporator test stand was constructed using heat-treated and pressure tested glass tubing and consisted of 32 passes of 19 mm (¾ in) O.D, 3.12 mm (0.123 in) wall thickness, 12.7 mm (½ in) I.D. by 1,219 mm (48 in) in length. The passes were connected using compression fittings and copper "U" bends. The purpose of this laboratory test stand was to try to explain the increased evaporator performance found in field tests through a better understanding of the bi-phase flow characteristics throughout the evaporators when switching from Baseline TXV operation to ABF/HVFT System operation.

The glass tube evaporator test stand was operated with constant conditions using R12 and measured conventional evaporator capacity ~ 1,120 watts 3,840 BTUH at steady state. In the conventional mode of operation a traditionally sized TXV operated at a targeted 4.4°C (8°F) superheat, and operated at a room temperature 22.2°C (72°F) and 45% RH. The evaporator coil was monitored[i] visually for identifiable flow regimes, liquid flow patterns of advancement and recession, suction pressure and for evaporator coil surface temperature. Coil surface temperature was monitored at inlet, the midway point and at the outlet. The evaporator coil surface temperature was measured by electronic thermocouple and with infrared thermometer for comparative purposes. All coil temperature measurements were made at each of the two o'clock and four o'clock positions. Readings were recorded after operation had stabilized for a period of one hour, and the readings shown (Table 1) are representative of the series of readings taken. The large diameter of the evaporator clearly demonstrated the impact of the flow differences. The glass composition of the tube wall substantially impacted temperature readings but presented greater differentials than anticipated. This pronounced differences between the flow regimes to the benefit of the researchers. Work is presently underway to provide greater instrumentation with higher detail to sampling, with emphasis given to heat transfer comparisons between the glass and a common copper tube wall construction.

The glass tube was used to confirm theoretical views of the flow regimes in each of the pre-retrofit and post-retrofit operations. Stratified–Wavy Flow was most commonly witnessed at the inlet of the evaporative coil in the pre-retrofit mode (Baseline System). Slug-Flow was occasionally seen. The Stratified-Wavy flow ran consistently through the initial eight to ten passes of the evaporator coil evaporator, with the refrigerant becoming higher in vapor percentage as it progressed through the coil. Intermittent and Annular Flow could occasionally be seen during brief periods toward the tenth to twelfth passes, and the partial dry-out was witnessed occasionally over about two passes of the evaporator and was directly effected by the advancing and receding point of furthest liquid feed, which had a movement of about 1.2 m (4 ft). The balance of the evaporative coil had no visible liquid present and was measured to be superheated vapor. Occasionally, during transient operation, liquid in varying levels would ride the bottom to bottom third of the tubing in fluctuation periods and extend well into the superheated area.

When operated in the post-retrofit mode (ABF/HVFT System), the inlet 2 to 4 passes were in Intermittent-Flow, and gave indication frequently of Wavy Flow. The third through twenty-ninth passes of the evaporator were primarily in Annular Flow with the last two passes of the evaporator coil in either Annular or Annular with Partial Dry-out. (Figure 2). Superheat was increased so as to visualize the advancement and recession of the point of furthest liquid feed. Following this adjustment, the last pass of the evaporator became superheated. The point of furthest liquid feed fluctuated between 38.1 to 50.8 mm (1.5 to 2 inches) as compared to 1.2 m (4 ft) using a conventional TXV in the pre-retrofit operation.

Figure 2 ABF/HVFT System Glass Evaporator



Other significant differences were measured in the surface readings of the glass tubing. The evaporator pressure, discharge temperature, inlet evaporator tubing surface reading at 2 and 4 o'clock, midpoint evaporator tubing surface reading at 2 and 4 o'clock, and outlet evaporator tubing surface reading at 2 and 4 o'clock were recorded. Note the fluctuation in temperatures (Table 1) in the Baseline with relation to the coil outlet, as the TXV hunted. The difference between the 2 o'clock and 4 o'clock readings indicates the absence of liquid at the upper portion of the tubing. The second reading indicates the reaction of the expansion valve, in that inlet coil temperatures begin to change to reduce the superheat at the outlet, during which time instantaneous infrared temperature measurements on the copper line feeding the evaporator indicated fluctuating degrees of sub-cooled liquid (.28C/.5F to 2.78C/5F). A third reading indicates a throttling back of the refrigerant feed to react to the now colder outlet.

