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Designing Systems to Use Simpler Expansion Devices

C. W. Bullard and I. E. Davidson

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For additional information:

Air Conditioning and Refrigeration Center
University of Illinois
Mechanical & Industrial Engineering Dept.
1206 West Green Street
Urbana, IL 61801

(217) 333-3115

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Designing Systems to Use Simpler Expansion Devices
C. W. Bullard, Principal Investigator

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Abstract

This report analyzes a broad spectrum of strategies for actively or passively controlling the inlet state of fixed-geometry expansion devices such as capillary and orifice tubes, to match compressor mass flow rates with minimal performance degradation in an efficient R410A a/c system. A TXV system was selected as the baseline for an exhaustive series of design options, including from heat exchanger sizing to use of receivers, internal heat exchangers, bladder accumulator and simple air flow modulation. Results yielded insights that can be generalized to other refrigerants and systems. Orifice tubes were found to produce higher efficiency than capillary tubes across the entire range of operating conditions, although the difference can be mitigated by proper choice of the latter's length and diameter. Only one configuration appears to be capable of matching TXV system performance across a wide range of operating conditions: a short tube orifice with low side receiver and internal heat exchanger.

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Chapter 1. Executive Summary

This chapter provides an overview of the project, the fundamental physics underlying the operation of fixed and variable expansion devices, and summarizes results of the analyses performed to compare them. Almost all the design options failed to match TXV system performance but for a few, the COP shortfall was less than 3% at outdoor ambient temperatures >27°C. However at lower ambient temperatures the only configuration capable of matching TXV system performance employed a short tube orifice with low side receiver and internal heat exchanger.

1.1 Project description

This project examines how refrigerant mass flow requirement varies with operating conditions, and proceeds to identify and analyze possible active or passive expansion devices capable of delivering that flow. The objective is to identify opportunities for developing expansion devices that are simpler and more reliable than the thermostatic and electronic valves currently used, while delivering higher performance than conventional orifice tubes and captubes. By focusing on the way mass flow requirements relate to other system states, and analyzing the way in which component designs and charge management can alter those states, results can be generalized to many kinds of a/c and refrigeration systems.

1.2 Mass flow requirement

This project is working backwards from the ideal flow requirement needed at off-design conditions, where the expansion device must match that provided by the compressor. The mass flow rate through captubes and short tubes depends mainly on the inlet state. Therefore so the goal is to design the rest of the system in a way that causes the necessary thermodynamic state to materialize at the expansion device inlet, across a wide range of system operating conditions.

1.3 Baseline system: single-speed compressor with TXV or EEV

An ideal compressor with a constant displacement rate, supplied with a constant suction vapor density (i.e. constant T_{evap} and ΔT_{sup}) would pump a constant amount of refrigerant, regardless of outdoor temperature. However the actual refrigerant flow rate is diminished at high discharge pressures by volumetric efficiency and (in piston compressors) the clearance volumetric efficiency. Therefore a compressor that pumps a given mass of refrigerant at the ARI-A design condition will pump about 3% (scroll) to 10% (recip) more when discharge pressure is low on a mild day. If everything else remained constant, and the expansion device delivered this increased flow to the evaporator, cooling capacity would increase 3-10%. However there is another factor at work: the evaporator inlet quality decreases from about 22% to 10% in an R410A a/c system as condensing temperature declines, thus delivering about 12% more liquid refrigerant to be evaporated. Therefore as the ambient temperature falls, the TXV is able to deliver plenty of refrigerant to match the gradual increase compressor mass flow rate.

If the TXV were to deliver this additional refrigerant to the evaporator, T_{evap} would need to decrease in order to increase LMTD enough to evaporate the extra liquid refrigerant. Since the TXV forces all the liquid to vaporize before leaving the evaporator, T_{evap} does indeed decrease and the compressor immediately sees a lower vapor density at its suction inlet. Thus the system is unable to take advantage of the increased compressor pumping capacity being made available by the milder outdoor ambient temperatures. With the evaporator and blower sized for the 35 ºC design condition and the TXV maintaining a constant superheat, the evaporating temperature must decrease in order to accommodate any increased cooling capacity. The resulting reduction in suction vapor density reduces mass flow rate through the compressor, which is then matched by the TXV and the system reaches a new equilibrium at the lower T_{evap}. The lower suction pressure causes the system to operate less efficiently due to the higher temperature lift, as shown in Figure 1.1.

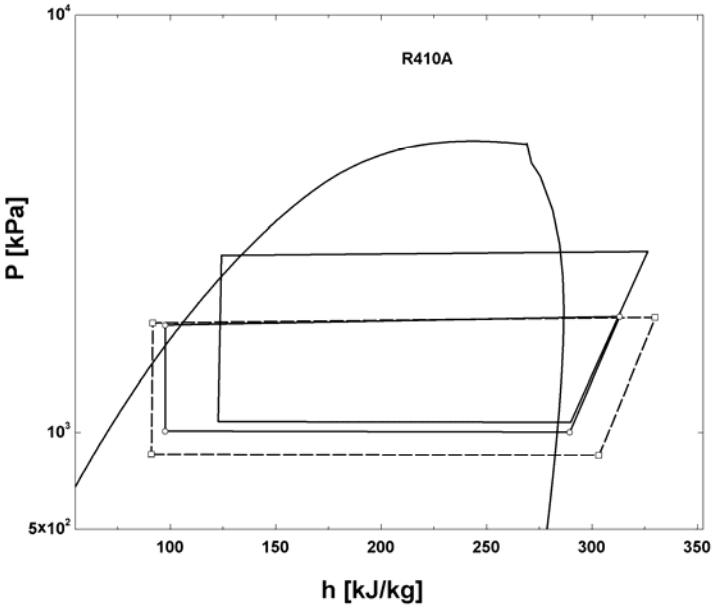

Figure 1.1. Effect of outdoor temperature: TXV and captube

The TXV is indeed an ideal expansion device because it keeps the evaporator surface fully utilized, and can accommodate whatever refrigerant flow rate is delivered by the compressor. It is the single-speed compressor and fixed evaporator effectiveness – combined with the shape of the vapor dome – that causes the real thermodynamic cycle to operate at an evaporating temperature lower than the ideal for a vapor compression system. Even with zero clearance volume and a perfect volumetric efficiency, the slope of the refrigerant's saturated liquid line would cause a single-speed system's efficiency to deteriorate significantly as outdoor temperature fell below the design point.

The following analysis compares captube and orifice tube systems with a baseline TXV system to quantify the *additional* performance degradation attributable to the fixed expansion device. The baseline system was chosen to be a very efficient one (COP=4.0; EER=14 at ARI-A), almost the maximum achievable in a critically-charged system with a single-speed compressor, PSC fan and blower motors and heat exchangers with 7mm tubes and louvered fins. Moreover the heat exchangers' internal volumes were sized to ensure that subcooling remained nearly constant to ensure a liquid inlet to the TXV over a wide range of operating conditions.

1.4 Choice of compressor (single-speed)

Both scroll and reciprocating compressors – sized for the ARI-A capacity rating condition – will produce higher mass flows as ambient temperature drops and causes discharge temperature to decrease. Both captubes and orifice tubes have the opposite trend. Equalization of flow rates depresses the suction pressure, decreasing system

COP as described above. Since the mismatch is smaller for the scroll than for the recip, most of the following discussion assumes that a scroll compressor was selected.

1.5 Comparing orifice tubes and captubes to TXV

A series of computational experiments was performed to compare three 3-ton split a/c systems. As shown in Table 1.1 below, the fixed expansion devices were sized to provide the capacity and efficiency as the TXV system at the ARI capacity rating condition (with 5°C superheat and subcooling).

Mass flow through both kinds of fixed expansion devices depends mainly on inlet pressure and subcooling. At mild ambient temperatures when inlet pressure drops, more subcooling is needed to match the compressor's mass flow rate. Figure 1.2 shows how, for a given inlet pressure, all captubes require more subcooling than an orifice tube to provide a given mass flow rate as inlet pressure falls. Since increased subcooling decreases COP by raising condensing temperature, the orifice tube is apparently preferred. On the other hand a set of parallel captubes may eliminate the need for a distributor, so both devices will be considered in the following discussion. The pressure differential across an orifice tube is smaller than that across a captube, making its mass flow rate much more responsive to modest changes in subcooling.

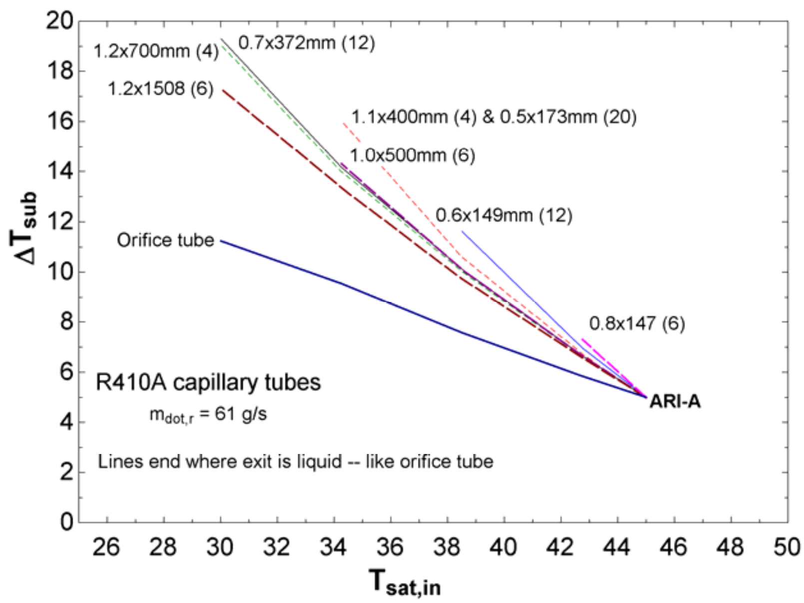

Figure 1.2. Subcooling needed to maintain mass flow rate

Note also in Figure 1.2 that the amount of subcooling required for a captube depends on its L and D, while the orifice tube's off-design performance is independent of L and D. Since for both devices, many combinations of {L,D} can provide the same mass flow at a given design condition, a smaller- diameter orifice tube can be selected to reduce its sensitivity to machining tolerances, while captube diameters can be based on the need to match a given number (say, 6) of parallel circuits in the baseline system's evaporator. The 6 captubes were selected to act as a distributor.

Table 1.1 compares off-design steady-state performance of systems equipped with the three kinds of expansion devices at 27°C ambient (near ARI-B) and also at a more extreme case of 19°C. Clearly the orifice tube outperforms the properly-sized captube, relative to the TXV baseline. Note that the performance degradation with both fixed expansion devices decreases rather slowly as ambient temperature falls from 35 to 27°C, then much more sharply at milder conditions.

Table 1.1. Summary of system performance at off-design conditions (critically charged)

	All	TXV		Orifice		Capillary	
T_{amb}[°C]	**35**	**27**	**19**	**27**	**19**	**27**	**19**
COP_{sys}	4.0	5.1	6.5	5.2	6.3	5.2	6.2
COP_{sens}	3.0	3.7	4.6	3.6	4.2	3.6	4.0
T_e [°C]	10.9	10.2	9.6	8.9	5.8	8.2	4.8
T_c [°C]	45.0	37.5	30.1	37.5	26.7	37.4	29.5
ΔT_{sub}[°C]	5.0	5.0	4.7	6.4	11.7	6.9	8.0
ΔT_{sup}[°C]	5.0	5.0	5.0	12.9	17.7	15.3	21.0

Note in Table 1.1 that system COP remains essentially unaffected by the presence of a fixed expansion device, until outdoor ambient temperature falls below 27°C. A closer look, however, reveals that sensible COP falls by a significant amount while the system's latent capacity is increasing because the evaporating temperature is falling. Given a constant thermostat setting, it is the sensible COP that determines the system's runtime and therefore its energy consumption.

1.6 Role of receivers

To protect the compressor during transients an accumulator or low-side receiver is needed. A receiver would maintain a saturated vapor state at the evaporator outlet at all steady state operating conditions, and would provide the extra charge needed to provide the subcooling needed to increase flow through the expansion device. Simulations showed that such a redesign could be accommodated by slight downsizing of the evaporator, with minimal effect on the COP's shown in Table 1.1.

1.7 Other design/control strategies considered

Several other approaches for avoiding off-design performance degradation, involving redesign or modulation of other components and/or adding new components, were analyzed as part of this project. All of the approaches were aimed at increasing subcooling at the inlet of the expansion device as outdoor ambient temperature decreased, in an attempt to match the compressor's demand for increased refrigerant flow. Simple passive control options and some active ones were investigated, in an attempt to diminish the performance degradation shown in Table 1.1. Eight unsuccessful approaches are listed in Table 1.2. Only one approach appears to be capable of matching TXV system performance: performance: use a liquid-suction heat exchanger (LSHX) in combination with a low-side receiver. It is discussed in the paragraphs that follow.

Table 1.2. Ways to modulate mass flow through fixed expansion devices

Strategy	Result
1. Reduce blower speed as a simple function of ambient temperature, sufficient to keep evaporator fully wetted, preventing excessive superheat.	Net loss of evaporator UA as 2-phase area loss averted, but U decreases at a faster rate; substantial COP penalty.
2. Reduce outdoor fan speed as a simple function of ambient temperature, to provide sufficient subcooling for expansion device to match compressor mass flow rate.	Enlargement of subcooled zone forces condensing temperature too high; shrinks condensing area at same time air-side resistance increases and heat sink is diminished by lower air flow rate.
3. Bladder accumulator adds charge to condenser as needed to increase subcooling.	High condenser subcooling increases condensing temperature.
4. Decrease evaporator internal volume (tube diameters) so it can fill faster as charge is shifted from condenser at low ambients; idea is to avoid excessive superheat.	Fixed expansion device cannot reallocate charge like a TXV. Evaporator charge is determined by superheat, which in turn is determined by suction pressure reduction needed to match compressor mass flow rate.
5. Increase condenser volume (tube diameter) so more charge goes to evaporator at low ambients when condenser's 2-phase density decreases.	All the charge thus released is needed for subcooling in the larger-diameter tubes; none left for the evaporator. Again, evaporator charge determined by suction density reduction needed to match compressor mass flow.
6. Add subcooler to obtain extra subcooling outside condenser; perhaps modulate with separate fan as function of T_{amb}.	Max subcooling still limited by difference between condensing temperature and ambient (generally <10°C). Captubes need more. An efficient condensing unit probably a better choice because it can further reduce air-refrigerant ΔT.
7. Obtain extra subcooling downstream of the condenser by using a liquid-suction heat exchanger as a heat sink that is colder than ambient air.	Cannot provide the 5-10°C *additional* subcooling needed at low ambients, because heat transfer is roughly proportional to (T_c-T_e). Therefore mass flow rate decreases instead of increasing as ambient temperature falls.
8. Design fixed orifice tube {L,D} for low-ambient condition, and charge for 5-6°C subcooling at that condition (same as TXV). Use discharge-to-liquid heat exchanger to decrease subcooling by adding the 600W needed to reach 1°C subcooling at 35°C ambient. This would keep system mass flow constant across the entire range, matching COP of TXV system.	UA and effectiveness of this internal heat exchanger would be constant, so it would heat the liquid line at a rate proportional to (T_{dis}-T_{amb}). This difference varies enough to provide 600 W more on hot day than mild day, but the ~1200 W provided on the mildest day would decrease COP more than modulating orifice tube mass flow would increase it.

A potentially promising approach. Recall that a low-side receiver ensures full utilization of the evaporator's heat transfer surface. At the same time it holds excess charge not needed at certain operating conditions, thus allowing the condenser charge inventory to reach a level that increases subcooling and/or subcooling to the extent necessary for the fixed expansion device to match the compressor's mass flow rate.

By using a liquid-suction heat exchanger (LSHX) in combination with a low-side receiver, it is possible to provide the extra ~10°C subcooling needed by the orifice tube at the mild ambient condition. Unlike strategy #7 in Table 1.2, the idea is to allow the hot side of the internal heat exchanger to fill ~95% of its length with 2-phase flow from the condenser at the design (35°C ambient) condition, so it would provide only ~5°C subcooling at the orifice tube inlet. This would be accomplished by sizing the orifice tube and charging the system to achieve ~3°C condenser outlet subcooling for that cool outdoor temperature. At that condition, the hot side of the internal heat exchanger would be filled with 100% liquid, subcooled to about 10°C at the inlet of the expansion device.

This is exactly the amount of subcooling required to maintain a constant mass flow rate across the range of operating conditions, as shown in Figure 1.2.

At the ARI-A condition the condenser outlet quality is about 14%, but the internal heat exchanger provides the additional heat transfer needed to provide the nominal amount of subcooling required at the expansion device inlet. Then as ambient temperature decreases and “the compressor calls for more mass flow”, the inlet subcooling at the orifice tube increases gradually to ~10°C. The receiver provides the extra charge needed to fill the entire length of the LSHX with subcooled liquid. System simulation analyses showed that ~10°C subcooling at the mild operating condition could be achieved with a relatively simple LSHX made of a “sandwich” of three microchannel tubes about 0.7 m long. Many other designs would also work; no effort was made to optimize that component design.

The analyses show that this simple, totally passive approach could match TXV system’s evaporator and condenser performance across the entire range of operating conditions, but it would require three components to replace the TXV: an orifice tube; low-side receiver; and liquid-suction heat exchanger.

Unfortunately it is not possible to show system COP results for this option alongside the results for the simpler ones shown in Table 1.1. The reason has to do with shortcomings in the compressor submodels currently available for use in system simulations (whether based on prescribing isentropic efficiencies, or based on compressor calorimeter “maps” which use data obtained at constant suction temperature). Neither type of model accounts for the effect of suction temperature variations on mass flow and power, especially for compressors with low-side sumps which use suction gas for motor cooling. The internal heating due to motor cooling will also vary with suction temperature. The unknown variations in refrigerant density *at the suction port* inside the compressor shell can affect both mass flow rate and isentropic efficiency. In systems without internal heat exchangers, suction temperatures remain relatively constant, so this has not been an issue for a/c systems in the past. While an accurate assessment of compressor power must await experimental results or more detailed compressor modeling, the results shown in Table 1.3 suggest that the overall system response is very favorable.

Note in Table 1.3 that the orifice tube sees a large increase in inlet subcooling as ambient temperature falls, while the LSHX allows the condenser to continue to operate efficiently with a large 2-phase zone. The receiver keeps the evaporator fully utilized, so it is not surprising that the system’s evaporating and condensing temperatures are very similar to those of the TXV system.

Table 1.3. A passive alternative to TXV or EEV

	TXV vs. Orifice tube + rec + LSHX					
	35		**27**		**19**	
Q_{sens}	10.5	10.5	11.3	11.5	12.1	12.4
Q_{lat}	2.6	2.6	3.1	3.3	3.6	3.8
T_e [°C]	10.9	10.8	10.2	10.0	9.6	9.3
T_c [°C]	45.4	43.7	37.5	36.8	30.1	30.0
ΔT_{sub}[°C]	5.0	2ph/4.0*	5.0	2ph/6.0*	4.7	3.4/11.0*
ΔT_{sup}[°C]	5.0	0	5.0	0	5	0
* indicates subcooling at condenser exit and LSHX exit						

These results may call for reconsideration of the role of LSHX in systems using all the new HFC's. Recall that the thermodynamic properties of R22 imposed a ~5% COP penalty, so internal heat exchange was never employed in those systems. All R134a refrigerators and some R404A display cases successfully employ LSHX in an attempt to capture the ~5-10% thermodynamic benefit. Rough approximations (e.g. adiabatic compression and constant isentropic efficiency) suggest that an LSHX should have no effect, positive or negative, on COP due to the thermodynamic properties of R410A, because the increased refrigerating effect is exactly offset by the penalty at the compressor. A more precise evaluation of this approach must await the available of compressor test data in which suction gas temperature is varied while suction and discharge temperatures are held constant.

