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Improvements in Refrigerant Flow Distribution Using an Expansion Valve with Integrated Distributor

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ABSTRACT

Many factors, such as loading, flow circuiting, phase separation, and distributor effectiveness determine how uniformly refrigerant is distributed within a multicircuited heat exchanger. This work describes the development of a test procedure for evaluation of refrigerant flow distribution in a multicircuited round tube evaporator as well as a comparison between a typical valve/distributor combination and a novel expansion device that integrates the distributor into the valve body. The results of this study show that refrigerant flow distribution can be made more uniform through implementation of this device.

1. INTRODUCTION

The importance of proper refrigerant flow distribution through a multi-circuited evaporator is well known. Choi et al., 2003, showed that as much as 30% capacity degradation can come from refrigerant flow maldistribution, even at the same superheat setting. Kaern et al., 2013, indicated that better matched individual circuit outlet superheat in an interlaced evaporator can lead to as much as 7% improvement in overall UA and 2.4% improvement in system COP. Fay and Hrnjak, 2011, indicated that a 4% and 5% gain in COP and capacity, respectively, is possible when moving from a refrigerant flow distribution where some circuits have two-phase exit, to a matched superheat condition. Achieving the desired refrigerant flow distribution using typical refrigerant distributors after the expansion device can be challenging. Li *et al.* (2002) demonstrated that the shape of the distributor base, the distributor orientation, and the location of the orifice with respect to the distributor can significantly alter the refrigerant flow distribution through the parallel circuits. Chen (1993) found that only when a stratified flow regime is avoided at the inlet of the distributor, will distributor orientation play less of a role in refrigerant flow distribution than distributor geometric non-uniformity. This points to an idea that many in the industry already try implement in the coupling of the expansion device and the distributor, namely, avoid any flow development or separation between the fluid expansion and the distributor. Some designers even implement a secondary orifice with the aim of rehomogenizing the flow prior to the distributor. Bowers (2009) attempted to quantify this distance required to establish separation of the two phases directly after an expansion device and showed that this varies greatly with the mass flux, tube diameter, and quality exiting the expansion device. This large variation in separation characteristics can make it difficult to ensure that there is no separation occurring prior to the distributor.

The novel design of expansion valve and distributor examined in this work seeks to limit the likelihood of phase separation between the expansion device and distributor, and thus ensure uniform refrigerant flow distribution, by

integrating the distributor into an electronic expansion valve directly at the outlet of the orifice. An exploded view of this device is shown in Figure 1. This is noticeably different from conventional valves where the distributor is brazed some length downstream of the orifice. In addition to combining the distributor and expansion device a check valve for use in heat pump mode is integrated into the body of the device.
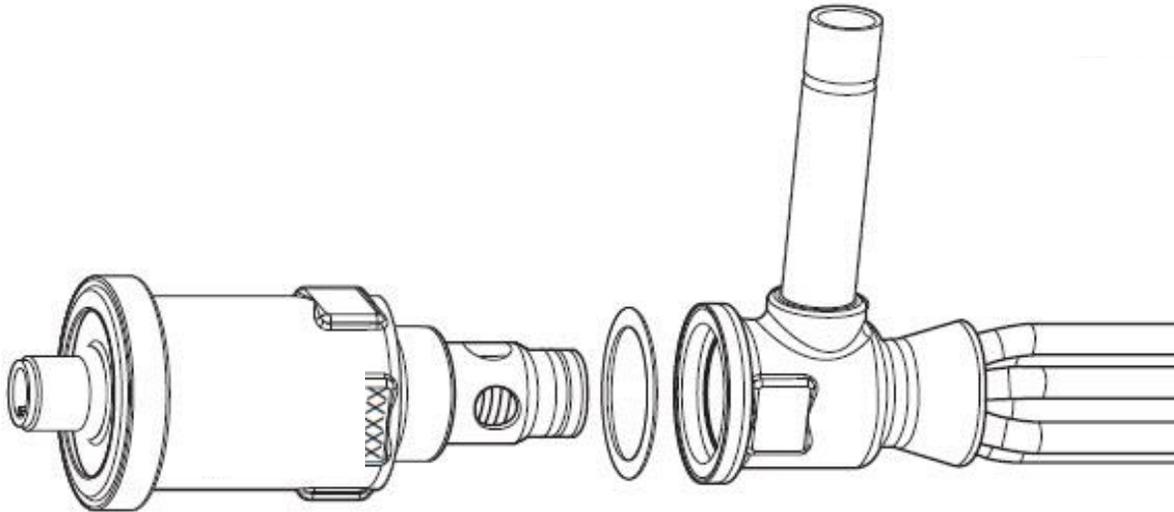
Figure 1: Integrated Expansion Valve and Distributor

The work herein presented aimed to show the effect of using such a device on refrigerant flow distribution, in comparison to using a more traditional combination of separated valve and distributor. This was done by applying both technologies to a 3 ton round-tube-plate-fin evaporator at various conditions and measuring the total refrigerant flow through each circuit. The baseline valve was a typical thermostatic expansion valve modified in such a way that the control bulb was replaced with a mechanical control that allowed the user to set the orifice opening. The outlet of the baseline valve had a fitting for connecting to the distributor. The lines exiting this distributor were brazed into the inlets of the evaporator. Both the baseline valve/distributor combination and the integrated valve distributor connected to the test evaporator are shown in Figure 2. It can be seen from Figure 2 that a secondary benefit of using this integrated design is that there is no need for have a joint between two dissimilar metals, a known potential for corrosion.