Table 1: Baseline TXV Fed Glass Evaporator

evaporator pressure	discharge temperature	inlet evaporator tubing surface reading at 2 o'clock	inlet evaporator tubing surface reading at 4 o'clock	midpoint evaporator tubing surface reading at 2 o'clock	midpoint evaporator tubing surface reading at 4 o'clock	outlet evaporator tubing surface reading at 2 o'clock	outlet evaporator tubing surface reading at 4 o'clock
kPa (psi)	°C (°F)	°C (°F)	°C (°F)	°C (°F)	°C (°F)	°C (°F)	°C (°F)
110.3 (16)	58.89 (138)	2.83 (37.1)	0.61 (33.1)	0.56 (33)	-2.89 (26.8)	8.50 (47.3)	8.39 (47.1)
113.8 (16.5)	60.3 (141)	-6.39 (20.5)	-6.67 (20.0)	.011 (32.02)	-4.13 (24.57)	0.04 (32.0)	-3.26 (26.13)
111.5 (16.25)	59.44 (139)	2.944 (37.3)	0.28 (32.5)	1.06 (33.9)	-2.56 (27.4)	0.94 (33.7)	-0.22 (31.6)

The ABF/HVFT System fed glass evaporator with a distributed enthalpy and with low degree superheat is as indicated in Table 2. Note the uniformity between the 2 o'clock and 4 o'clock readings indicating the presence of a liquid film at the upper portion of the tubing. The disparity between the 2 o'clock and 4 o'clock readings at the outlet of the coil is indicative of partial dry-out. The ABF/HVFT Flow glass evaporator is then shown with the slightly increased superheat as indicated for the visualization of liquid line advancement. Note the uniformity between the 2 o'clock and 4 o'clock readings indicating the absence of a liquid film at the upper portion of the tubing at the coil outlet. Temperatures throughout the balance of the coil remained uniform as annular flow was visualized. The higher suction pressure is consistent with our expectations, being higher due to better utilized surface area, and as the glass tubing impacted heat transfer.

Table 2 VIB Flow Regime Device Fed Glass Evaporator

evaporator pressure	discharge temperature	inlet evaporator tubing surface reading at 2 o'clock	inlet evaporator tubing surface reading at 4 o'clock	midpoint evaporator tubing surface reading at 2 o'clock	midpoint evaporator tubing surface reading at 4 o'clock	outlet evaporator tubing surface reading at 2 o'clock	outlet evaporator tubing surface reading at 4 o'clock
kPa (psi)	°C (°F)	°C (°F)	°C (°F)	°C (°F)	°C (°F)	°C (°F)	°C (°F)
182.7 (26.5)	47.78 (118)	2.33 (36.2)	1.72 (35.1)	4.06 (39.2)	2.22 (36)	3.61 (38.5)	1.22 (34.2)
186.5 (27)	49.44 (121)	6.72 (44.1)	5.89 (42.6)	46.1 (40.3)	3.61 (38.5)	2.67 (36.8)	2.33 (36.3)

In ongoing work, a few observations will receive additional study. Discharge temperatures and pressures appear to be benefited by the ABF/HVFT Flow. As noted earlier, one of the observations during the visualization was of the impact of vapor refrigerant upon the sensing bulb, and that the communication of the vapor refrigerant to the bulb was very poor and slowed, causing delayed response during the transient operation of the baseline. The distributed enthalpy affect and the extension of the partial dry-out regime toward the bulb in the form of a liquid film during the ABF/HVFT Flow provided better communication to the bulb and minimized feed fluctuation greatly. Another significant observation was that during the baseline operation, refrigeration oil was gradually logged near the inlet of the evaporator. The ABF/HVFT Flow regime prevented this oil from building.

Additionally, an observation that will be examined in ongoing work will be the formation of frost, as it formed quite differently when the two systems were compared. The ABF/HVFT Flow provided significantly uniform formation when conditions were allowed to cause frost formation. This observation was not the focus of this paper.

4. ANALYTICAL MODEL

In effort to quantify the benefit seen in increasing the vapor fraction within the evaporator, especially since the reduction in enthalpic capacity in light of increased performance can be counter intuitive, the project was undertaken to create a theoretical model in which several variables are assumed to be constants for sake of calculation. An evaporator will be calculated for the length necessary to change vapor faction. R-134a in a straight length of tube will be used will be used due to the extensive foundation of data provided in recent technical works.