1.8 Variable speed compressor systems

Simulations of systems equipped with a variable speed compressor have demonstrated conclusively that there is no simple way to modulate flow through fixed expansion devices over the range required (factor of 3 to 5 variation). Such systems are usually equipped with variable speed drives on the fan and blower as well. This enables them to independently control capacity (compressor speed) and humidity (blower speed), and to reduce outdoor fan speed at very low load conditions to save power when less condenser UA is needed. Stated another way, such systems can be designed for optimal performance at the load condition where most operating hours occur. Then the fan, motor and compressor speeds can move to high-speed operation on extremely hot days with some sacrifice of efficiency, dehumidification and quiet operation. Adding a TXV or EEV to such a system therefore enables a variety of benefits that fixed expansion devices cannot support, because the adjustable expansion device can track the compressor mass flow rate across the entire range of outdoor temperatures and sensible and latent loads.

1.9 Summary

Previous attempts to develop temperature- or pressure-sensitive orifice tubes have been described in the literature but not employed in practice for a variety of reasons. This project used a system simulation model to explore opportunities for modulating flow through capillary tubes or short orifice tubes, by controlling the refrigerant state at the expansion device inlet. Both passive and simple active approaches were examined. One passive approach appears capable of matching TXV system performance, but it involves adding a low-side receiver and liquid-to-suction heat exchanger to the system. Results also showed how proper choice of the compressor type and expansion device L and D can minimize performance degradation at off-design conditions.

Chapter 2. Baseline System: Single-Speed Compressor with TXV or EXV

Consider as a baseline system a 10.5 kW split a/c system, using R410A and designed to operate very efficiently at the ARI-A capacity rating condition, with a system COP = 4.0 (EER = 13.7). A highly efficient system was selected as the baseline because it will probably be more representative of future technology trends, and because an efficient system may react differently to off-design conditions than the less efficient systems of the past.

The efficiency of the baseline system is very near the maximum achievable in a critically-charged system with a single-speed compressor, permanent split capacitor (PSC) fan and blower motors, and heat exchangers with 7 mm tubes and louvered fins. The 3-row evaporator and 1-row condenser frontal areas (0.3 and 1.4 m^2, respectively) and their face velocities are not significantly different than today's commercially available units. Air-to-refrigerant approach temperature differences at the heat exchanger exits were <2°C. Moreover the heat exchangers' internal volumes were sized to ensure that subcooling remained nearly constant at ~5°C to ensure a liquid inlet to the TXV over a wide range of operating conditions.

A single speed compressor was selected for this analysis, because simulation modeling (described later in this report) showed convincingly that a fixed expansion device could not accommodate the large (3x to 5x) variation in refrigerant flow rates that a variable-speed compressor would require.

An *ideal* compressor with a constant displacement rate is a constant volumetric flow device. Supplied with a constant suction vapor density (i.e. constant T_{evap} and ΔT_{sup}) it would deliver a constant mass flow rate of refrigerant, regardless of outdoor temperature. However the actual refrigerant flow rate is diminished at high discharge pressures by reduced volumetric efficiency ($\eta_v < 1$) due to internal leakage, and (in piston and rotary compressors) by the additional effect of clearance volumetric efficiency. Therefore a compressor that pumps 100 g/s at the ARI-A design condition will pump about 3% (Copeland ZP32K scroll) to 10% (Bristol H81J273ABC recip) more refrigerant when discharge pressure is low on a mild day. The change in volumetric efficiency of rotary compressors lies somewhere between those of scroll and reciprocating compressors. If everything else remained constant and the expansion device delivered this increased flow to the evaporator, cooling capacity of a scroll compressor would increase around 3%.

However there is another factor at work: the evaporator inlet quality decreases from about 22% to 10% as condensing temperature declines, delivering about 12% more liquid refrigerant to be evaporated.[1] Since the compressor volumetric flow rate is constant and the TXV keeps the evaporator superheat essentially constant, the single-speed compressor cannot handle a 12% increase in vapor flow leaving the evaporator. This effect, determined by the slope of the saturated liquid line on a P-h chart, exacerbates the increase in compressor mass flow rate caused by the improvement in volumetric efficiency as condensing pressures decline as outdoor ambient temperature decreases.

Fundamentally, these two effects combine to increase compressor capacity at low ambient temperatures. Or instead of thinking about the compressor becoming oversized on mild days, it is equivalent to think of the rest of

[1] For R-410A, as condensing temperature decreases from 45 to 30ºC. For R22 the decrease is from 19 to 9%, which also adds about 12% to the liquid flow rate.

the system (evaporator and condenser and their air flow rates) becoming undersized. The result is that the LMTD's of both the evaporator and condenser need to increase, in order to handle the substantially higher heat transfer rates. Therefore the evaporating temperature, for example, would have to decrease about 1°C in order to increase the LMTD enough to vaporize the additional liquid refrigerant.

This is not the end of the story, however. The lower evaporating temperature reduces the vapor density at the compressor suction inlet, which reduces mass flow rate through the compressor, which is then matched by the TXV as the system reaches a new equilibrium at the lower T_{evap}. As a result, the compressor operates less efficiently due to the higher temperature lift caused by the lower suction pressure. The net increase in evaporator capacity also shifts the sensible heat ratio downwards, as the colder surfaces see increased latent load.

Note that these effects are triggered mostly by the properties of the refrigerant, specifically the slope of the vapor dome, and to a lesser extent by the properties of the compressor (scroll, rotary or reciprocating). It occurs with a TXV as described in this example, but the same penalty also appears in the case of capillary or orifice tubes.

Nevertheless, the TXV is indeed an "ideal" expansion device in the sense that it keeps the evaporator surface fully utilized by minimizing superheat at all operating conditions, while accommodating whatever refrigerant flow rate is pumped by the compressor. It is not the TXV that degrades system performance at off-design conditions; it is the fixed displacement rate of the compressor. The lack of a variable-speed compressor, combined with the fixed size of the evaporator and condenser, causes the real thermodynamic cycle to depart from the ideal for a vapor compression system.[2] Even an ideal single-speed compressor – one with zero clearance volume and a perfect volumetric efficiency – would force a single-speed system's efficiency to deteriorate significantly as outdoor temperatures fell below the design point, because such a compressor would deliver a constant volumetric flow rate (hence an increasing mass flow rate). This is inconsistent with the mass flow requirement of the ideal vapor compression cycle, which changes continuously as evaporator inlet quality responds to changing outdoor temperatures and the building's thermal loads, allowing the high-side pressure to move downwards to the wider part of the vapor dome.

Since the TXV neither exacerbates nor negates the inefficiencies introduced by a single-speed compressor, the following analysis compares captube and orifice tube systems with a baseline TXV system. The purpose is to quantify the *additional* performance degradation attributable to the use of a fixed expansion device in systems equipped with single-speed compressors.

[2] As compressor capacity grows, it becomes oversized relative to evaporator and condenser, thus requiring a higher temperature lift to increase LMTD's enough to accommodate higher heat transfer.

Chapter 3. Comparison to Systems with Fixed Expansion Devices

3.1 Comparing captubes to TXV (single-speed)

A series of computational experiments was performed to compare three 3-ton split a/c systems, designed for identical capacity and efficiency at the ARI capacity rating condition (95/80/67F) with 5°C superheat and subcooling.

Figure 3.1 illustrates the responses of the TXV and captube systems as outdoor air temperature drops below the 35°C design point. Both systems were equipped with a Copeland ZP32K scroll compressor sized for 10.5 kW load and 0.75 sensible heat ratio at the design condition. With 6 capillary tubes (1 x 600 mm) feeding 6 evaporator circuits (i.e. also serving as a distributor) the captube system's (dashed line) evaporating temperature dropped more than 6°C, compared to only about 1°C in the TXV system (solid line) as outdoor air temperature decreased from 35 to 19°C. Since the captubes could not open like the TXV to supply enough flow to keep the superheated region of the evaporator reached 43% of the total at the mild condition compared to 6% in the TXV system. With only a fraction of the evaporator wetted, the captube system was forced to operate at a lower evaporating temperature to establish a larger LMTD. The resulting drop in compressor suction density caused the system's compressor and captube mass flow rates to equalize after a net reduction of about 16%. In contrast, the reduction is only about 1% in the case of TXV systems because the valve can open to permit the evaporator to operate at a higher pressure, producing a higher-density suction gas to keep the mass flow rate and cooling capacity high.

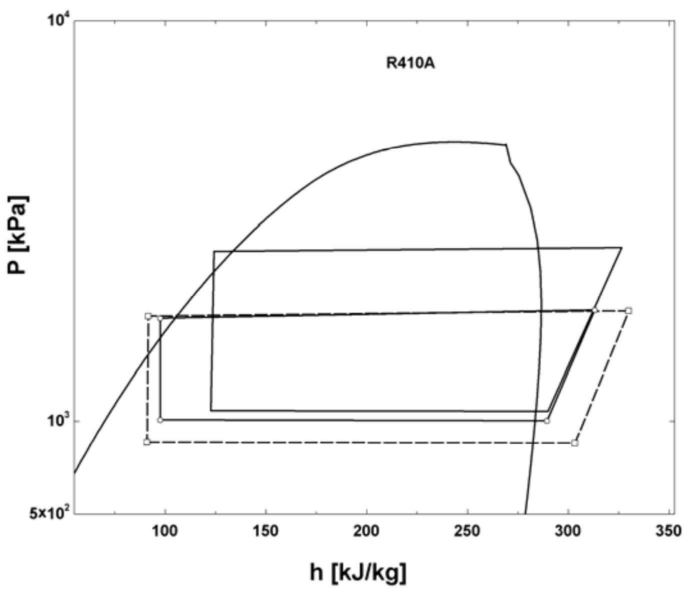

Figure 3.1. Effect of outdoor ambient temperature: TXV and captube

The captube system's COP remained within 1% of the baseline TXV system at outdoor temperatures above 25°C, but the gap widened quickly to 5% at 19°C. The captube's inability to provide as much mass flow as the TXV was only partially offset by the increased refrigerating effect at the 19°C operating condition, so the captube system was forced to operate at a lower evaporating temperature to establish a larger LMTD.[3] The net effect of the captube-compressor mass flow mismatch was a 7% loss of total capacity and colder evaporating temperature that decreased sensible heat ratio from 0.75 to 0.66 (vs. 0.75 to 0.71 in the TXV system).[4]

These steady-state simulation results suggest that captubes would result in only a minor loss of overall performance, broadly defined. The relatively small 7% loss of energy efficiency at 19°C is somewhat misleading because COP is based on total capacity, not sensible capacity. The captube system's sensible COP is 12% lower at that condition, requiring a correspondingly greater energy consumption over the extended runtime. The negative effect of longer runtime would be only partially offset by avoidance of extremely short cycles. Short cycles increase energy usage because charge is not optimally distributed among components after off-cycle migration of refrigerant through the captube. On the other hand the lower evaporating temperatures may in fact be desirable from a comfort standpoint, because the dehumidifying effect would be stronger. Latent (moisture) loads due to infiltration persist undiminished at nighttime while sensible (conduction and radiation) loads decrease significantly.

Due to the captube system's lower refrigerant mass flow rate, its condenser heat transfer rate (hence LMTD) is slightly lower than the baseline system's. The condensing temperatures remain within 1°C for the two systems at the mildest outdoor temperature, despite the fact that the subcooled area more than doubled (to 24%, from 10% at the design condition) in the captube system in order to provide the correct flow through the fixed expansion device. The similar condensing temperatures result from the decreased mass flow rate, which results from the increased enthalpy change seen in Figure 3.1 for the captube system. Note the large increase in evaporator superheat on a mild day occupies 43% of the area with the scroll compressor and 45% when the recip was used.

Results are summarized in Table 1.1.

3.2 Comparing orifice tube to TXV

Replacing the 6 captubes by a single 1.8 x 15 mm orifice tube and 6-circuit distributor, and repeating the simulation analyses described above, the results appear much more favorable. Since the scroll compressor system's off-design performance is again better than the recip, the following results compare to the scroll system. Compared to the baseline TXV case, the sensible COP was 7% lower because the suction pressure did have to decrease for the compressor to match the orifice tube's mass flow rate at 19°C ambient. The net effect would be a 8% longer runtime and a lower sensible heat ratio at the mild operating condition.[5] Thus the comfort tradeoff: if the thermostat is set to maintain a constant dry bulb temperature, an 8% energy penalty is incurred and a dehumidification benefit is obtained. To control humidity independently would require adjusting the blower speed.

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[3] Besides the shape of the dome, the increased refrigerating effect is attributable to slightly increased subcooling as refrigerant migrated from evaporator to condenser in response to increased superheat at the evaporator exit. The refrigerant's latent heat of vaporization is also larger at the depressed evaporating temperature, so a given change in quality produces a greater cooling effect.

[4] These effects are magnified in the case of the reciprocating compressor due to its greater mass flow sensitivity to discharge pressure. COP is same as TXV down to 27°C, then degradation reaches 9% at 19°C. Evaporation temperature drops 7°C between 35 and 19°C with captube, compared to 1.8 for the TXV system.

[5] For the case of a reciprocating compressor the effect is greater.

In Figures 3.2 and 3.3 the bottom sets of dashed lines show the set of feasible captube and orifice tube flow rates at five subcooling conditions, while the upper sets of lines show compressor mass flow rate at the design condition where T_{evap}=10°C. Because the compressor pumps more refrigerant than the fixed expansion device can handle (due to increased volumetric efficiency at this low-lift condition), charge accumulates in the condenser and is depleted from the evaporator. The resulting increase in condenser subcooling allows more flow through the expansion device. The locus of large solid dots shows where the compressor and expansion device lines eventually intersect to equalize mass flow rates after the compressor line shifts downwards to reduce suction density by lowering the evaporating temperature.

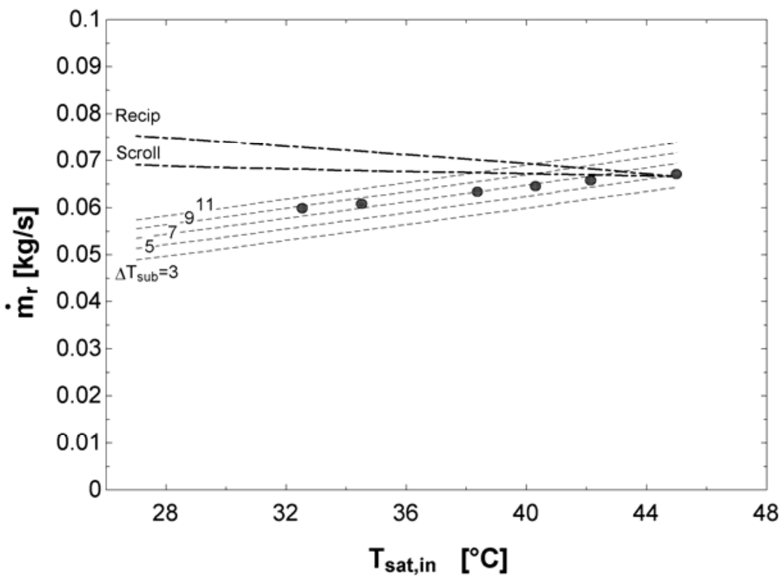

Figure 3.2. Captube and compressor flow rates

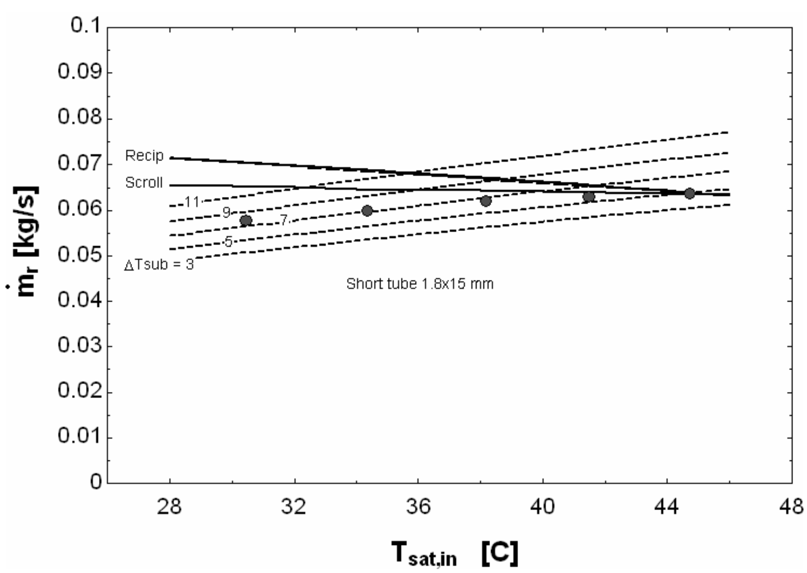

Figure 3.3. Orifice tube and compressor flow rates

Note that the orifice tube's mass flow rate is more sensitive to subcooling than that of the capillary tube. Moreover the lines of constant mass flow rate for the orifice tube in Figure 3.3 are slightly flatter than those for the capillary tube in Figure 3.2. As a result the gap between compressor and orifice tube mass flow rates was relatively small, so a smaller drop in suction density was required to equalize the flows.

It is apparent that the depression of evaporating temperature follows directly from the size of the gap between the compressor and expansion device flow rates. In both cases the evaporating temperature drops lower than the TXV case because the fixed expansion devices cannot open to provide enough flow to fully utilize the evaporator surface area.

Results are summarized in Table 1.1.

3.3 Optimizing geometry of capillary and orifice tubes

The whole-system responses shown in Figures 3.2 and 3.3 reflect the results of choosing the captube and orifice tube geometries (length, diameter) that yielded the most favorable off-design (mild weather or low-lift) performance characteristics. The choices and tradeoffs are described below.

Capillary tubes. The exit of an adiabatic captube, shown in Figure 3.4, is normally choked at a pressure somewhere on the isenthalp below the dome but above the evaporating pressure.[6] As a result, the captube mass flow rate is completely unaffected by the evaporating pressure. Figure 3.5 shows two independent sets of data. The typical capillary tube performance is shown by lower set of dashed lines; it depends only on the inlet state. The compressor mass flow rate depends on both evaporating and condensing pressures; the downward sloping lines show performance for the typical operating condition T_{evap}~10ºC. While the mass flow rates are equal at the design condition (~45ºC condensing temperature), they diverge at all mild-temperature operating conditions. For the mass flows to equalize at, say, a condensing temperature of ~32ºC corresponding to an outdoor ambient temperature of 19ºC, the saturated compressor suction temperature would need to drop enough for the compressor mass flow rate to fall into the 50-60 g/s range that the capillary tube is capable of handling. Figure 3.5 illustrates the effect on compressor mass flow rate as the evaporating temperature drops about 5ºC, and explains the approximate location of the resulting equilibrium point shown in Figure 3.2.