Flow characteristics and heat transfer coefficients will be adapted over the length of the evaporator. The evaporator will have constant temperature, as well as constant return air temperature. Also assumed for sake of making the point firmly, that entry to this theoretical evaporator is 2.78°C (5°F) sub-cooled. Superheat will be calculated for the baseline model at 5.56°C (10°F).

In the work, Flow Boiling in Horizontal Tubes, 1998, the best heat transfer coefficients for R-134a were calculated between thirty percent (30%) and ninety percent (90%) vapor fraction, with the heat transfer coefficient dropping off dramatically at ninety-eight percent (98%) vapor. To pursue this thought fully, VIB flow regime device fed evaporator would therefore best calculated to have at entry thirty percent (30%) vapor at a mass velocity indicative of annular flow and would sustain the flow regime throughout the evaporator, exiting without superheat, but rather a ninety-eight percent (98%) vapor faction in annular flow with partial dry-out. The evaporative coil is thereby transferring heat at its maximum and performing its work over the maximum length of tubing.

Theoretical Evaporator Model Assumptions

- **Evaporating Temperature t_1 = 4.44°C (40°F)**
- **Return Air Temperature 18.33°C (65°F)**
- **Refrigerant 134a**
- **Sub-cooling 2.78°C (5°F)**
- **Superheat 5.56°C (10°F)**
- **Mass Flow 1.13 kg/min (2.5 lb/min)**
- **Psat = 341.09 kPa (49.471 psia)**

Calculations for Evaporator Tube Length

Since is sub-cooling of 5°F was experienced in the glass evaporator test, assume the liquid R-134a is sub-cooled 5°F:
Temp = 40°F - 5°F = 35°F
Enthalpy $h_1^{35°F}$ = 23.274 Btu/lb. @ 35°F
Enthalpy $h_1^{40°F}$ = 24.890 Btu/lb. @ 40°F

$$Q_{sub\text{-}cool} = 2.5\,(24.890 - 23.274) = 4.04 \text{ Btu/min heat required to remove sub-cooling} \quad (1)$$

To find the length of evaporator tubing needed to supply the sub-cooling heat, use the equation

$$Q = H_1 A_1 \,(?t) \quad (2)$$

H_1 is the inside liquid heat transfer coefficient. A is the inside area of the evaporator tube in square feet per foot of tubing length, and ?t is the log mean temperature difference between the air stream and the refrigerant.

The inside heat transfer coefficient is found from the Dittus and Boelter equation and is equal to 72 Btu per hour per sq. Ft. Per degree F. (408.9 W/m^{2}K). The inside area of an 11mm ID tube is 0.11338ft^2/ft of length (0.03456m^2/m). The following diagram (Table 3) illustrates the sub-cooling path:

Table 3 - Sub-cooling Log Mean Diagram

air ↓ 65°F

refrigerant / 35° → [ ] 40°F →

↓ 50°F

$$?\,t_{lm} = [(65\text{-}40) - (50\text{-}35)]/\log_m(25/15) - 19.6 \quad (3)$$

$$q = 72(0.211338)\,(19.6) = 160 \text{ Btu/hr per foot of tubing (153.8 W per meter of tubing)} \quad (4)$$

Then the length of tubing required to supply the energy to heat the refrigerant from 35°F to 40°F is found as follows:

$$\text{Length} - 4.04\,(60)/160 = 1.52 \text{ feet } (0.463 \text{ m}) \quad (5)$$

Additional Examples of Calculations:

FROM 0.85 TO 0.9

$$\text{Evaporative heat} = (2.25\text{-}2.125)\,(84) = 10.5 \text{ Btu/min} \quad (6)$$

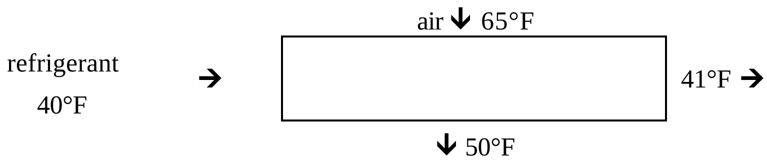
$$(744.35)\,(0.11338)\,(16.4) = 1384.1 \text{ Btu/hr per foot of tubing} = 23.07 \text{ Btu/min./ft.} \quad (7)$$

$$L = \frac{10.5}{23.07} = 0.46 \text{ feet} \quad \leftarrow \text{From 0.85 to 0.9} \tag{8}$$