[6] The speed of sound in 2-phase flow is very low, about 20-60 m/s, depending mainly on quality.

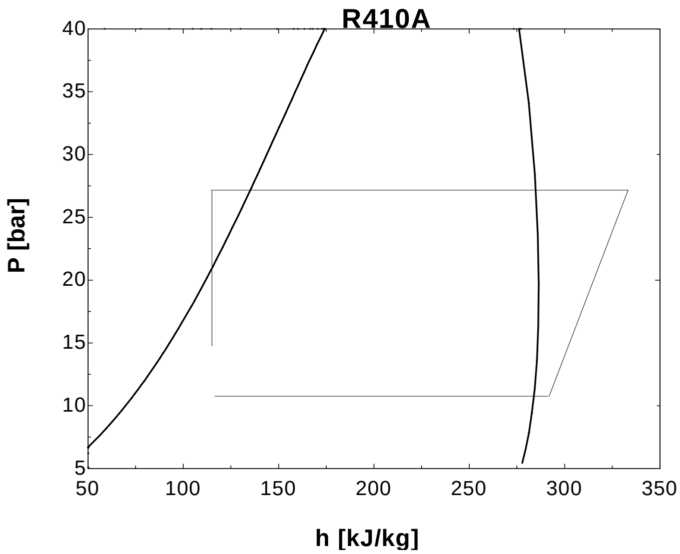

Figure 3.4. Choked exit of capillary tube

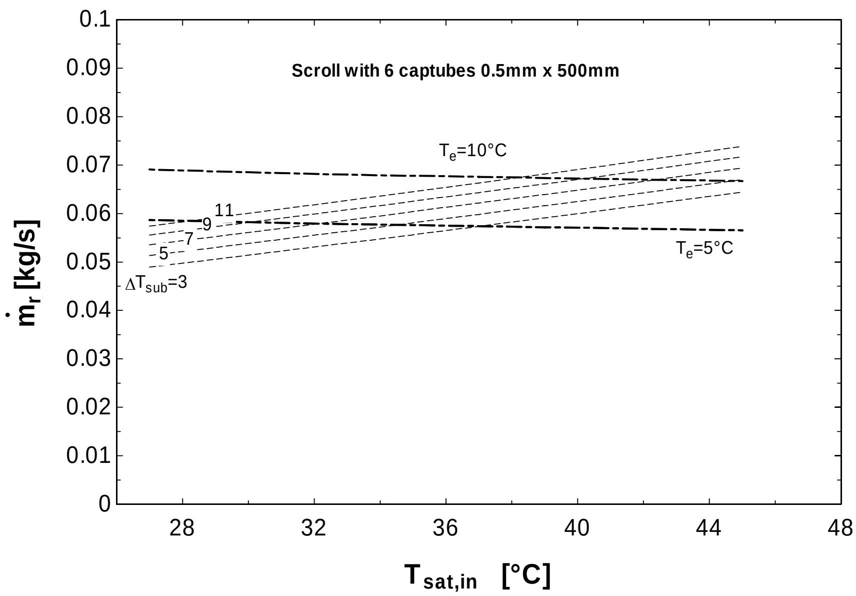

Figure 3.5. Effect of lower suction pressure

As the compressor transfers more charge to the condenser than the expansion device can take out, charge accumulates in the condenser. The increased subcooling increases the outflow through the expansion device, and at the same time decreases compressor flow rate by increasing the discharge pressure until a new equilibrium is reached. Note also that there is a practical upper limit on the amount of subcooling achievable in an efficient system. To achieve high efficiency at the 35ºC ambient temperature design condition, the condensing unit is usually sized to operate at $T_{sat,in}$ ~ 45ºC. The maximum possible subcooling is the difference between these temperatures: 10ºC in this case. At milder ambient temperatures the same condensing refrigerant-to-air temperature difference is required at the condenser because the heat rejection requirement is virtually unchanged.

Since the ability of the condenser to increase subcooling is therefore quite limited, the system must accommodate the mismatch between compressor and refrigerant mass flow rates primarily by reducing the compressor suction inlet density – i.e. by forcing the compressor to operate along lines of lower evaporating temperature.

Reduction of the evaporating temperature results in efficiency loss. So does the increase in condensing temperature caused by the higher subcooling needed for the fixed expansion device to match the compressor mass flow rate. Therefore, it is desirable to select a capillary tube {L,D}combination that requires the least subcooling to maintain a given mass flow rate as the pressure at the inlet of the expansion device decreases in response to mild ambient temperatures. Figure 3.6 shows the impact of various L, D choices. The smaller diameter (and shorter) tubes tend to require more subcooling as inlet pressure drops, and lose their choked exit as the length of the subcooled region grows rather quickly to occupy the entire length of the tube. The longer, larger diameter tubes can operate over a broader range of inlet conditions, and require somewhat less subcooling to offset the loss of head pressure. Therefore the capillary tubes selected for a system requiring 61 g/s at the design condition would be taken from the lower envelope of the curves shown here: 6 tubes @ 1 mm x 500 mm each, to provide adequate length for connections that will distribute flow evenly to six parallel evaporator circuits.

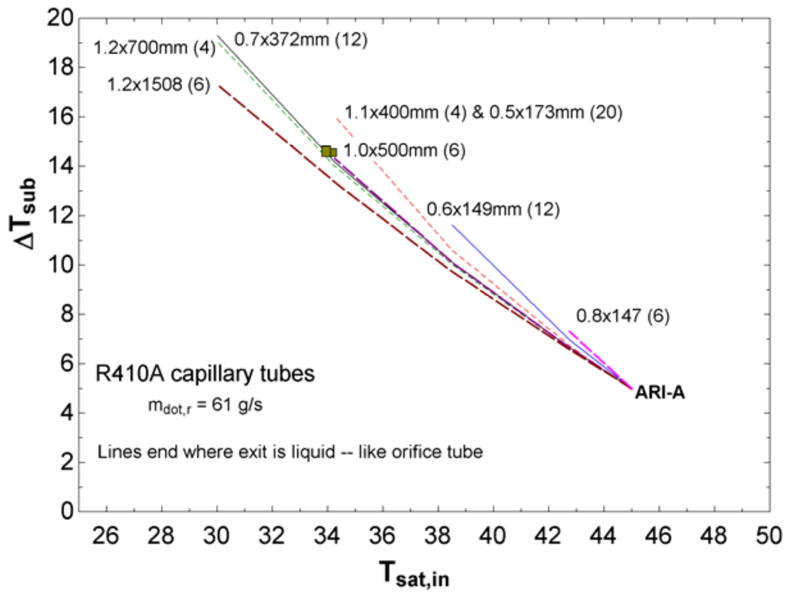

Figure 3.6. Selecting capillary tube L and D

Orifice tubes. The behavior of orifice tubes is fundamentally different, because they are too short to generate enough friction to flash the refrigerant inside the tube – except, of course if subcooling is very small (<2 ºC) in which case only a small amount of friction is required. The predominant feature of orifice tube flow is the presence of thermodynamically nonequilibrium conditions that must be well understood in order to predict the mass flow rate through the device. Similar nonequilibrium conditions are present in TXV's, but their effects are managed automatically by the valve's ability to adjust throat area to control the mass flow rate.

Figure 3.7(c) shows a white region indicating bubbles forming in the orifice tube as the pressure drops slightly in saturated liquid as it accelerates isentropically through the vena contracta – the areas of separated flow that form immediately downstream of the sharp-edged entrance. This white region indicates gas bubbles have formed. The pressure drop in the vena contracta is also triggers bubble formation in subcooled liquid when ΔT_{sub} = 2.8ºC (Figure 3.7(b)), but they apparently persist for only a few milliseconds in the nonequilibrium subcooled flow through the tube. In Figure 3.7(a) no bubbles are formed, due to the highly subcooled condition. The two-phase inlet condition shown in Figure 3.7(d) can indeed have a choked exit because frictional pressure drop is sufficient to establish a nearly-homogeneous equilibrium multiphase flow in which the speed of sound is very low.

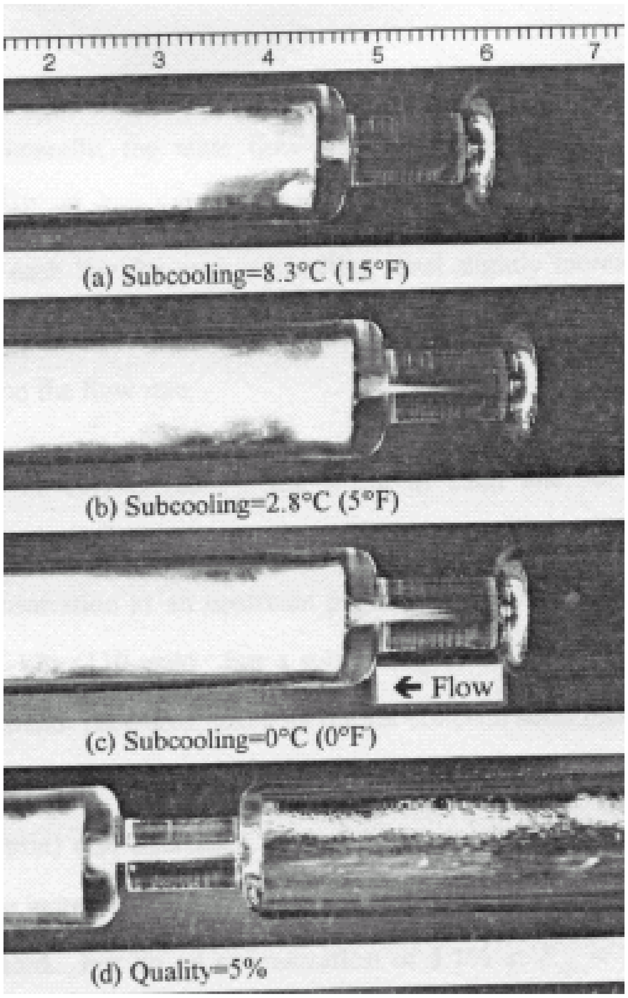

Figure 3.7. Flow through transparent orifice tube

The subcooled liquid exits the orifice tube at a subsonic velocity, because the speed of sound in liquid is very high. However the pressure downstream of the exit is far below saturation, causing the flow to flash immediately upon exiting the tube. At this point the nonequilibrium or metastable flow structure resembles that encountered by other underexpanded jets, such as the expansion of octane shown in Figure 3.8. Note that the liquid jet persists in a metastable state, with the rate of flash evaporation limited by its surface area. The schematic shows an idealized approach to modeling fuel injectors; the 2-phase mixture of flashing liquid and vapor travels roughly normal to the jet surface, accelerating to M>1 (the speed of sound in 2-phase flow is quite low, around 20-60 m/s depending mainly on quality). The pressure quickly falls far below the downstream evaporating pressure as the flow accelerates due to expansion. Then the pressure jumps discontinuously across the shock wave to match the downstream pressure. Thus the pressure at the exit plane of the jet remains quite close to the saturation pressure, while the shock wave isolates it from “seeing” the lower downstream pressure in the evaporator (Simões-Moreira, J. R. and C. W. Bullard, 2003)

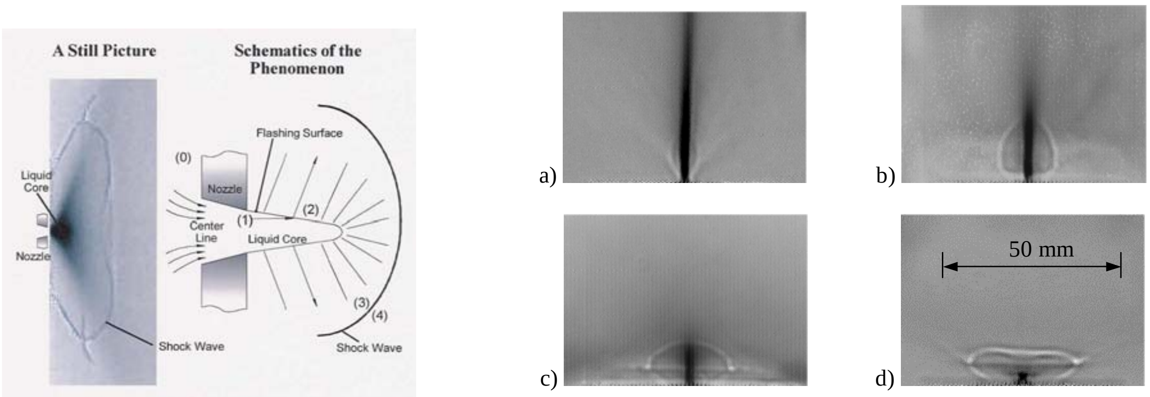

Figure 3.8. Underexpanded jet of octane isolated from downstream pressure by shock wave (Simões-Moreira, 2000)

The complexity of these flow regimes near the exit of orifice tubes, and the absence of thermodynamic equilibrium conditions, makes it extremely difficult to develop an accurate physical model of the process as was the case for capillary tubes. Recall that flow inside a capillary tube transitions from one equilibrium (liquid) state to 2-phase Fanno flow which accelerates gradually to M=1 at which the flow chokes. The choked condition at the cap tube exit defines the outlet state and allows the equations to be solved by marching upstream inside the tube, leaving the complexities of subsequent supersonic expansion and shock waves irrelevant. Unfortunately in the case of orifice tubes, the shock structure apparently interacts with the tube walls and tube exit geometry in ways that affect the exit plane pressure – causing it to diverge slightly from the saturation pressure. Therefore mass flow rate predictions must be based on empirical models developed from extensive testing of R22 (Kim & O’Neal, 1994), R410A (Payne & O’Neal, 1999; Kim et al. 2005) and R134a (Singh et al. 2000) for a wide range of orifice tube length and diameter combinations. However, even the results of these exhaustive tests must be used with caution because there have been no systematic attempts to quantify their sensitivity to geometric details of the exit tubing.

From comparing Figures 2.2 and 2.3 it is apparent that the orifice tube’s mass flow rate is more sensitive than the capillary tube to inlet subcooling. The reason is straightforward: the subsonic orifice tube flow “sees” an outlet pressure very close to the vapor dome. This driving pressure difference decreases with increased subcooling due to the slope of the vapor dome (See Figure 3.1). Therefore in orifice tubes a small increase in subcooling can increase the driving pressure differential by a relatively large percentage (Compare Figures 2.2 & 2.3). In capillary tubes the driving pressure differential is larger, reaching to the pressure at the choked exit below the dome. Counterintuitively, subcooling decreases the driving pressure differential in a captube because it increases the length of the liquid segment and shortens the 2-phase region where most of the pressure drop occurs. With less pressure drop, the choked exit of the capillary tube occurs at a pressure higher than in the low-subcooling case. However the mass flow rate actually increases despite the loss of driving pressure difference because the critical (sonic) mass flux is greater because of the higher density at the higher exit pressure. Thus subcooling acts in fundamentally different ways to increase the flow through capillary and orifice tubes. This explains the difference between the lines of constant mass flow rate shown in Figures 2.2 and 2.3.

The curve in Figure 3.9 shows the locus of {L,D} pairs for orifice tubes that can deliver a given mass flow rate at the design condition. The range of the empirical data set on which it is based is shown by the rectangle. The curve shows how the empirical model captured the effect of the orifice tube's frictional and entry/exit pressure drops. Figure 3.9 shows a rapidly increasing slope for tube diameters over 1.8 mm. As diameter increases, much greater tube length would be required to generate enough friction to hold mass flow rate constant because cross sectional area is increasing much faster than the area of the tube walls where frictional pressure drop is generated. Therefore as diameter increases, tolerances on tube diameter become more critical.

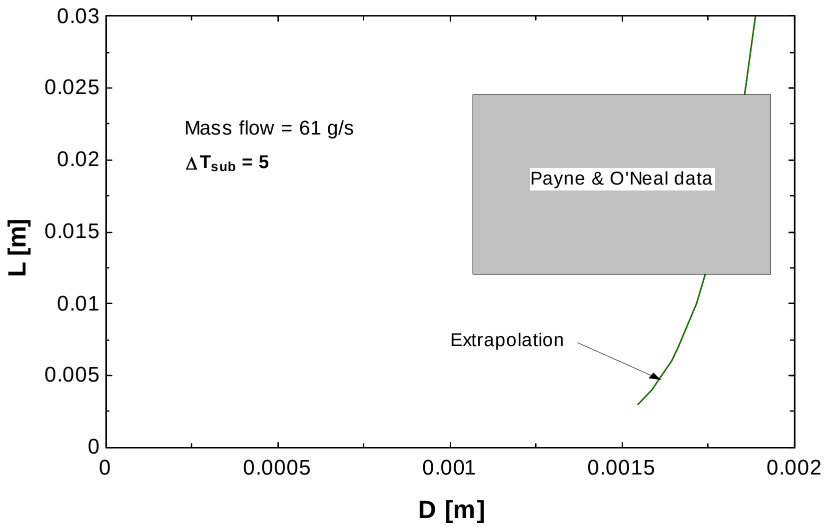

Figure 3.9. Feasible short tube L,D for given R410A flow

Figure 3.10 illustrates how the orifice tube's response to off-design inlet pressure is independent of its L and D. Physically, this results from the flash evaporation of the liquid exiting the tube, after experiencing a relatively small frictional pressure drop inside the tube. The pressure field created by the flash evaporation at the exit determines the driving pressure differential seen by the subsonic liquid flow through the tube. Orifice tubes require less subcooling than captubes to maintain a given mass flow rate as inlet pressure drops. Less subcooling translates into a smaller refrigerant-to-air temperature difference and more heat transfer from the condenser. In this respect they outperform even the best {L,D} combination for capillary tubes.

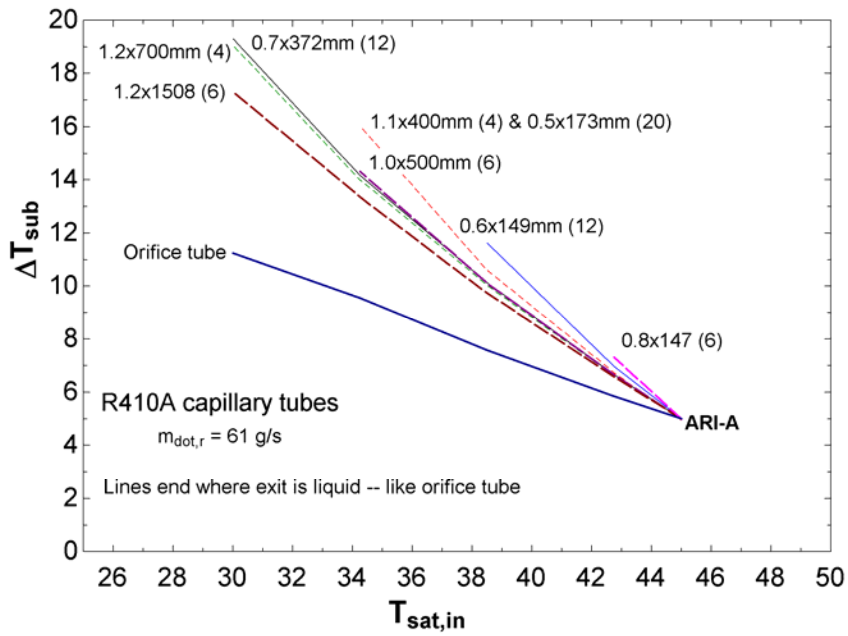

Figure 3.10. Orifice tube off-design performance

Mathematically, this result follows directly from the empirical equation developed by Kim et al. (2005) for R410A, which can be written in the following form:

$$\dot{m} = \mathbf{f}(L, D) \cdot \mathbf{g}(\Delta T_{sub}, P_{up} - P_{sat}, \mu_f, \mu_g, \rho_f, \rho_g, T_c, P_c)$$

where density and viscosity are evaluated at P_{sat}, the saturation pressure where the isenthalp crosses the vapor dome. . The separable nature of this equation indicates that any two different orifice tubes having equal values of **f** will respond in an identical manner to changes in inlet conditions, because the function **g** is independent of L and D. Internally the function f shows that mass flow rate depends mainly on D as illustrated in Figure 3.9, with the frictional pressure drop due to length L having only a minor effect. In fact due to limitations on the range of data underlying this empirical equation, it is valid for only a factor of two variation in L, where doubling the length reduced mass flow rate by only 12%.

$$f(L, D) = D^{2.1904} L^{-0.1904}$$

These curve fits of empirical data (max deviation only ±5%) are also consistent with the physical phenomena described earlier. Consider two orifice tubes having different {L,D} but the same mass flow rate at a given design condition. The exit plane pressure seen by the subsonic liquid flow is determined by the structure of the shock wave structure generated by the flash evaporation of the exiting liquid jet. Since the flash evaporation occurs at a saturation pressure defined by the inlet enthalpy $P_{sat}(h_{in})$ which is the same for each tube, it is reasonable to expect that the exit plane pressures will be roughly the same. The shock structure can and does influence the exit plane pressure, causing it to deviate from P_{sat}, otherwise the standard orifice equation would apply and no empirical correction factors would be needed. It is logical that those correction factors would depend on the viscosities and densities of the two-phase mixture that forms the shock structure, as suggested by the empirical function **g** – only the physical details are unknown.

Before proceeding to use this and other empirical models of orifice tube flow, a caveat is necessary. All the experiments cited were obtained in a single facility, so the downstream tube diameter remained unchanged as L, D and the operating conditions were varied. To the extent that the downstream tube diameter (12.7 mm in Kim, Payne and O'Neal's experiments) may have influenced the downstream shock structure and hence the exit plane pressure, such variations would not have been detected in their experiments, nor reflected in their empirical curve fits.

3.4 Using different expansion devices in parallel

Since mass flow through capillary and orifice tubes responds to inlet conditions in fundamentally different ways, the question arises: Could benefits be obtained by using different devices in parallel to produce the desired mass flow response? The short answer is no. The details are discussed below.

Basically there are three types of fixed expansion devices that may – by themselves – fail to deliver the mass flow rate required by the system, but might have potential if they were operating in parallel:

- *Short tube orifice.* The driving pressure differential is determined by the pressure at the exit plane, immediately downstream of the subsonic liquid exit. The flash evaporation and expansion that occurs at that point creates a shock structure as described above. Empirical data (e.g. Kim and ONeal, 1994) has shown that this results in an exit pressure that is somewhat lower than the saturation pressure at the point where the isenthalp crosses the saturation dome. As subcooling increases, the subsonic liquid flow through the tube sees a larger driving pressure differential, and mass flow rate increases accordingly.
- *Capillary tube.* The captube's exit pressure is determined by a choked flow condition; occurring at point along the isenthalp farther below the dome where the 2-phase flow has accelerated to M=1. Captubes with relatively long L are capable of producing a larger frictional pressure drop, and therefore penetrate farther below the dome where their exit plane quality is relatively high. To generate this large frictional ΔP they require a large D to keep the low-quality flow subsonic as the high-quality exit is approached. Conversely, in order to generate the same design mass flow rate with a shorter L, a smaller D is needed to reach sonic velocity with less pressure drop. It is the higher pressure and density at the sonic exit that enables the higher mass flux. Because of the dominance of frictional pressure drop, captubes react differently as inlet subcooling increases, shortening the 2-phase region of the captube. Despite the fact that subcooling reduces the driving pressure differential, the higher pressure and density at the choked exit causes mass flow to increase.
- *Subsonic tube.* By selecting a very large L and D it is possible to ensure that a sonic exit condition is never reached. In this case the mass flow rate through the expansion device is determined by the requirement that the frictional pressure loss equal the difference between the system's high- and low-side pressures. Thus the mass flow rate, which must match that of the compressor, depends on the evaporating pressure. In this respect the long subsonic tube is similar to the orifice tube, which also has a subsonic exit and a mass flow rate determined by the pressure differential. Since the long tube's driving pressure difference (reaching all the way to the evaporator pressure) is much greater than that of the orifice tube (reaching only to the dome), a small change in inlet pressure has a correspondingly smaller effect on mass flow rate. Moreover, a change in subcooling at constant inlet pressure will cause the subsonic tube to react in a manner quite similar to a captube. The shortening of the 2-phase region of the tube where most of the

friction is generated will result in a higher exit density (lower quality as isenthalp moves leftward on the P-h diagram.

First, it is clear from Figure 3.10 that orifice tubes provide a better match (than a captube) to either a scroll or reciprocating compressor. However Figures 3.2 and 3.3 demonstrate that neither orifice tubes nor capillary tubes – when sized for the design condition – can match the compressor’s mass flow rate at mild conditions. The resulting adjustment process depresses the evaporating temperature, imposing a COP penalty. Since the orifice tube delivers the better performance of the two (i.e. less performance degradation), there is no benefit obtainable by adding a captube in parallel.

The third possibility investigated, a tube longer and larger diameter than a captube sized to produce the desired pressure drop at the design condition, was also found to be deficient. The results of system simulations with this kind of device showed the system performance to be slightly worse than that of the captube system as ambient temperature decreased. Details are discussed in Chapter 4.

Since all three devices fail to deliver enough mass flow to match the compressor at all off-design conditions, there is nothing to be gained by using two or more of the devices in parallel. The only advantage of capillary tubes or subsonic tubes is their ability to serve a dual purpose: expansion device and distributor. Since all three devices tend to degrade system performance (relative to a TXV), the strategy for maximizing system COP is to pick the one that minimizes performance degradation, namely the orifice tube.

Chapter 4. Modulating Inlet State of Fixed Expansion Devices

Both captubes and orifice tubes diminish system performance relative to that of systems equipped with TXV or EEV, as shown in Table 1.1. The root cause of the degradation is the failure of the fixed expansion device to match the mass flow rate of the compressor across a wide range of operating conditions. Since the flow through fixed expansion devices depends mainly on the thermodynamic state of the refrigerant at their inlet, this chapter describes both active and passive approaches to avoiding off-design performance degradation in systems with fixed expansion devices.

Active approaches include modulating system charge, or fan or blower speed in response to a simple control signal such as outdoor ambient temperature. Passive approaches include proper sizing of heat exchanger internal volumes, and adding components such as receivers or internal heat exchangers. The goal of all of the approaches is to increase subcooling at the inlet of the expansion device as outdoor ambient temperature decreases, in an attempt to match the compressor's demand for increased refrigerant flow. Eight unsuccessful approaches were listed in Table 1.2, and are described in detail below along with other issues in Sections 4.1 - 4.8. Only one approach appears to be capable of matching TXV system performance: performance: use a liquid-suction heat exchanger (LSHX) in combination with a low-side receiver. It is described in Section 4.9.

4.1 Relative size of heat exchangers

To avoid premature flashing and choking inside a TXV it is important to keep its inlet subcooled over a wide range of operating conditions. This is accomplished in a critically-charged system by properly sizing the evaporator and condenser internal volumes. At milder outdoor conditions the valve opens to let excess charge out of the condenser, providing exactly the amount of charge the evaporator needs to maintain the superheat setpoint at its new operating condition. Thus in the process of equalizing mass flow rates through the expansion device and the compressor, the valve effects a redistribution of charge: the condenser needs less because of its lower vapor density at lower pressures, while the evaporator needs more because of its lower inlet quality. These same component charge inventory requirements exist in a captube or orifice tube system. Could the relative sizes of heat exchangers be adjusted to prevent excessive superheating (charge depletion) in the evaporator?

As ambient temperature decreases, the baseline TXV system's condenser gives up about 74 g of charge to other components: primarily to the evaporator which needs it to accommodate the lower inlet quality, and to the liquid line that needs to accommodate the greater liquid density on mild days. The charge is available because much of the condenser's vapor turned to liquid as the vapor density increased while condensing pressure fell. Some of this liquid was needed to accommodate the enlargement of the condenser's 2-phase zone as the falling discharge temperature reduced the size of the desuperheating zone. The remainder was reallocated to the evaporator, and to the liquid line to offset the increase in density as the liquid cools.

Simulation analyses suggest that altering the relative sizes of the heat exchangers will not have much effect on system efficiency. Even doubling the internal volume of the condenser, while halving that of the evaporator by reducing the tube diameter, had very little effect on systems equipped with fixed expansion devices. Close inspection of the simulation results revealed that the underlying problem remained – the inability of the fixed expansion device to accommodate the compressor's increased mass flow rate without degrading COP. The fundamental problem is that the condenser's subcooled region must become larger as ambient temperature drops, if mass flow rate is to be maintained. It is apparent from Figures 2.2 and 2.3 that subcooling must be increased as much as possible in order to close the gap between compressor and expansion device mass flow rates. This requires that the condenser (no matter what its size) contain more charge on mild days than a TXV system that keeps enough refrigerant in the evaporator to maintain ~5°C superheat on even the mildest days. If the extra charge needed for condenser subcooling is taken from the evaporator, it produces excessive superheat which in turn lowers evaporating pressure. Increasing the size of the subcooled region also raises condensing pressure. Both of these consequences degrade COP, with the increase in condensing pressure being the largest contributor.

Thus it is not the internal volume but the behavior of the condenser, when its exit is obstructed by a fixed expansion device, which determines the mass flow rate. Similarly when the mass flow rate into the evaporator is obstructed its pressure will fall until the specific volume of the suction gas increases enough to satisfy the compressor. Increasing the evaporator's internal volume would therefore have no effect on the equilibrium suction pressure being sought.

More generally, it is apparent from the above discussion that any increase in subcooling requires an increase in subcooled heat transfer surface area of the condenser coil. More surface area means more tube volume, and more tube volume means more charge, and more charge in the condenser means that less is available for the evaporator. If increased subcooling is needed for the expansion device to match the flow rate through the compressor, there is no way (in a critically charged system) this level of subcooling can be achieved through cleverly redesigning the relative internal volumes or the circuiting of the evaporator or condenser. To increase condenser subcooling without starving the evaporator would require adding charge to the condenser at mild outdoor temperatures.

4.2 Increasing system charge when needed

Consider now the possibility of adding charge to the system during mild operating conditions – enough charge to create enough subcooling and/or high-side pressure increase to allow the fixed expansion device to meet or exceed the mass flow rate of the TXV. Although it is hard to see how such this could be done passively (perhaps with some kind of bladder accumulator with its liquid volume modulated by ambient temperature) with less complexity than a TXV, analyses were done anyway just to see whether charge management could offset the performance degradation caused by fixed expansion devices. [7]

This hypothesis was tested by simulating performance of the scroll compressor system, first with a TXV and then with the six capillary tubes. Recall in the base case simulations where charge was held constant, that subcooling also remained nearly constant as the ambient temperature decreased, because the relative internal volumes of the evaporator and condenser were properly sized to maintain a liquid seal at the TXV inlet. Therefore if subcooling and superheat were held constant at 5ºC, the NxN system of equations would call for the same (constant) amount of system charge across the entire range of ambient temperatures. Thus for the TXV system, with subcooling and superheat already at their ideal values, adding or removing charge at off-design conditions cannot improve performance.

[7] A similar concept has recently been patented for use in a transcritical CO_2 system (Manole, 2006).

For the capillary tube system an optimization analysis was conducted instead of simulations. System COP was maximized by letting charge vary, to calculate the optimal charge required at each outdoor ambient temperature. A lower bound of 5ºC was placed on subcooling to ensure a liquid seal, and a 5ºC lower bound on superheat to protect the compressor. Neither constraint proved to be binding. The optimal charge for these off-design conditions increased by only 60g as T_{amb} decreased to 29 ºC, but increased rapidly by an additional 300 grams (to 1840g) as ambient temperature fell to 19ºC. Figure 4.1 shows how adding this charge could cut the system COP penalty associated with captubes from ~5% to only about 3%. Notice that the captube system with additional charge can never achieve the efficiency of the TXV system because the role of the extra charge is to increase the captube mass flow rate by increasing both subcooling and condensing pressure. Systems with excessive subcooling will always be less efficient than one that modulates the valve opening to provide the appropriate mass flow rate while maintaining subcooling at some minimal value like 5 ºC.

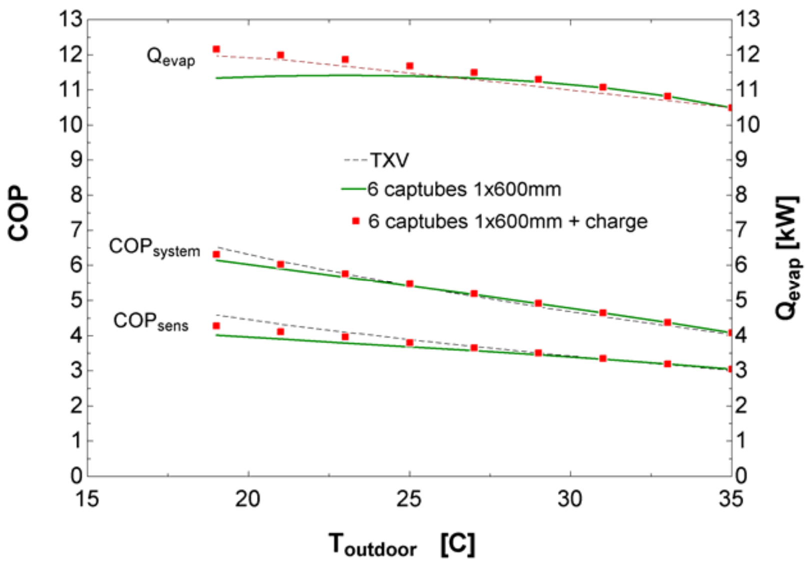

Figure 4.1. Additional charge diminishes efficiency penalty

Other aspects of system performance are affected in only minor ways by adding this extra charge during mild conditions. The subcooled zone of the condenser on a 19ºC day grows from 23 to 39% forcing the condensing temperature up 2.2ºC higher than if no charge were added. Partially offsetting these obviously negative effects were several positive impacts. The positive effects include greater improvements in evaporator performance, as the superheated zone decreased from 43% to 24%, allowing and the evaporating temperature to increase almost 3ºC compared to the fixed-charge case. The reduction in superheated area offset the loss of LMTD and increased evaporator capacity to a level even greater than that of the TXV system. This is attributable partly to the increased refrigerating effect associated with the increased subcooling. Note, however, that much of the evaporator capacity increase was latent heat transfer, as indicated by the fact that the optimally charged captube system still has a lower sensible COP than the TXV system.

Overall there is only minor benefit to be gained by somehow injecting additional charge into the captube system when outdoor temperatures are mild – even if one could find a simple way to accomplish it. The analysis also illustrates how difficult it is to anticipate (intuitively) the effects of altering a system when it is operating during [suboptimal] off-design operation.

4.3 Role of receivers

Automotive a/c systems have a receiver to hold extra charge (a hedge against leakage through flexible hoses and the compressor shaft seal), and to control the outlet state of the evaporator or condenser. Some systems use a high side receiver to control the condenser outlet at x=0 during steady state operation, but a TXV is required to ensure that the compressor is protected from liquid slugs during transients. Another method of protecting the compressor is to use a low side receiver to maintain the evaporator outlet at x=1, thus allowing the use of a fixed expansion device. Although the latter method offers no direct control of the expansion device inlet state, it does provide indirect control.

The low-side receiver controls the expansion device inlet indirectly, by providing the condenser with as much additional charge as it needs to produce the subcooling and head pressure required to match the compressor mass flow rate. As ambient temperature falls, the capillary tube or orifice tube has insufficient inlet pressure and subcooling to match the compressor's mass flow rate. In a critically-charged system, charge shifts to the condenser and the evaporator's charge is depleted. If a receiver is present to receive highly superheated vapor leaving the evaporator, that warm vapor is immediately saturated as some of the liquid evaporates. Thus the newly-evaporated liquid leaves the receiver as saturated vapor and circulates through the system. This process continues until a new equilibrium is reached when enough charge has migrated from the receiver to the evaporator to eliminate the superheat at the evaporator exit.

Referring to Figure 4.2, consider several system modifications in sequence, focusing on the mildest ambient operating temperature of 19°C. The captube-equipped system is considered first, because the differences are larger and easier to see. Replacing the TXV with captubes decreased off-design performance considerably, as shown by the difference between the dashed and solid lines. As described earlier, the main cause is the reduction of evaporating pressure (suction density) needed to bring the compressor's mass flow rate down from 63 g/s with the TXV to the 62 g/s allowed by the captube. At 19°C the captube system's condenser contains about 140 g more refrigerant than the TXV system, increasing subcooling from 5 to 8°C and producing a 21°C evaporator superheat. This, in turn caused a corresponding reduction in evaporating temperature from 9.6 to 4.8°C, contributing to the 12% shortfall in sensible COP. Interestingly, the lower mass flow rate prevented the condensing temperature from rising, so only a modest increase in subcooling was required to match the compressor's substantially reduced flow rate.