For 1°F Superheat, t = 40+1 = 41°F

$$\text{Superheat} = c_p m(?t) = 0.2182\ (2.5)\ (41\text{-}40) = .55 \text{ BTU/min.} \tag{9}$$

Table 4 - Superheat Log mean Diagram



$$?\,t_{lm} = \frac{(65\text{-}41) - (50 - 40)}{\ln\left(\frac{24}{10}\right)} = \frac{14}{0.8755} = 16°F \tag{10}$$

$$q \text{ transferred} = h_g\, A(?t)_{lm} = (74.23)(0.11338)(16) = 134.6 \text{ Btu/hr.ft.} = 2.24 \text{ Btu/min per ft.} \tag{11}$$

$$L = \frac{.55}{2.24} = .25 \text{ feet} \quad \leftarrow \text{1°F Superheat} \tag{12}$$

Table 5 Theoretical Evaporator Model – Calculated Tube Segments

Refrigerant Quality Range (vapor percentage)	**Length of Segment (feet)**	**Total Evaporator length (feet)**	**Average BTU/min/ft**	**BTU/min /length**	**Flow* BTU/min /length**
Sub-cooled	1.52	1.52	– 4.04		Liquid
0.0 to 0.1	5.41	6.93	3.88	20.99	Strat-Wavy
0.1 to 0.3	5.05	11.98	8.31	41.97	Intermittent
0.3 to 0.8	5.77	17.75	18.19	104.96	Annular
0.8 to 0.85	0.44	18.19	23.61	10.39	Annular
0.85 to 0.9	0.46	18.65	23.07	10.61	Annular
0.9 to 0.98	1.24	19.89	13.51	16.75	Ann/Strat-Wavy w/ dry-out
10°F Superheat	2.58	22.47	2.13	5.52	Vapor
Evaporator BTU/min		22.47		211.19	

Table 5 shows the length of each segment required to change the quality of refrigerant for the indicated range under the theoretical conditions.

In the conventional way of thinking, it is desirable to have the refrigerant enter the evaporating coil as a sub-cooled liquid or as a saturated liquid with 0% vapor. In this extreme way of thinking, it is preferred to have the refrigerant enter the coil as a vapor-liquid mixture with around 30 weight percent vapor, and at a mass flow such that annular flow prevails.

Table 6 Theoretical ABF/HVFT Evaporator Model – Calculated Tube Segments

Refrigerant Quality Range (vapor percentage(	Length of Segment (feet)	Total Evaporator length (feet)	Flow* BTU/min/ft		
0.3 to 0.8	5.77	5.77	18.19	104.96	Annular
0.8 to 0.85	0.44	6.21	23.61	10.39	Annular
0.85 to 0.9	0.46	6.67	23.07	10.61	Annular
0.9 to 0.98	1.24	7.91	13.51	16.75	Ann/Strat-Wavy w/ dry-out
1°F Superheat	.275	8.19	2.24	.616	Vapor
Evaporator BTU/min		8.19		143.326	

Table 6 shows the length of each segment required to change the quality of refrigerant for the indicated range under the theoretical conditions.

Calculations show that for a conventional liquid entering with five degree Fahrenheit (5°F) sub-cooling, it requires a total length of 22.47 feet of ½ inch OD tubing to produce a vapor with five degree Fahrenheit (5°F) superheat. An ABF/HVFT vapor-liquid mixture of 30% vapor entering the coil requires 8.19 feet of ½ inch OD tubing to produce a vapor with one degree Fahrenheit (1°F) superheat. The same total mass flow of 2.5 lb/min of R-134a was used in each case.

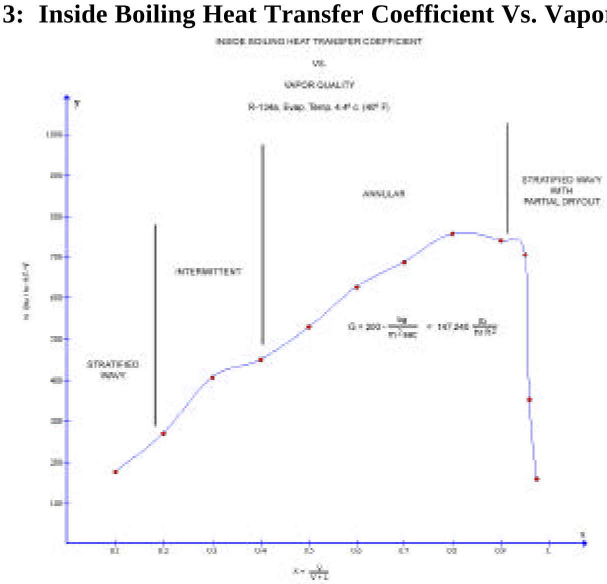
In the Baseline theoretical case, the total heat required to remove the sub-cooling, for evaporation, and for superheating is 12,671 Btu/hr. In the ABF/HVFT case the total heat for evaporation and for superheating is 8,600 Btu/hr. We can make the following calculations:

$$8{,}895 / 13{,}007 = 0.67871, \text{ or } 67.87\ \%, \text{ and} \tag{13}$$

$$8.19 / 22.47 = 0.3644851, \text{ or } 36.45\ \% \tag{14}$$

Under the conditions given above, an ABF/HVFT evaporator can transfer 67.87 % of the conventional DX heat load with only 36.45 % of the tubing length. Since the best performance of the coil is found in the segments of the evaporator between .3 and .98 (as indicated in Figure 3) with the mass-velocity indicative of annular flow, it is desired that the evaporator utilize only this range for optimization. As visualized in the ongoing glass evaporator test, the ABF/HVFT evaporator coil has achieved this.

Figure 3: Inside Boiling Heat Transfer Coefficient Vs. Vapor Quality



5. CONCLUSION

The refrigeration industry has focused upon high side savings for reduction in energy consumption, and has addressed air-side concerns in its approach to improve evaporator performance. The bi-phase region of a refrigeration system is not well understood, and advancement in our industry hinges on improvements in this area. It is commonly held that the differing heat transfer coefficients of the various possible flow regimes are so similar that the small change cannot improve evaporator efficiency.

The ABF/HVFT (Altered Bi-phase) and HVFT (High Vapor Faction and Turbulent) flow at the evaporator inlet and extending this flow throughout the evaporator to safely minimize superheat as demonstrated in the ABF/HVFT glass evaporator. Field applications have repeatedly out performed the Baseline DX evaporator as reported in previous ASHRAE and IIR work, which has demonstrated that the existing evaporator surface area can repeatedly be utilized more efficiently. While 4% to 6% steady state efficiency improvements have been experienced, transient operation demonstrates energy reduction or capacity increase. ABF/HVFT evaporator is a viable means of improving heat transfer and overall evaporator efficiency.

TEST GUIDELINES MET

All Thermocouples and Sensors are certified as matched, and have been certified together using standards having traceability to the NIST and were manufactured in accordance with the guidelines set forth by ISO 9001. All infrared readings were used as confirmation of thermocouple readings and are not reported

ACKNOWLEDGEMENTS

The authors thank Wilbert F. Stoecker, ASHRAE Fellow, Professor Emeritus University of Illinois at Urbana-Champaign for his ongoing contributions.

The authors would also like to thank Michael Micak of Carrier Commercial Refrigeration for his project observations and contributions.

REFERENCES

* *Report Dated November 1999 – Test Comparison Between Conventional Refrigeration and XDX®*, Underwriter's Laboratories, 1999
* ASHRAE Handbook, 1993, ASHRAE, Atlanta , GA.
* A Basis for Rational Design of Heat Transfer Apparatus, 1915, Wilson , E. E., Trans. ASME, Volume 37
* Industrial Refrigeration Handbook, 1988 - 1995, Wilbert Stoecker
* Report on the Recent Studies of Flow Boiling in Horizontal Tubes, The Journal of Heat Transfer, Volume 120pp 140-165 (1998)
* Flow of Fluids Through Valves, Fittings, and Pipes, Technical Paper No. 410, 1998 Crane Co
* ASHRAE Paper, July 2002, *Improved Performance Characteristics of a Fin-Tube Heat Exchanger Using High Vapor Fraction and Turbulent (HVFT) Entering Fluid*, Wightman, Wendrow, Sweetser, Worek