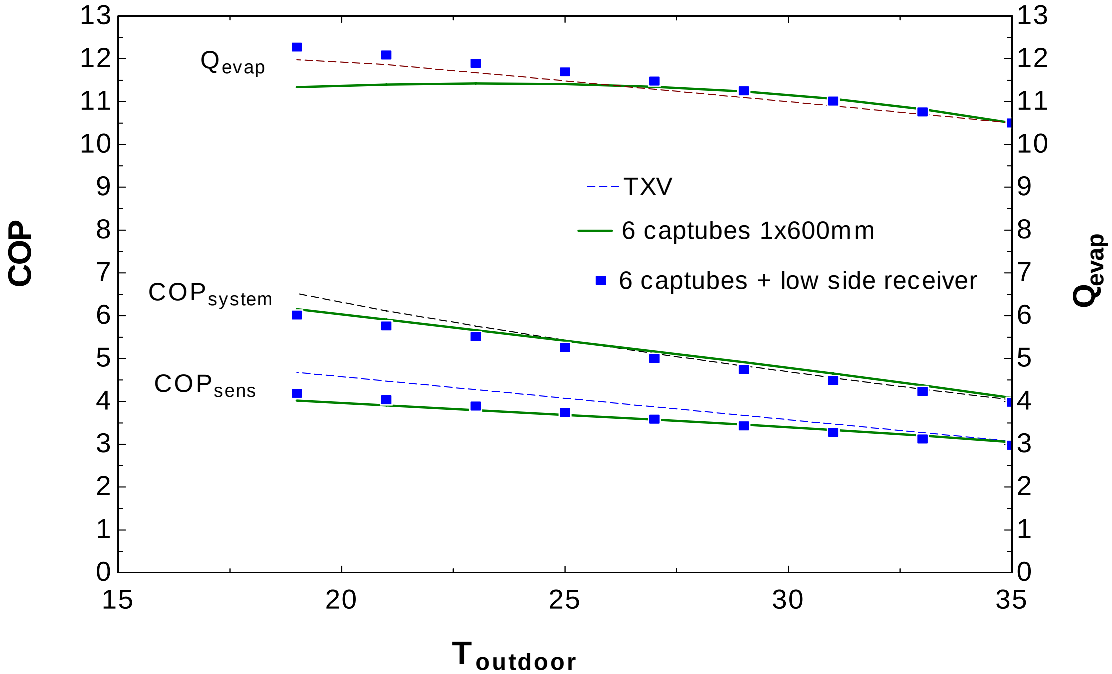

Figure 4.2. System performance with captubes and low-side receiver

Next consider the effect at this same off-design operating condition of adding a receiver to the captube system. By keeping the evaporator full (exit quality = 1.0) the evaporating temperature was restored to about the same level as the TXV system. The resulting high density of the saturated vapor leaving the receiver caused the compressor mass flow rate to actually increase from 62 to 63 g/s, which in turn required much higher subcooling and discharge pressure for the captube to match that flow. They increased from 8 to 12.9°C and from 29.5 to 33.5°C, respectively. System COP fell from 6.2 (without the receiver) to 6.0, but sensible COP actually increased from 4 to 4.2 as shown in Figure 4.2.

Thus the primary impact of the low-side receiver is to protect the compressor from the possibility of liquid slugging during transients. Compared to the captube system without receiver, it increases sensible COP a few percent, thus reducing runtime. Compared to the baseline TXV system, however, it still suffers the performance penalties shown in Figure 4.2, primarily because of the increased condensing pressure caused by supplying the extra charge needed in the condenser to produce the additional subcooling.

For the orifice tube system, the same trends are apparent but the performance degradation is smaller, even at the mildest outdoor temperature. Adding a receiver to the orifice tube system can improve its off-design capacity, but the impact on system COP is very small, as shown in Figure 4.3. The capacity increase, again, results from keeping the evaporator fully wet. However to supply the greater mass flow rate handled by the compressor at this higher evaporating pressure, the receiver had to deliver enough extra charge to the condenser to increase both subcooling by 3°C and condensing temperature by 2°C . This increased compressor work enough to offset the increase in capacity, leaving both system COP and sensible COP essentially unchanged.

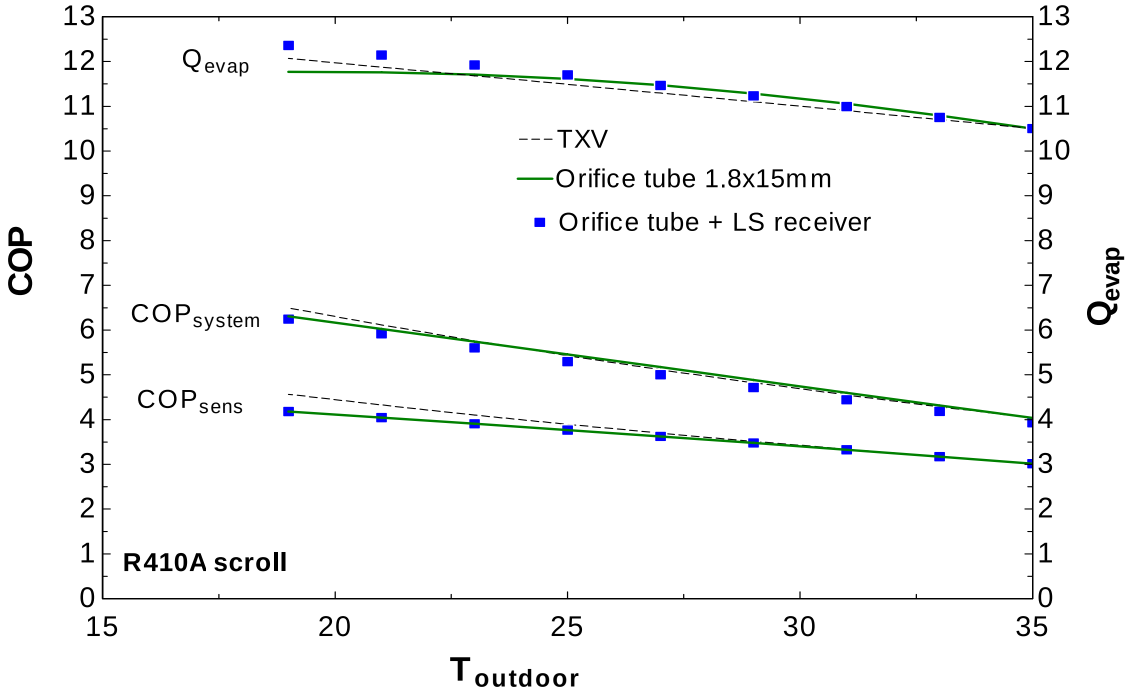

Figure 4.3. System performance with orifice tube and low-side receiver

4.4 Subsonic friction tube

This section discusses the effect on system performance of selecting captubes having an L and D large enough to generate sufficient friction to dissipate the entire pressure differential between the condenser and evaporator, without the 2-phase flow reaching sonic velocity. Thus the mass flow rate through the tube, which must match that of the compressor, now depends on the evaporating pressure: it is determined by the simple adiabatic Fanno flow relation setting the frictional pressure loss equal the difference between the system's high- and low-side pressures. This is in contrast to capillary tubes, in which the flow becomes choked at an exit pressure that is always greater than the evaporator pressure.

Recall from Figure 3.6 that the COP-maximizing strategy for selecting captubes was to choose those having relatively large L and D because they require less inlet subcooling to compensate for the loss of inlet pressure at off-design conditions. Of course if L is too large the tubes become too unwieldy and costly to install, and if D is too large the number of tubes may be less than the number of evaporator circuits to be fed. Tubes having very small L and D have exit pressures nearer the vapor dome, while the captubes analyzed earlier (6@1x600mm) had relatively large L and D and their choked exit pressures were much lower, and closer to the evaporating pressure across the entire range of operating conditions. Figure 4.4 compares the results of simulations comparing these choked captubes to a set of 6 larger subsonic tubes having 1.2 mm diameter and 1060 mm length. Their driving pressure difference between inlet and exit was therefore virtually identical to that of the captubes because the choked exits of the captubes were at pressures only slightly higher than the evaporator.

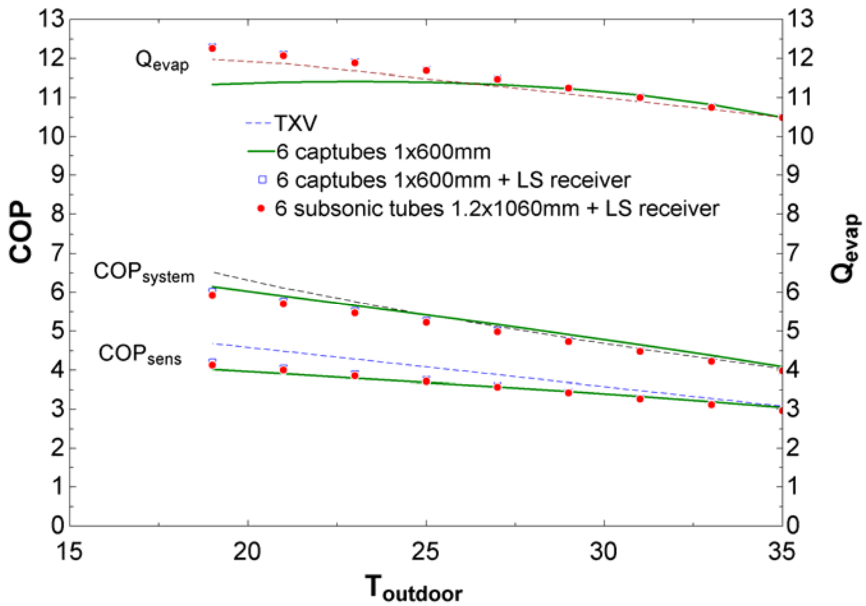

Figure 4.4. Subsonic vs. choked capillary tubes

Not surprisingly, the performance of the system equipped with subsonic tubes performed in a manner virtually identical to that of the captube system (solid dots coincident with open squares). Both systems had low-side receivers to protect the compressor from liquid slugging during transients.

4.5 Conventional liquid-suction heat exchanger (LSHX)

It is well-known that serious instabilities can occur when a LSHX is used in a critically charged system equipped with a fixed expansion device. Consider a small transient that causes liquid droplets to leave the evaporator and enter the LSHX. The resultant increase in subcooling at the inlet to the expansion device increases mass flow rate, which in turn reduces evaporator exit quality and the instability continues until the compressor is flooded. This possibility can be eliminated by installing a receiver at the evaporator outlet to ensure that only saturated vapor enters the LSHX.

It is also well-known that the thermodynamic cycle efficiency of some refrigerants like R12 and R134a, R404A and R744 can be improved substantially by using suction gas to provide liquid subcooling, thereby increasing the refrigerating effect by decreasing evaporator inlet quality. However this benefit is partially offset by the consequent increase in specific volume of the suction vapor, which increases compressor work (recall dw = v*dP). The magnitudes of these effects are strong functions of the refrigerant's thermodynamic properties at the particular operating condition. For that reason LSHX was not used in R22 a/c systems because it causes a net decrease (~5%) in cycle COP.

For R410A the same simple thermodynamic cycle calculations (assuming constant compressor isentropic and volumetric efficiencies) show that the increase in refrigerating effect is offset almost exactly by the increase in compressor power: no net benefit.

To assess the role of LSHX in the residential a/c system, it was assumed that the heat exchanger was composed of 3 microchannel tubes of 0.5 m length (middle tube carries liquid in 60 ports @ 1 mm; outer tubes carrying suction vapor have 30 ports @ 2 mm). Such heat exchangers were developed in earlier ACRC projects for systems using R744 and R404A (Boewe et al. 2001; Chandrasekharan and Bullard, 2005). Experiments demonstrated that measured effectiveness was very near the predicted value, while pressure drop could be reduced to near-negligible levels by simply increasing the number or diameter of the microchannel ports. The particular LSHX simulated here had $\varepsilon \sim 0.75$ across the entire range of outdoor air temperatures because mass flow rate was nearly constant. This LSHX geometry was selected rather arbitrarily; it is not an optimized design. Our experience indicates that with some additional material (longer or wider microchannel tubes) and careful optimization it should be possible to increase effectiveness to about 0.90 with minimal pressure drop.

Simulations were conducted to examine the effect of suction line heat exchange in systems with fixed expansion devices, together with a low-side receiver to protect the compressor and to keep the evaporator fully charged. At first glance, liquid-suction heat exchange appears to be an attractive option because it offers access to a colder heat sink that could provide the required additional subcooling at the inlet of the expansion device without increasing high-side pressure or condenser charge. By providing additional subcooling downstream of the condenser, a LSHX would not increase the size of the condenser's subcooled zone and thereby cause condensing temperature to increase.

The simulations revealed nonlinear effects that negate these expected advantages. The essential point is that a simple internal heat exchanger provides subcooling at *all* operating conditions, not just when ambient temperatures are low. In fact the internal heat exchanger is incapable of providing anywhere near the amount of *additional* subcooling that is needed at low ambient temperatures. Because its UA and effectiveness remain nearly constant (due to nearly constant mass flow), it actually transfers less heat at mild temperature conditions as the ΔT_{max} decreases with temperature lift.[8] Therefore any additional subcooling must be provided by the condenser – pre-cooling the refrigerant before it enters the internal heat exchanger. Thus in qualitative terms, it is subject to the same performance limits as a system without an internal heat exchanger.

Quantitatively, however, the LSHX actually *diminishes* the effect of increased condenser subcooling on the amount of subcooling at the inlet of the expansion device. This can be illustrated by comparing performance of the captube system and without LSHX. Recall that the captube's inability to match the compressor mass flow rate at off-design conditions not only required condensing pressure to rise, but also increased condenser subcooling as ambient temperature dropped from 35 to 19°C. Adding the LSHX provided about 14°C subcooling at the 35°C ambient condition, and only 4°C at the 19°C ambient, due to the reduced temperature lift. That forced the condenser to compensate for the loss of subcooling in the LSHX, in addition to providing the increased subcooling that would otherwise be required to match the condenser mass flow rate. To accommodate this increased subcooling, 2-phase zone of the condenser shrunk from 77% to 27%, forcing condensing temperature to reach 39°C at the 19°C ambient (compared to 29.5°C in the captube-only case).

Thus the LSHX provided subcooling, but severely degraded condenser performance at off-design conditions, imposing a severe energy penalty. Simulations of an LSHX in an orifice tube system revealed the same

[8] The suction vapor is the stream having the minimum (limiting) heat capacity, and its temperature change can be no larger than the temperature lift between the evaporator and condenser.

kind of behavior, but accurate quantification was not possible due to the 11°C upper bound on inlet subcooling imposed by the empirical equations employed.

4.6 Discharge-liquid heat exchanger (DLHX)

Instead of designing the fixed expansion device to match the compressor mass flow rate at the maximum-load design condition and trying to find a heat *sink* to provide the extra subcooling needed at moderate outdoor temperatures, consider designing it to operate adiabatically on cool days and find a heat *source* that could decrease subcooling on hot days.

The simplest way to envision this mode of operation would be to imagine an electric heater wrapped around the liquid line upstream of the expansion device (for purposes of this discussion, an orifice tube). On the coolest days (say, T_{amb} = 19°C) the expansion device matches the compressor mass flow rate exactly, and the system has about 5°C subcooling and superheat. At this point the orifice tube system would have the same performance as the TXV system. As the ambient temperature rises, up to 600W of heat is added to gradually reduce the subcooling to near zero at 35°C. Throughout this transition the mass flow rate through the orifice tube should remain roughly constant, thus mimicking the TXV system performance.

Obviously it would be inefficient to provide this heat electrically. However if it were supplied by an internal heat exchanger using condenser heat that would have otherwise been rejected, system power need not increase. Therefore a passive discharge-to-liquid heat exchanger was considered. It would transfer heat to the liquid line from the compressor discharge line, across a ΔT that is roughly proportional to the difference between the compressor discharge and the condensing temperatures. It would need to provide 600 more Watts at the 35°C ambient temperature design condition, in addition to whatever it would supply at the mild operating condition.

As a passive device, such an internal heat exchanger would have a constant heat transfer surface area. Since its goal is to maintain a constant system mass flow rate, its effectiveness would remain constant across all operating conditions. Therefore its heat transfer rate would vary with the maximum temperature differential between the two streams (approximately compressor discharge temperature minus condensing temperature). This difference is about a factor of 1.5 larger at the design condition than at the mild operating condition (30°C and 20°C, respectively). Therefore discharge-to-liquid heat exchanger would have to be designed to transfer 1200 W at the mild condition and 1.5x1200 = 1800 W at the 35°C ambient in order to span the 600 W range needed to decrease subcooling by 5°C.

Unfortunately this strategy proves infeasible because of the large (1200 W) requirement at the mild operating condition, where subcooling needs to be 5°C at the *exit* of the internal heat exchanger. This requires that the *inlet* subcooling be ~15°C, which in turn means that the condensing temperature must be at least 15°C above the ambient temperature at the mild-day operating condition. Clearly operating at such a high condensing temperature would offset any benefits obtained by modulating the amount subcooling at the expansion device inlet.

This failure illustrates an inherent shortcoming a passive device as a source (or sink) of energy needed for control purposes. The hypothetical electric heater described at the beginning of this section provided 0 W at the mild condition and 600 W at the high ambient temperature. In order for the passive internal heat exchanger to meet this criterion it would need to tap a heat source/sink having a temperature difference that varied from 0 to some larger value.

4.7 Modulating fan and blower speeds

The cost of variable speed is decreasing, and they offer comfort- and noise-related benefits. Despite the fact that they might not be amenable to very simple control strategies (e.g. speed as a function of ambient air temperature), their compatibility with fixed expansion devices was explored in two sets of analyses of a system equipped with captubes and a scroll compressor, just to see if the approach might hold promise.

Variable speed condenser fan. Simulations conducted for a captube system showed that decreasing condenser fan speed as outdoor air temperature dropped was not an efficient way to increase flow through the expansion device. By raising the condensing pressure it had the desired effects of increasing mass flow rate, preventing the evaporating temperature from falling too far, and keeping the evaporator fully wetted at mild conditions. However by decreasing the magnitude of the condenser UA and heat sink, the condensing temperature increased so much that the higher temperature lift offset all these advantages. Instead of improving the captube system's performance the net effect was to decrease COP an additional 20% at the mildest off-design condition.

Variable speed blower. The same approach was applied for the blower with more success. The simulation indicated that it is indeed possible to maintain a constant evaporator superheat across the entire range of outdoor temperatures by adjusting blower speed. However, the captube system's COP was degraded by 5% at the 19ºC condition when blower speed was lowered to maintain 5° C of superheat. This decrease in COP was caused by a decrease in evaporating temperature. By that measure it is not feasible to add a variable speed blower to a captube system. The off-design sensible COP decreased slightly faster with ambient temperature in this case (variable speed blower) than in the case of the variable speed condenser fan. Thus neither strategy (varying fan or blower speed to maintain constant superheat) proved successful. They both degrade performance. The single speed fan and blower system provided the higher off-design sensible COP.

4.8 Variable speed compressor

Another set of computational experiments tested the compatibility of fixed expansion devices with variable speed compressors. System simulation results are shown in Figure 4.5 for a simplified TXV system, to illustrate the range of refrigerant flow rates that a fixed expansion device would have to handle.[9] The nearly horizontal solid line in the middle shows the mass flow rate delivered by a single-speed scroll compressor sized for the 35ºC design condition, after the mass flow rates through the valve and compressor equalize. The sloped solid line shows the mass flow rate delivered by a variable speed compressor having a turndown ratio of only 3; at maximum speed it can meet loads on a 40ºC day, and it would cycle to meet loads at ambient temperatures <27ºC. At all other conditions it adjusts to match exactly the flow rate through the expansion device. Some compressors can achieve turndown ratios of 5 or more, and would therefore span an even greater range mass flow rates. Note that this range greatly exceeds those which could be accommodated by either the capillary tubes or the orifice tube shown in Figures 3.2 and 3.3.

[9] In this illustrative analysis, evaporating temperature was held constant at 12ºC, equivalent to assuming that a variable speed blower was used to control dehumidification.

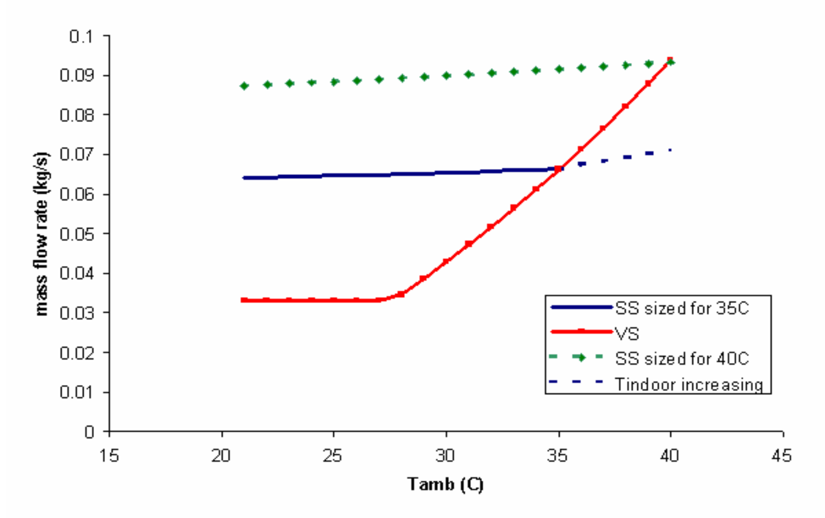

Figure 4.5. Mass flow rate range for variable speed compressor

More detailed system simulation results confirmed that COP is substantially degraded when a fixed expansion device is used with a variable speed compressor. This is due to the fact that the only way that a fixed expansion device can match the compressor's low mass flow rate is to force the system into an operating mode that produces a two-phase condenser outlet condition and a low condensing pressure. Therefore at low mild outdoor conditions and very low refrigerant flow rates the fixed expansion device is unable to maintain the saturation temperature lift required for the heat exchangers to function properly. This results in the evaporator and compressor being flooded with liquid at such conditions.

4.9 A promising approach: 2-phase internal heat exchanger with receiver

Recall that a low-side receiver ensures full utilization of the evaporator's heat transfer surface. At the same time it holds excess charge not needed at certain operating conditions, thus allowing the condenser charge inventory to reach a level that increases subcooling and/or condensing pressure to the extent necessary for the fixed expansion device (orifice tube) to match the compressor's mass flow rate.

Recall also that the analysis in section 4.5 considered a conventional liquid-suction heat exchanger (LSHX) inserted in a system designed and optimized for the ARI-A capacity rating condition. The 5°C superheat and subcooling targets for the evaporator and condenser exit states were not changed. COP was severely degraded at mild conditions.

In contrast, the approach considered here uses a low-side receiver and a LSHX that is designed to optimize system performance at the mild 19°C ambient operating condition. At that condition the LSHX is sized to provide almost all the subcooling required to deliver the same mass flow rate as the baseline TXV system (about 10-12°C, as indicated in Figure 3-10). Since the temperature lift (ΔT_{max}) at the mild condition is approximately 20°C, a LSHX having reasonable effectiveness should be able to accomplish this task: refrigerant would leave the condenser with only a few degrees of subcooling and the LSHX would provide the rest. Unlike the options investigated earlier, the condenser would be operating very efficiently at this 19°C ambient condition, with only minimal subcooling.

As ambient temperature rises to 35°C, several components require less charge – particularly the condenser (150 g). Although the vapor density increases due to the high condensing pressure, this effect is more than offset by the condenser charge decreasing as the condenser outlet quality rises to 14%. The subcooling at the orifice tube inlet decreases steadily from ~10°C to ~5°C, just as it did in the orifice tube-only case, while the hot side of the LSHX fills with 2-phase flow. Since there is no large subcooled zone in the condenser, its pressure is far lower than in the orifice tube-only case. The amount of charge in the microchannel LSHX is negligible. The evaporator charge decreases ~100 g as ambient temperature rises because of the change in inlet quality: from 5% at the 19°C ambient condition, to 20% at 35°C.

The orifice tube size is identical to that needed when an LSHX is not present, because of the requirement to provide the mass flow necessary to achieve 10.5 kW capacity at the ARI-A capacity rating condition. The system charge required at that condition is also the same, because the LSHX contains only a negligible amount.

At mild conditions more charge is required, so superheat will appear in the evaporator at the 19°C condition. Charge should then be added until measurable subcooling appears at the condenser exit. The sizing of the LSHX ensures that the necessary ~10°C subcooling is achieved at the expansion device inlet.

Note that this system is critically charged at the 19°C operating condition, in the sense that the receiver contains liquid only at warmer ambient temperatures. At that point it contains only vapor, and the LSHX provides the ~10°C subcooling needed by the orifice tube at that condition. Then as the ambient temperature increases to 35°C, the receiver collects the excess charge no longer needed by the evaporator and condenser. About 95% of the LSHX length would be filled with 2-phase flow from the condenser. The remaining 5% of the length would provide ~5°C subcooling at the orifice tube inlet.

System simulation analyses showed that ~10°C subcooling at the mild operating condition could be achieved with a relatively simple LSHX made of a "sandwich" of three microchannel tubes about 0.7 m long. The two outside tubes would have thirty 2mm ports, and the middle (counterflow) tube would have 60 ports @ 1mm. Many other designs would also work; no effort was made to optimize that component design for these preliminary calculations.

The analyses summarized in Table 1.1 show that this simple, totally passive approach could nearly match the TXV system's evaporator and condenser performance across the entire range of operating conditions. The only cost would be the requirement for three components to replace the TXV: an orifice tube; low-side receiver; and LSHX. Another potential disadvantage is that any leakage of charge would impair performance at the mildest operating condition, compared to the option of using only the orifice tube and receiver that could store excess charge as insurance against leaks.

A more relevant comparison with a receiver-equipped orifice tube system is shown in Table 4.1 below. The necessity for using a receiver in systems equipped with fixed expansion devices was established in Section 4.3. Therefore the only additional component required for this option is the LSHX.

The main difference in the performance of the two systems is the lower condensing and evaporating temperatures produced by the LSHX. By moving the subcooling out of the condenser, the more effective 2-phase zone extends all the way to the exit. Although the subcooling at the inlet to the orifice tube is nearly the same for both systems, the enthalpy is reduced due to the lower condensing pressure. As a result, the quality is lower at the evaporator inlet, so evaporation of the additional liquid produces a consistently greater cooling capacity at the off-design conditions. This need for increased LMTD accounts for the lower evaporating temperatures seen in Table 4.1. Further analysis is needed to determine whether this increased capacity is overwhelmed by the increased compressor work required to handle the lower-density suction vapor leaving the LSHX.

Table 4.1. Adding LSHX to orifice tube system with receiver

	Orifice tube + receiver vs. adding LSHX					
	35		**27**		**19**	
Q_{sens}	10.5	10.5	11.3	11.5	12.1	12.4
Q_{lat}	2.6	2.6	3.2	3.3	3.8	3.8
T_e [°C]	11.1	10.8	10.3	10.0	9.6	9.3
T_c [°C]	45.2	43.7	38.2	36.8	31.8	30.0
ΔT_{sub}[°C]	5.0	2ph/4.0*	7.5	2ph/6.0*	10.4	3.4/11.0*
ΔT_{sup}[°C]	5.0	0	5.0	0	5.0	0
* indicates subcooling at condenser exit and LSHX exit						

Unfortunately it is not possible to compare system COP results for this option in Table 4.1. The reason has to do with shortcomings in the compressor submodels currently available for use in system simulations (whether based on prescribing isentropic efficiencies, or based on compressor calorimeter "maps" which use data obtained at constant suction temperature). Neither type of model accounts for the effect of suction temperature variations on mass flow and power, especially for compressors with low-side sumps which use suction gas for motor cooling. The internal heating due to motor cooling will also vary with suction temperature. The unknown variations in refrigerant density *at the suction port* inside the compressor shell can affect both mass flow rate and isentropic efficiency. In systems without internal heat exchangers, suction gas temperatures remain relatively constant across the range of operating conditions, so this has not been an issue for a/c systems in the past. While an accurate assessment of compressor power must await experimental results or more detailed compressor modeling, the results shown in Table 4.1 suggest that the overall system response is very favorable.

These results may call for reconsideration of the role of LSHX in systems using all the new HFC's. Recall that the thermodynamic properties of R22 imposed a ~5% COP penalty, so internal heat exchange was never employed in those systems. All R134a refrigerators and some R404A display cases successfully employ LSHX in an attempt to capture the ~5-10% thermodynamic benefit. Rough approximations (e.g. adiabatic compression and constant isentropic efficiency) suggest that an LSHX should have no effect, positive or negative, on COP due to the thermodynamic properties of R410A, because the increased refrigerating effect is exactly offset by the penalty at the compressor. A more precise evaluation of this approach must await the available of compressor test data in which suction gas temperature is varied while suction and discharge temperatures are held constant.

References

Kim, Y. and D. L. O'Neal, "Two-Phase Flow of Refrigerant-22 through Short-Tube Orifices," ASHRAE Transactions, Vol. 100:1, 323-334, 1994.

Kim, Yongchan, V. Payne, J. Choi, and P. Domanski, "Mass flow Rate of R-410A through Short Tubes Working Near the Critical Point," *International Journal of Refrigeration*, 28:4, 547-553, 2005.

Manole, D., (August 29, 2006) Transcritical Vapor Compression System and Method of Operating Including Refrigerant Storage Tank and Non-Variable Expansion Device. US Patent 7,096,679 B2.

Payne, W. V. and D. L. O'Neal, "Multiphase Flow of Refrigerant 410A through Short Tube Orifices," ASHRAE Transactions, Vol. 105:2, 66-74, 1999.

Simões-Moreira, J. R. and C. W. Bullard, "Pressure Drop and Flashing Mechanisms in Refrigerant Expansion Devices," *International Journal of Refrigeration*, 26:7, 840-848, 2003.

Singh, G. M., P. S. Hrnjak, and C. W. Bullard, "Flow of Refrigerant 134a Through Orifice Tubes," *International Journal of HVAC&R Research*, 7:3, 245-262, 2001.

Appendix A. Simulation Results for 14 System Configurations

1. Baseline system (scroll compressor) with TXV
2. Baseline with short tube
3. Baseline with short tube and receiver
4. Baseline with captube
5. Baseline with captube and receiver
6. Baseline with captube and variable charge
7. Baseline with subsonic tube
8. Baseline with short tube and receiver and conventional 1-phase LSHX
9. Baseline with captube and receiver and conventional 1-phase LSHX
10. Baseline with short tube and receiver and 2-phase LSHX
11. Baseline with captube and variable speed condenser fan
12. Baseline with captube and variable speed blower
13. Baseline with TXV and reciprocating compressor
14. Baseline with captube and reciprocating compressor

Taevapin	D	L	Vdotcond	Vdotevap	charge	Tamb	dTsub	dTsup	mdotr	beta	Tcrit	Trevapin	Tsatoutc	COPsys	xevapin	Tsatsuc	COPsens	mcond	mevap	mliqline	mcomp	fsubc	Tsatdis	Taoutc	Qevap	Qsens	f2phc	Tdis	xcondout	fsupevap	EER	Taoutevap	Troutcond	Wdotcomp	Wdotcond	Wdotevap	SHR	Tsuc	Thotout	eps_ihx	rhosuc
TXV BASELINE W/ SCROLL																																									
26.7	0.00182	0.015	1.4	0.5208	1.54	35	5	5	0.063	0.92			43.6	4	0.22	10.9	3	0.91	0.22	0.25	0.16	0.12	45	42.9	10.5	7.9	0.79	70.7	-0.06	0.06	13.8	13.8	38.6	2.298	0.161	0.095	0.75				41.4
26.7			1.4	0.5208	1.54	33	5	5	0.063	0.92			41.7	4.3	0.21	10.7	3.2	0.9	0.23	0.25	0.16	0.12	43.1	40.9	10.7	8	0.8	67.9	-0.06	0.06	14.6	13.7	36.7	2.196	0.161	0.095	0.74				41.2
26.7			1.4	0.5208	1.54	31	5	5	0.063	0.92			39.8	4.5	0.19	10.6	3.3	0.89	0.24	0.25	0.16	0.12	41.2	38.9	10.9	8	0.8	65	-0.06	0.06	15.5	13.5	34.7	2.098	0.161	0.095	0.74				41
26.7			1.4	0.5208	1.54	29	5	5	0.063	0.92			37.8	4.8	0.18	10.4	3.5	0.88	0.24	0.26	0.16	0.11	39.4	36.9	11.1	8.1	0.81	62.2	-0.05	0.06	16.5	13.4	32.7	2.003	0.161	0.095	0.73				40.8
26.7			1.4	0.5208	1.54	27	5	5	0.063	0.92			35.8	5.1	0.16	10.2	3.7	0.87	0.25	0.26	0.16	0.11	37.5	34.9	11.3	8.2	0.82	59.4	-0.05	0.06	17.5	13.3	30.8	1.911	0.161	0.095	0.73				40.6
26.7			1.4	0.5208	1.54	25	5	5	0.062	0.92			33.8	5.4	0.15	10.1	3.9	0.85	0.26	0.26	0.16	0.11	35.7	32.9	11.5	8.3	0.82	56.7	-0.05	0.06	18.5	13.1	28.9	1.822	0.161	0.095	0.72				40.5
26.7			1.4	0.5208	1.54	23	4.9	5	0.062	0.92			31.8	5.8	0.13	9.9	4.1	0.84	0.27	0.26	0.16	0.1	33.8	30.9	11.7	8.4	0.83	53.9	-0.05	0.06	19.6	13	26.9	1.735	0.161	0.095	0.72				40.3
26.7			1.4	0.5208	1.54	21	4.8	5	0.062	0.92			29.8	6.1	0.12	9.8	4.3	0.83	0.28	0.27	0.16	0.1	31.9	28.9	11.9	8.4	0.84	51.2	-0.05	0.06	20.9	12.9	25	1.649	0.161	0.095	0.71				40.1
26.7			1.4	0.5208	1.54	19	4.7	5	0.062	0.92			27.8	6.5	0.1	9.6	4.6	0.82	0.29	0.27	0.16	0.09	30.1	26.9	12.1	8.5	0.85	48.5	-0.04	0.05	22.2	12.7	23.1	1.565	0.161	0.095	0.71				39.9
SHORT TUBE BASE																																									
26.7	0.00182	0.015	1.4	0.5208	1.54	35	5	5.1	0.063	0.92			43.6	4	0.22	10.9	3	0.91	0.22	0.25	0.16	0.12	45	42.9	10.5	7.9	0.79	70.8	-0.06	0.06	13.8	13.8	38.6	2.298	0.161	0.095	0.75				41.4
26.7	0.00182	0.015	1.4	0.5208	1.54	33	5.4	7.1	0.062	0.92			41.8	4.3	0.21	10.5	3.2	0.9	0.22	0.25	0.16	0.13	43.1	40.9	10.8	8	0.78	69.8	-0.06	0.09	14.7	13.7	36.4	2.199	0.161	0.095	0.74				40.3
26.7	0.00182	0.015	1.4	0.5208	1.54	31	5.7	9.1	0.062	0.92			39.9	4.6	0.19	10	3.3	0.9	0.22	0.25	0.16	0.14	41.3	39	11.1	8	0.78	68.8	-0.06	0.12	15.7	13.6	34.1	2.104	0.161	0.095	0.73				39.2
26.7	0.00182	0.015	1.4	0.5208	1.54	29	6.1	11	0.061	0.92			38	4.9	0.17	9.5	3.5	0.9	0.22	0.26	0.16	0.15	39.4	37	11.3	8.1	0.77	67.8	-0.07	0.16	16.7	13.5	31.9	2.011	0.161	0.095	0.71				38.1
26.7	0.00182	0.015	1.4	0.5208	1.54	27	6.4	13	0.06	0.92			36	5.2	0.16	8.9	3.6	0.9	0.22	0.26	0.16	0.15	37.5	35	11.5	8.1	0.76	66.7	-0.07	0.2	17.6	13.5	29.6	1.921	0.161	0.095	0.7				37
26.7	0.00182	0.015	1.4	0.5208	1.54	25	6.8	15	0.059	0.92			34.1	5.5	0.14	8.3	3.8	0.91	0.21	0.26	0.15	0.16	35.6	32.9	11.6	8.1	0.76	65.5	-0.07	0.24	18.6	13.6	27.3	1.832	0.161	0.095	0.69				35.9
26.7	0.00182	0.015	1.4	0.5208	1.54	23	7.1	16	0.058	0.92			32.1	5.7	0.13	7.6	3.9	0.91	0.21	0.27	0.15	0.18	33.7	30.9	11.7	8	0.75	64.2	-0.07	0.28	19.6	13.6	25	1.744	0.161	0.095	0.68				34.8
26.7	0.00182	0.015	1.4	0.5208	1.54	21	7.4	18	0.057	0.92			30.1	6	0.12	6.8	4	0.92	0.2	0.27	0.15	0.19	31.8	28.8	11.8	7.9	0.74	62.8	-0.07	0.32	20.6	13.7	22.8	1.658	0.161	0.095	0.67				33.7
26.7	0.00182	0.015	1.4	0.5208	1.54	19	7.7	19	0.055	0.92			28.1	6.3	0.1	6	4.2	0.92	0.2	0.27	0.15	0.2	29.8	26.7	11.8	7.8	0.73	61.3	-0.07	0.36	21.5	13.9	20.5	1.572	0.16	0.095	0.67				32.6
SHORT TUBE BASE W/ RECEIVER																																									
26.7	0.00184	0.015	1.4	0.518	1.55	35	5	1	0.065	0.94			43.8	3.9	0.22	11.1	2.9	0.91	0.23	0.25	0.17	0.12	45.2	42.9	10.5	7.9	0.79	67.6	-0.06	0.01	13.4	13.8	38.8	2.368	0.161	0.094	0.75				43
26.7	0.00184	0.015	1.4	0.518	1.58	33	5.6	1	0.065	0.94			42	4.2	0.2	10.9	3.1	0.92	0.24	0.25	0.17	0.14	43.4	41	10.7	8	0.78	64.9	-0.07	0.01	14.3	13.6	36.4	2.269	0.161	0.094	0.74				42.7
26.7	0.00184	0.015	1.4	0.518	1.6	31	6.2	1	0.065	0.94			40.2	4.4	0.19	10.7	3.3	0.93	0.25	0.25	0.16	0.15	41.7	39	11	8.1	0.77	62.2	-0.07	0.01	15.2	13.4	34	2.174	0.161	0.094	0.73				42.5
26.7	0.00184	0.015	1.4	0.518	1.63	29	6.8	1	0.064	0.94			38.4	4.7	0.17	10.5	3.4	0.95	0.26	0.26	0.16	0.17	39.9	37	11.2	8.2	0.76	59.6	-0.07	0.01	16.1	13.3	31.6	2.083	0.161	0.094	0.73				42.2
26.7	0.00184	0.015	1.4	0.518	1.66	27	7.5	1	0.064	0.94			36.7	5	0.15	10.3	3.6	0.97	0.27	0.26	0.16	0.2	38.2	35.1	11.5	8.3	0.74	57	-0.08	0.01	17.1	13.1	29.2	1.997	0.161	0.094	0.72				42
26.7	0.00184	0.015	1.4	0.518	1.7	25	8.2	1	0.064	0.94			35	5.3	0.13	10.1	3.8	0.99	0.28	0.26	0.16	0.22	36.6	33.1	11.7	8.3	0.71	54.5	-0.08	0.01	18.1	12.9	26.8	1.914	0.161	0.094	0.71				41.8
26.7	0.00184	0.015	1.4	0.518	1.75	23	8.9	1	0.064	0.94			33.3	5.6	0.11	9.9	4	1.02	0.29	0.27	0.16	0.25	34.9	31.1	11.9	8.4	0.69	52.1	-0.09	0.01	19.1	12.8	24.4	1.835	0.161	0.094	0.71				41.5
26.7	0.00184	0.015	1.4	0.518	1.8	21	9.7	1	0.064	0.94			31.7	5.9	0.09	9.8	4.1	1.06	0.3	0.27	0.16	0.29	33.3	29.1	12.1	8.5	0.66	49.7	-0.09	0.01	20.2	12.6	22.1	1.76	0.161	0.094	0.7				41.3
26.7	0.00184	0.015	1.4	0.518	1.86	19	10	1	0.064	0.94			30.1	6.2	0.08	9.6	4.3	1.11	0.32	0.27	0.16	0.33	31.8	27.2	12.4	8.6	0.62	47.4	-0.1	0.01	21.3	12.5	19.8	1.687	0.161	0.094	0.7				41.1
CAPTUBE BASE																																									
26.7	0.00102	0.6	1.4	0.5222	1.54	35	5	5	0.062	0.91				4.1	0.22	10.9	3.1	0.91	0.23	0.25	0.16	0.12	44.9	42.8	10.5	7.9	0.79	70.7	-0.06	0.06	13.9	13.9	38.6	2.268	0.161	0.095	0.75	16	38.5	0	41
26.7	0.00102	0.6	1.4	0.5222	1.54	33	5.6	8.4	0.061	0.91				4.4	0.2	10.3	3.2	0.91	0.22	0.25	0.16	0.14	43.1	40.9	10.8	7.9	0.77	70.9	-0.07	0.11	14.9	13.7	36.2	2.172	0.161	0.095	0.73	18.7	36.2	0	40
26.7	0.00102	0.6	1.4	0.5222	1.54	31	6	11	0.06	0.91				4.7	0.18	9.6	3.3	0.92	0.22	0.25	0.16	0.15	41.2	38.9	11.1	8	0.76	70.6	-0.07	0.16	15.9	13.7	33.8	2.079	0.161	0.095	0.72	20.8	33.8	0	38
26.7	0.00102	0.6	1.4	0.5222	1.54	29	6.5	13	0.059	0.91				4.9	0.17	9	3.5	0.92	0.21	0.26	0.16	0.16	39.3	36.9	11.2	7.9	0.75	69.9	-0.07	0.21	16.8	13.8	31.5	1.987	0.161	0.095	0.71	22.3	31.5	0	37
26.7	0.00102	0.6	1.4	0.5222	1.54	27	6.9	15	0.058	0.91				5.2	0.15	8.2	3.6	0.93	0.2	0.26	0.15	0.18	37.4	34.8	11.3	7.9	0.74	68.8	-0.07	0.26	17.6	13.9	29.2	1.897	0.161	0.095	0.69	23.5	29.1	0	36
26.7	0.00102	0.6	1.4	0.5222	1.54	25	7.2	17	0.057	0.91				5.4	0.14	7.4	3.7	0.93	0.2	0.26	0.15	0.19	35.5	32.8	11.4	7.8	0.73	67.6	-0.07	0.31	18.5	14	26.9	1.808	0.161	0.095	0.68	24.4	26.8	0	34
26.7	0.00102	0.6	1.4	0.5222	1.54	23	7.5	18	0.055	0.91				5.7	0.13	6.6	3.8	0.94	0.19	0.27	0.15	0.2	33.5	30.7	11.4	7.7	0.72	66.2	-0.07	0.35	19.3	14.2	24.6	1.721	0.16	0.095	0.67	25	24.6	0	33
26.7	0.00102	0.6	1.4	0.5222	1.54	21	7.7	20	0.054	0.91				5.9	0.12	5.7	3.9	0.95	0.18	0.27	0.14	0.21	31.5	28.5	11.4	7.6	0.71	64.6	-0.07	0.39	20.2	14.4	22.3	1.635	0.16	0.096	0.66	25.5	22.3	0	32
26.7	0.00102	0.6	1.4	0.5222	1.54	19	8	21	0.053	0.91				6.2	0.11	4.8	4	0.96	0.17	0.27	0.14	0.23	29.5	26.4	11.3	7.4	0.7	62.9	-0.07	0.43	21	14.6	20.1	1.549	0.16	0.096	0.66	25.8	20.1	0	31
CAPTUBE BASE W/RECEIVER																																									
26.7	0.00103	0.6	1.4	0.5193	1.56	35	5	1	0.064	0.93	16.3	12		4	0.22	11.1	3	0.91	0.24	0.25	0.17	0.12	45.1	42.9	10.5	7.9	0.79	67.6	-0.06	0.01	13.6	13.8	38.7	2.335	0.161	0.094	0.75	12.1	38.7	0	43
26.7	0.00103	0.6	1.4	0.5193	1.6	33	6	1	0.064	0.93	15.5	11.8		4.2	0.19	10.9	3.1	0.94	0.24	0.25	0.16	0.15	43.4	40.9	10.8	8	0.77	65	-0.07	0.01	14.4	13.6	36.1	2.242	0.161	0.094	0.74	11.9	36	0	43
26.7	0.00103	0.6	1.4	0.5193	1.64	31	7	1	0.064	0.93	14.6	11.6		4.5	0.17	10.7	3.3	0.97	0.25	0.25	0.16	0.19	41.8	39	11	8.1	0.73	62.5	-0.08	0.01	15.3	13.5	33.4	2.155	0.161	0.094	0.73	11.7	33.3	0	42
26.7	0.00103	0.6	1.4	0.5193	1.7	29	8	1	0.064	0.93	13.8	11.4		4.8	0.15	10.5	3.4	1.01	0.27	0.26	0.16	0.23	40.2	37	11.3	8.2	0.7	60	-0.09	0.01	16.2	13.3	30.8	2.073	0.161	0.094	0.73	11.5	30.8	0	42
26.7	0.00103	0.6	1.4	0.5193	1.76	27	9.1	1	0.063	0.93	13.1	11.2		5	0.13	10.3	3.6	1.06	0.28	0.26	0.16	0.28	38.7	35	11.5	8.3	0.66	57.7	-0.1	0.01	17.1	13.1	28.3	1.997	0.161	0.094	0.72	11.3	28.2	0	42
26.7	0.00103	0.6	1.4	0.5193	1.83	25	10	1	0.063	0.93	12.3	11		5.3	0.12	10.1	3.7	1.12	0.29	0.26	0.16	0.33	37.3	33.1	11.7	8.3	0.61	55.5	-0.1	0.01	18	13	25.9	1.927	0.161	0.094	0.71	11.2	25.8	0	42
26.7	0.00103	0.6	1.4	0.5193	1.91	23	11	1	0.063	0.93	11.6	10.9		5.5	0.1	10	3.9	1.18	0.3	0.27	0.16	0.38	36	31.1	11.9	8.4	0.56	53.5	-0.11	0.01	18.8	12.9	23.6	1.863	0.161	0.094	0.71	11	23.6	0	42
26.7	0.00103	0.6	1.4	0.5193	2	21	12	1	0.063	0.93	10.9	10.7		5.8	0.08	9.8	4	1.26	0.31	0.27	0.16	0.44	34.7	29.1	12.1	8.5	0.51	51.5	-0.12	0.01	19.7	12.7	21.4	1.804	0.161	0.094	0.7	10.8	21.4	0	41
26.7	0.00103	0.6	1.4	0.5193	2.09	19	13	1	0.063	0.93	10.2	10.6		6	0.07	9.7	4.2	1.33	0.32	0.27	0.16	0.5	33.5	27.1	12.3	8.6	0.46	49.6	-0.13	0.01	20.5	12.6	19.2	1.748	0.161	0.094	0.7	10.7	19.2	0	41
CAPTUBE BASE W/CHARGE																																									
26.7	0.00102	0.6	1.4	0.5222	1.54	35	5	5	0.062	0.91	16	11.8		4.1	0.22	10.9	3.1	0.91	0.23	0.25	0.16	0.12	44.9	42.8	10.5	7.9	0.79	70.7	-0.06	0.06	13.9	13.9	38.6	2.268	0.161	0.095	0.75	16	38.5	0	41
26.7	0.00102	0.6	1.4	0.5222	1.54	33	5.5	8.6	0.061	0.91	14.8	11.2		4.4	0.2	10.2	3.2	0.91	0.22	0.25	0.16	0.14	43	40.9	10.8	7.9	0.77	71.1	-0.07	0.12	14.9	13.8	36.2	2.172	0.161	0.095	0.73	18.9	36.2	0	40
26.7	0.00102	0.6	1.4	0.5222	1.57	31	6.4	9.4	0.061	0.91	13.9	10.9		4.7	0.18	9.9	3.4	0.93	0.22	0.25	0.16	0.16	41.3	38.9	11.1	8	0.75	69.2	-0.07	0.13	15.9	13.6	33.6	2.084	0.161	0.095	0.72	19.4	33.6	0	39
26.7	0.00102	0.6	1.4	0.5222	1.6	29	7.2	11	0.06	0.91	13	10.5		4.9	0.16	9.6	3.5	0.96	0.23	0.26	0.16	0.19	39.6	37	11.3	8.1	0.72	67.7	-0.08	0.15	16.8	13.5	31.1	1.999	0.161	0.095	0.72	20.2	31.1	0	38
26.7	0.00102	0.6	1.4	0.5222	1.62	27	7.8	12	0.06	0.91	11.9	9.99		5.2	0.15	9	3.7	0.98	0.23	0.26	0.16	0.22	37.9	35	11.5	8.1	0.7	66.7	-0.08	0.18	17.7	13.5	28.7	1.915	0.161	0.095	0.71	21.4	28.7	0	37
26.7	0.00102	0.6	1.4	0.5222	1.66	25	8.6	13	0.059	0.91	10.9	9.6		5.5	0.13	8.6	3.8	1.01	0.23	0.26	0.16	0.25	36.2	33	11.7	8.2	0.67	65.1	-0.09	0.2	18.7	13.4	26.3	1.837	0.161	0.095	0.7	22	26.3	0	37
26.7	0.00102	0.6	1.4	0.5222	1.72	23	9.5	14	0.059	0.91	10.1	9.31		5.8	0.11	8.4	4	1.06	0.24	0.27	0.15	0.3	34.7	31	11.9	8.2	0.63	63.2	-0.09	0.21	19.6	13.3	23.9	1.767	0.161	0.095	0.69	22.2	23.9	0	36
26.7	0.00102	0.6	1.4	0.5222	1.76	21	10	15	0.058	0.91	9.08	8.83		6	0.1	7.9	4.1	1.1	0.24	0.27	0.15	0.33	33.1	28.9	12	8.2	0.59	61.9	-0.1	0.24	20.6	13.3	21.6	1.694	0.161	0.095	0.68	22.9	21.6	0	35
26.7	0.00102	0.6	1.4	0.5222	1.84	19	11	15	0.058	0.91	8.31	8.58		6.3	0.08	7.6	4.3	1.17	0.24	0.27	0.15	0.39	31.7	26.9	12.2	8.3	0.54	60.1	-0.1	0.24	21.5	13.2	19.4	1.634	0.161	0.095	0.68	23	19.4	0	35
CAPTUBE BASE SUBSONIC																																									
26.7	0.00115	1.057	1.4	0.5193	1.56	35	5	1	0.064	0.93		12		4	0.22	11.1	3	0.91	0.24	0.25	0.17	0.12	45.1	42.9	10.5	7.9	0.79	67.6	-0.06	0.01	13.6	13.8	38.7	2.334	0.161	0.094	0.75	12.1	38.7	0	43
26.7	0.00115	1.057	1.4	0.5193	1.6	33	6.1	1	0.064	0.93		11.8		4.2	0.19	10.8	3.1	0.94	0.25	0.25	0.16	0.16	43.4	40.9	10.8	8	0.76	65	-0.07	0.01	14.4	13.6	36	2.243	0.161	0.094	0.74	11.9	35.9	0	43
26.7	0.00115	1.057	1.4	0.5193	1.65	31	7.2	1	0.064	0.93		11.6		4.5	0.17	10.7	3.3	0.98	0.26	0.25	0.16	0.2	41.8	39	11	8.1	0.73	62.5	-0.08	0.01	15.3	13.4	33.3	2.157	0.161	0.094	0.73	11.7	33.2	0	42
26.7	0.00115	1.057	1.4	0.5193	1.71	29	8.3	1	0.064	0.93		11.4		4.7	0.15	10.5	3.4	1.02	0.27	0.26	0.16	0.24	40.3	37	11.3	8.2	0.69	60.2	-0.09	0.01	16.2	13.3	30.7	2.077	0.161	0.094	0.73	11.5	30.6	0	42
26.7	0.00115	1.057	1.4	0.5193	1.78	27	9.4	1	0.063	0.93		11.2		5	0.13	10.3	3.6	1.08	0.28	0.26	0.16	0.3	38.8	35	11.5	8.3	0.64	57.9	-0.1	0.01	17	13.1	28.1	2.004	0.161	0.094	0.72	11.3	28.1	0	42
26.7	0.00115	1.057	1.4	0.5193	1.86	25	10	1	0.063	0.93		11		5.2	0.12	10.1	3.7	1.14	0.29	0.26	0.16	0.35	37.5	33.1	11.7	8.3	0.59	55.8	-0.11	0.01	17.9	13	25.8	1.937	0.161	0.094	0.71	11.2	25.7	0	42
26.7	0.00115	1.057	1.4	0.5193	1.95	23	12	1	0.063	0.93		10.9		5.5	0.1	10	3.9	1.22	0.3	0.27	0.16	0.41	36.3	31.1	11.9	8.4	0.53	53.9	-0.12	0.01	18.7	12.9	23.5	1.877	0.161	0.094	0.71	11	23.5	0	42
26.7	0.00115	1.057	1.4	0.5193	2.04	21	13	1	0.063	0.93		10.7		5.7	0.08	9.8	4	1.3	0.31	0.27	0.16	0.48	35.1	29.1	12.1	8.5	0.47	52.1	-0.12	0.01	19.5	12.7	21.3	1.823	0.161	0.094	0.7	10.8	21.3	0	41
26.7	0.00115	1.057	1.4	0.5193	2.14	19	14	1	0.063	0.93		10.6		5.9	0.07	9.7	4.1	1.38	0.32	0.27	0.16	0.52	34.1	27.1	12.3	8.6	0.43	50.3	-0.13	0.01	20.2	12.6	19.2	1.773	0.161	0.094	0.7	10.7	19.2	0	41

Taevapin	D	L	Vdotcond	Vdotevap	charge	Tamb	dTsub	dTsup	mdotr	beta	Tcrit	Trevapin	Tsatoutc	COPsys	xevapin	Tsatsuc	COPsens	mcond	mevap	mliqline	mcomp	fsubc	Tsatdis	Taoutc	Qevap	Qsens	f2phc	Tdis	xcondout	fsupevap	EER	Taoutevap	Troutcond	Wdotcomp	Wdotcond	Wdotevap	SHR	Tsuc	Thotout	eps_ihx	rhosuc
CT MIGRATE 1PH IHX W/ RECEIVER AND ORIFICE TUBE																																								
26.7	0.00143	0.015	1.4	0.5176	1.6	35	5	1	0.056	0.82				4.6	0.1	10.8	3.4	0.89	0.3	0.25	0.16	0.12	44.2	42.7	10.5	7.9	0.77	83.6	-0.06	0.01	15.5	13.8	38.1	2.006	0.161	0.094	0.75	33.1	24.1	0.82	37
26.7	0.00143	0.015	1.4	0.5176	1.7	33	7.1	1	0.056	0.82				4.8	0.09	10.7	3.5	0.97	0.31	0.25	0.16	0.2	42.8	40.7	10.7	7.9	0.7	79.2	-0.08	0.01	16.2	13.7	34.7	1.939	0.161	0.094	0.74	30.2	22.3	0.81	37
26.7	0.00143	0.015	1.4	0.5176	1.81	31	8.9	1	0.056	0.82				4.9	0.08	10.6	3.6	1.08	0.32	0.26	0.16	0.3	41.7	38.6	10.8	8	0.61	75.4	-0.1	0.01	16.9	13.6	31.8	1.885	0.161	0.094	0.74	27.8	20.8	0.81	38
26.7	0.00143	0.015	1.4	0.5176	1.95	29	10	1	0.056	0.82				5.1	0.07	10.6	3.7	1.2	0.33	0.26	0.16	0.41	40.7	36.6	10.9	8	0.52	72.1	-0.12	0.01	17.4	13.5	29.4	1.842	0.161	0.094	0.74	25.7	19.6	0.8	38
26.7	0.00143	0.015	1.4	0.5176	2.07	27	12	1	0.056	0.82				5.2	0.06	10.5	3.8	1.31	0.34	0.26	0.16	0.5	39.9	34.6	11	8.1	0.43	69.2	-0.13	0.01	17.8	13.4	27.2	1.805	0.161	0.094	0.73	23.9	18.5	0.8	39
26.7	0.00143	0.015	1.4	0.5176	2.19	25	13	1	0.056	0.82				5.4	0.05	10.4	3.9	1.42	0.34	0.27	0.16	0.54	39.2	32.6	11.1	8.1	0.39	66.5	-0.14	0.01	18.3	13.4	25.1	1.772	0.16	0.094	0.73	22.2	17.5	0.79	39
26.7	0.00143	0.015	1.4	0.5176	2.3	23	15	1	0.056	0.82				5.5	0.05	10.4	4	1.52	0.35	0.27	0.16	0.58	38.4	30.5	11.1	8.1	0.36	63.9	-0.15	0.01	18.7	13.3	23.1	1.74	0.16	0.094	0.73	20.5	16.5	0.79	39
26.7	0.00143	0.015	1.4	0.5176	2.41	21	16	1	0.056	0.82				5.6	0.04	10.3	4.1	1.61	0.36	0.27	0.16	0.62	37.7	28.5	11.2	8.1	0.33	61.3	-0.16	0.01	19.1	13.3	21	1.709	0.16	0.094	0.73	18.9	15.6	0.78	40
26.7	0.00143	0.015	1.4	0.5176	2.5	19	17	1	0.056	0.82				5.7	0.03	10.2	4.1	1.7	0.37	0.27	0.16	0.65	37.1	26.5	11.3	8.2	0.31	58.7	-0.18	0.01	19.5	13.2	19	1.68	0.16	0.094	0.72	17.3	14.6	0.78	40
CT MIGRATE 1PH IHX W/ RECEIVER AND CAPTUBE																																								
26.7	0.001	1.018	1.4	0.5176	1.6	35	5	1	0.056	0.81				4.6	0.1	10.8	3.4	0.89	0.3	0.25	0.16	0.12	44.2	42.6	10.5	7.9	0.77	83.5	-0.06	0.01	15.6	13.8	38.1	1.995	0.161	0.094	0.75	33.1	24	0.82	37
26.7	0.001	1.018	1.4	0.5176	1.74	33	7.7	1	0.056	0.81				4.8	0.08	10.7	3.5	1.01	0.32	0.25	0.16	0.24	43	40.6	10.7	7.9	0.67	79.1	-0.09	0.01	16.3	13.6	34.3	1.939	0.161	0.094	0.74	29.9	22.1	0.81	38
26.7	0.001	1.018	1.4	0.5176	1.89	31	9.7	1	0.056	0.81				4.9	0.07	10.6	3.6	1.14	0.32	0.26	0.16	0.36	42.1	38.6	10.8	8	0.55	75.8	-0.11	0.01	16.7	13.6	31.5	1.899	0.161	0.094	0.74	27.5	20.6	0.81	38
26.7	0.001	1.018	1.4	0.5176	2.04	29	11	1	0.056	0.81				5	0.06	10.6	3.7	1.29	0.33	0.26	0.16	0.48	41.5	36.6	10.9	8	0.44	73	-0.13	0.01	17.1	13.5	29.2	1.869	0.161	0.094	0.74	25.6	19.5	0.8	38
26.7	0.001	1.018	1.4	0.5176	2.18	27	13	1	0.056	0.81				5.1	0.06	10.5	3.8	1.41	0.34	0.26	0.16	0.54	40.9	34.6	10.9	8	0.39	70.5	-0.14	0.01	17.4	13.5	27.1	1.844	0.161	0.094	0.74	23.8	18.5	0.8	39
26.7	0.001	1.018	1.4	0.5176	2.3	25	15	1	0.055	0.81				5.2	0.05	10.5	3.8	1.53	0.34	0.27	0.16	0.59	40.4	32.6	11	8.1	0.35	68.1	-0.16	0.01	17.8	13.4	25.1	1.821	0.16	0.094	0.73	22.2	17.5	0.79	39
26.7	0.001	1.018	1.4	0.5176	2.41	23	16	1	0.055	0.81				5.3	0.04	10.4	3.9	1.63	0.35	0.27	0.16	0.63	39.9	30.5	11.1	8.1	0.32	65.8	-0.17	0.01	18.1	13.4	23	1.799	0.16	0.094	0.73	20.5	16.5	0.79	39
26.7	0.001	1.018	1.4	0.5176	2.51	21	18	1	0.055	0.81				5.4	0.04	10.3	3.9	1.72	0.36	0.27	0.16	0.66	39.5	28.5	11.2	8.1	0.29	63.5	-0.19	0.01	18.4	13.3	21	1.778	0.16	0.094	0.73	18.9	15.6	0.78	40
26.7	0.001	1.018	1.4	0.5176	2.6	19	19	1	0.055	0.81				5.5	0.03	10.3	4	1.8	0.37	0.27	0.16	0.69	39	26.5	11.2	8.2	0.27	61.3	-0.2	0.01	18.6	13.3	19	1.759	0.16	0.094	0.73	17.3	14.7	0.78	40
CT MIGRATE TO 2-PHASE IHX W/ RECEIVER AND ORIFICE TUBE																																								
26.7	0.00184	0.015	1.4	0.518	1.27	35	-0.1	1	0.063	0.92				4.1	0.2	10.8	3.1	0.61	0.24	0.25	0.16	0	43.7	42.8	10.5	7.9	0.88	88.7	0.14	0.01	14.1	13.8	41.7	2.238	0.161	0.094	0.75	39.8	36.4	0.94	35.5
26.7	0.00184	0.015	1.4	0.518	1.29	33	-0.1	1	0.063	0.92				4.4	0.18	10.6	3.3	0.62	0.25	0.25	0.16	0	41.9	40.8	10.7	8	0.88	84.9	0.12	0.01	15	13.6	40	2.145	0.161	0.094	0.74	38.1	34	0.94	35.6
26.7	0.00184	0.015	1.4	0.518	1.31	31	-0.1	1	0.063	0.92				4.7	0.16	10.4	3.4	0.63	0.26	0.26	0.16	0	40.2	38.9	11	8.1	0.89	81	0.1	0.01	15.9	13.4	38.2	2.056	0.161	0.094	0.73	36.3	31.6	0.93	35.8
26.7	0.00184	0.015	1.4	0.518	1.33	29	-0.1	1	0.063	0.92				5	0.14	10.2	3.6	0.63	0.27	0.26	0.16	0	38.5	36.9	11.2	8.2	0.9	77.1	0.08	0.01	16.9	13.3	36.4	1.969	0.161	0.094	0.73	34.4	29.1	0.93	35.9
26.7	0.00184	0.015	1.4	0.518	1.36	27	-0.1	1	0.062	0.92				5.3	0.12	10	3.8	0.65	0.28	0.26	0.16	0	36.8	34.9	11.5	8.2	0.9	73.2	0.06	0.01	17.9	13.1	34.6	1.885	0.161	0.094	0.72	32.6	26.6	0.92	36
26.7	0.00184	0.015	1.4	0.518	1.39	25	-0.1	1	0.062	0.92				5.6	0.1	9.8	4	0.66	0.3	0.27	0.16	0	35	33	11.7	8.3	0.91	69.2	0.04	0.01	19	13	32.8	1.803	0.161	0.094	0.71	30.6	24	0.9	36.2
26.7	0.00184	0.015	1.4	0.518	1.42	23	-0.1	1	0.062	0.92				5.9	0.08	9.6	4.2	0.68	0.31	0.27	0.16	0	33.3	31	11.9	8.4	0.91	65	0.02	0.01	20.2	12.8	31.1	1.723	0.161	0.094	0.71	28.3	21.4	0.87	36.4
26.7	0.00184	0.015	1.4	0.518	1.48	21	0.4	1	0.062	0.92				6.3	0.07	9.4	4.4	0.72	0.33	0.27	0.16	0.01	31.6	29	12.2	8.5	0.92	59.9	0	0.01	21.4	12.6	28.9	1.645	0.161	0.094	0.7	25	18.8	0.8	36.9
26.7	0.00184	0.015	1.4	0.518	1.56	19	3.6	1	0.062	0.92				6.6	0.05	9.3	4.6	0.78	0.34	0.28	0.16	0.07	30	27.1	12.4	8.6	0.86	54.4	-0.03	0.01	22.7	12.5	24.1	1.572	0.161	0.094	0.7	21.1	16.5	0.79	37.6
CAPTUBE BASE W/ VARIABLE SPEED CONDENSER FAN																																								
26.7	0.00102	0.6	1.366	0.5222	1.54	35	5	5	0.062	0.91	16.1	11.8		4.1	0.22	10.9	3.1	0.9	0.23	0.25	0.16	0.12	45	43	10.5	7.9	0.79	70.9	-0.06	0.06	13.9	13.9	38.8	2.275	0.152	0.095	0.75	16	38.7	0	41
26.7	0.00102	0.6	1.14	0.5222	1.54	33	5.3	5	0.062	0.91	15.8	11.8		4.3	0.21	10.8	3.2	0.9	0.23	0.25	0.16	0.12	44.4	42.5	10.6	7.9	0.79	69.7	-0.07	0.06	14.5	13.8	37.8	2.242	0.099	0.095	0.75	15.9	37.7	0	41
26.7	0.00102	0.6	0.972	0.5222	1.54	31	5.7	5	0.062	0.91	15.5	11.7		4.4	0.2	10.7	3.3	0.9	0.23	0.25	0.16	0.12	43.9	42.2	10.7	7.9	0.79	68.7	-0.07	0.06	15	13.8	36.8	2.213	0.069	0.095	0.74	15.8	36.8	0	41
26.7	0.00102	0.6	0.846	0.5222	1.54	29	6	5	0.062	0.91	15.2	11.6		4.5	0.2	10.7	3.3	0.89	0.23	0.25	0.16	0.12	43.4	41.8	10.7	8	0.79	67.6	-0.07	0.06	15.4	13.7	36	2.186	0.05	0.095	0.74	15.8	36	0	41
26.7	0.00102	0.6	0.745	0.5222	1.54	27	6.3	5	0.062	0.91	15	11.6		4.6	0.19	10.6	3.4	0.89	0.24	0.25	0.16	0.12	42.9	41.4	10.8	8	0.79	66.7	-0.07	0.06	15.8	13.7	35.3	2.163	0.038	0.095	0.74	15.7	35.2	0	41
26.7	0.00102	0.6	0.662	0.5222	1.54	25	6.5	5	0.062	0.91	14.8	11.5		4.7	0.18	10.6	3.5	0.88	0.24	0.25	0.16	0.12	42.6	41.1	10.9	8	0.79	65.8	-0.08	0.06	16	13.6	34.6	2.143	0.029	0.095	0.74	15.7	34.6	0	41
26.7	0.00102	0.6	0.594	0.5222	1.54	23	6.7	5	0.062	0.91	14.6	11.5		4.8	0.18	10.5	3.5	0.88	0.24	0.25	0.16	0.12	42.2	40.9	10.9	8	0.79	65.1	-0.08	0.06	16.3	13.6	34.1	2.127	0.023	0.095	0.74	15.6	34.1	0	41
26.7	0.00102	0.6	0.536	0.5222	1.54	21	6.9	5	0.062	0.91	14.5	11.4		4.8	0.18	10.5	3.5	0.88	0.24	0.25	0.16	0.12	42	40.7	11	8.1	0.79	64.3	-0.08	0.06	16.5	13.6	33.6	2.113	0.019	0.095	0.74	15.6	33.6	0	41
26.7	0.00102	0.6	0.488	0.5222	1.54	19	7.1	5	0.062	0.91	14.4	11.4		4.9	0.17	10.5	3.6	0.88	0.24	0.25	0.16	0.12	41.7	40.5	11	8.1	0.79	63.7	-0.08	0.06	16.6	13.5	33.2	2.101	0.015	0.095	0.73	15.5	33.2	0	41
CAPTUBE BASE W/ VARIABLE SPEED BLOWER																																								
26.7	0.00102	0.6	1.4	0.5136	1.54	35	4.9	5	0.062	0.91	15.9	11.8		4.1	0.22	10.8	3	0.91	0.22	0.25	0.16	0.12	44.8	42.8	10.5	7.8	0.79	70.7	-0.06	0.06	13.9	13.8	38.6	2.266	0.161	0.091	0.75	15.9	38.6	0	41
26.7	0.00102	0.6	1.4	0.4545	1.54	33	5.2	5	0.061	0.91	14.6	10.9		4.3	0.2	10	3.1	0.9	0.23	0.25	0.16	0.13	42.9	40.7	10.5	7.4	0.79	68.1	-0.06	0.06	14.7	12.7	36.4	2.163	0.161	0.067	0.71	15	36.3	0	40
26.7	0.00102	0.6	1.4	0.405	1.54	31	5.4	5	0.059	0.91	13.2	10		4.5	0.19	9.1	3.1	0.9	0.23	0.25	0.16	0.13	40.8	38.5	10.4	7.1	0.79	65.4	-0.06	0.06	15.5	11.7	34.1	2.063	0.161	0.05	0.68	14.2	34	0	39
26.7	0.00102	0.6	1.4	0.3599	1.54	29	5.6	5	0.058	0.91	11.8	9.12		4.8	0.18	8.2	3.1	0.9	0.23	0.26	0.15	0.14	38.8	36.4	10.4	6.9	0.79	62.8	-0.06	0.05	16.2	10.4	31.8	1.965	0.16	0.037	0.66	13.3	31.8	0	38
26.7	0.00102	0.6	1.4	0.3272	1.54	27	5.8	5	0.056	0.91	10.4	8.21		5	0.16	7.3	3.2	0.89	0.24	0.26	0.15	0.14	36.8	34.2	10.3	6.7	0.78	60.1	-0.06	0.05	16.9	9.3	29.6	1.871	0.16	0.029	0.65	12.3	29.5	0	37
26.7	0.00102	0.6	1.4	0.2988	1.54	25	6	5	0.055	0.91	8.92	7.26		5.2	0.15	6.3	3.3	0.89	0.24	0.26	0.15	0.15	34.7	32.1	10.2	6.5	0.78	57.4	-0.06	0.05	17.6	8.2	27.3	1.779	0.16	0.023	0.63	11.4	27.3	0	36
26.7	0.00102	0.6	1.4	0.2735	1.54	23	6.1	5	0.053	0.91	7.43	6.25		5.4	0.14	5.3	3.4	0.89	0.24	0.27	0.15	0.15	32.6	29.9	10.1	6.3	0.78	54.7	-0.06	0.04	18.4	7	25.1	1.687	0.16	0.019	0.62	10.4	25.1	0	35
26.7	0.00102	0.6	1.4	0.2515	1.54	21	6.2	5	0.052	0.91	5.92	5.21		5.6	0.13	4.3	3.5	0.89	0.24	0.27	0.14	0.16	30.5	27.7	10	6.2	0.78	51.9	-0.06	0.04	19.1	5.8	22.8	1.597	0.16	0.015	0.62	9.3	22.8	0	34
26.7	0.00102	0.6	1.4	0.2322	1.54	19	6.3	5	0.05	0.91	4.4	4.15		5.8	0.12	3.2	3.6	0.89	0.24	0.27	0.14	0.16	28.4	25.5	9.9	6	0.78	49.1	-0.06	0.04	19.9	4.5	20.6	1.509	0.16	0.013	0.61	8.3	20.6	0	33
TXV BASELINE W/ BRISTOL RECIP																																								
26.7			1.4	0.5222	1.54	35	5	5	0.063	0.97				4.1	0.22	10.9	3.1	0.91	0.22	0.25	0.16	0.12	45	42.9	10.5	7.9	0.79	70.5	-0.06	0.06	14	13.9	38.7	2.264	0.161	0.095	0.75	16	38.6	0	41
26.7			1.4	0.5222	1.54	33	5.1	5	0.063	0.97				4.3	0.21	10.6	3.2	0.9	0.23	0.25	0.16	0.12	43.2	40.9	10.8	8	0.79	67.8	-0.06	0.06	14.8	13.7	36.7	2.19	0.161	0.095	0.74	15.7	36.7	0	41
26.7			1.4	0.5222	1.54	31	5.1	5	0.064	0.97				4.6	0.19	10.4	3.3	0.89	0.23	0.25	0.16	0.12	41.4	39	11.1	8.1	0.8	65.1	-0.06	0.06	15.6	13.5	34.8	2.114	0.161	0.095	0.73	15.5	34.7	0	41
26.7			1.4	0.5222	1.54	29	5.1	5	0.064	0.97				4.9	0.18	10.2	3.5	0.88	0.24	0.26	0.16	0.12	39.6	37.1	11.3	8.2	0.81	62.4	-0.06	0.06	16.5	13.3	32.8	2.037	0.161	0.095	0.72	15.3	32.8	0	41
26.7			1.4	0.5222	1.54	27	5.1	5	0.064	0.97				5.1	0.16	10	3.7	0.87	0.25	0.26	0.16	0.11	37.8	35.1	11.6	8.3	0.81	59.7	-0.05	0.06	17.5	13.1	30.9	1.959	0.161	0.095	0.72	15	30.9	0	40
26.7			1.4	0.5222	1.54	25	5.1	5	0.064	0.97				5.4	0.15	9.8	3.9	0.86	0.26	0.26	0.16	0.11	36	33.1	11.8	8.4	0.82	57	-0.05	0.06	18.6	12.9	29	1.88	0.161	0.095	0.71	14.8	29	0	40
26.7			1.4	0.5222	1.54	23	5.1	5	0.065	0.97				5.8	0.13	9.6	4.1	0.85	0.26	0.26	0.16	0.11	34.2	31.2	12.1	8.5	0.82	54.3	-0.05	0.06	19.7	12.8	27.1	1.799	0.161	0.095	0.71	14.6	27.1	0	40
26.7			1.4	0.5222	1.54	21	5	5	0.065	0.97				6.1	0.12	9.3	4.3	0.84	0.27	0.27	0.16	0.1	32.4	29.2	12.4	8.6	0.83	51.5	-0.05	0.05	21	12.6	25.2	1.717	0.161	0.095	0.7	14.4	25.2	0	40
26.7			1.4	0.5222	1.54	19	4.9	5	0.065	0.97				6.5	0.1	9.1	4.5	0.83	0.28	0.27	0.16	0.1	30.6	27.3	12.6	8.7	0.84	48.8	-0.05	0.05	22.3	12.4	23.3	1.634	0.161	0.095	0.69	14.2	23.3	0	39
CAPTUBE BASE W/ BRISTOL RECIP																																								
26.7	0.00102	0.6	1.4	0.5222	1.54	35	5	5	0.062	0.96				4.1	0.22	10.9	3.1	0.91	0.23	0.25	0.16	0.12	44.9	42.8	10.5	7.9	0.79	70.4	-0.06	0.06	14.1	13.9	38.6	2.237	0.161	0.095	0.75	16	38.5	0	41
26.7	0.00102	0.6	1.4	0.5222	1.54	33	5.7	8.8	0.062	0.96				4.4	0.2	10.2	3.2	0.91	0.22	0.25	0.16	0.14	43.1	40.9	10.9	8	0.77	71.1	-0.07	0.12	15.1	13.7	36.1	2.159	0.161	0.095	0.73	19	36.1	0	39
26.7	0.00102	0.6	1.4	0.5222	1.54	31	6.2	12	0.061	0.96				4.7	0.18	9.4	3.3	0.92	0.21	0.25	0.16	0.16	41.3	39	11.2	8	0.76	71	-0.07	0.18	16	13.7	33.8	2.081	0.161	0.095	0.71	21.2	33.7	0	38
26.7	0.00102	0.6	1.4	0.5222	1.54	29	6.7	14	0.06	0.96				4.9	0.17	8.6	3.4	0.93	0.2	0.26	0.15	0.17	39.4	37	11.4	8	0.74	70.5	-0.07	0.23	16.8	13.7	31.4	2.004	0.161	0.095	0.7	22.8	31.4	0	36
26.7	0.00102	0.6	1.4	0.5222	1.54	27	7.1	16	0.058	0.96				5.2	0.16	7.8	3.5	0.94	0.2	0.26	0.15	0.19	37.5	34.9	11.5	7.9	0.73	69.7	-0.08	0.28	17.6	13.9	29	1.926	0.161	0.095	0.69	23.9	29	0	35
26.7	0.00102	0.6	1.4	0.5222	1.54	25	7.5	18	0.057	0.96				5.4	0.14	6.9	3.6	0.94	0.19	0.26	0.15	0.2	35.6	32.9	11.5	7.8	0.72	68.6	-0.08	0.33	18.3	14	26.8	1.85	0.161	0.095	0.68	24.6	26.8	0	34
26.7	0.00102	0.6	1.4	0.5222	1.54	23	7.8	19	0.056	0.96				5.6	0.13	6	3.7	0.95	0.18	0.27	0.14	0.21	33.7	30.8	11.6	7.7	0.71	67.4	-0.08	0.37	19	14.2	24.5	1.775	0.161	0.095	0.67	25.2	24.5	0	33
26.7	0.00102	0.6	1.4	0.5222	1.54	21	8	21	0.055	0.96				5.8	0.12	5	3.8	0.96	0.17	0.27	0.14	0.23	31.7	28.6	11.5	7.6	0.7	66	-0.08	0.42	19.7	14.3	22.2	1.701	0.16	0.096	0.66	25.6	22.2	0	31
26.7	0.00102	0.6	1.4	0.5222	1.54	19	8.2	22	0.053	0.96				6	0.11	3.9	3.9	0.97	0.17	0.27	0.14	0.24	29.7	26.5	11.5	7.5	0.69	64.6	-0.08	0.45	20.3	14.5	20	1.63	0.16	0.096	0.65	25.9	20	0	30