Figure 2: Modified Thermostatic Expansion Valve With External Distributor And Integrated Valve and Distributor

2. EXPERIMENTAL FACILITIES

The measurements for this work were be performed in a facility designed to focus on the performance of an integrated electronic expansion valve/distributor. This facility is shown schematically in Figure 3 while a representative cycle is plotted on an R-410A pressure-specific enthalpy diagram is shown in Figure 4. A 3 Ton R-410A round-tube-plate-fin evaporator with six circuits was situated in the wind tunnel to obtain an air side capacity. The blower is not shown in the schematic for simplicity's sake, but it is located downstream of the nozzles. The wind tunnel was placed in an environmental chamber that provides the desired ambient air conditions (temperature, hunidity, etc…). Uneven loading on the evaporator could be achieved by manipulation of the air flow over parts of the coil or by ball valves on the individual circuits after the condenser. Superheat in each circuit was monitored using individual pressure and temperature measurements. In the case that two-phase flow existed at the exit of the evaporator, outlet calorimeters were used to determine the exit quality of each circuit. After passing through the evaporator, each circuit passes through individual glycol cooled condensers and subcooler at which point the subcooling was determined with pressure and temperature measurements. Each circuit then passed through individual mass flow meters, yielding the total mass flow rate in each circuit. After passing through the mass flow meters, the circuits will be joined in a header. It should be noted that great care was taken in the construction of the facility to keep line lengths and pressure drop in each of the circuits the same so as not to influence distribution. The liquid was pumped using a pump capable of the required flow rates and pressure ratios seen in a typical R-410a vapor compression system; so as to best mimic the inlet conditions to the integrated valve/distributor. An inlet calorimeter was used after the pump to reduce the subcooling to the appropriate condition at the valve/distributor inlet. All testing was performed with pure R410A.
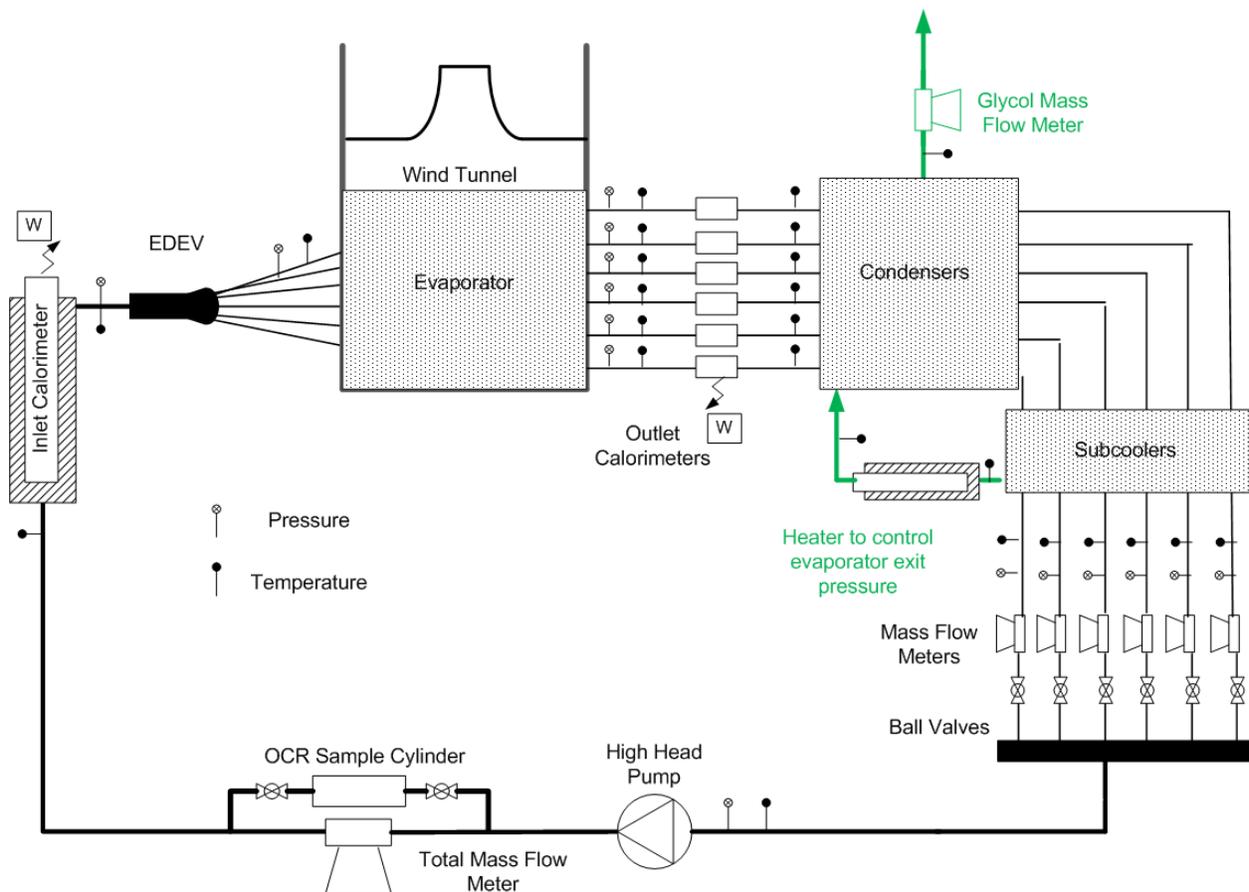
Figure 3: Experimental Facility Schematics
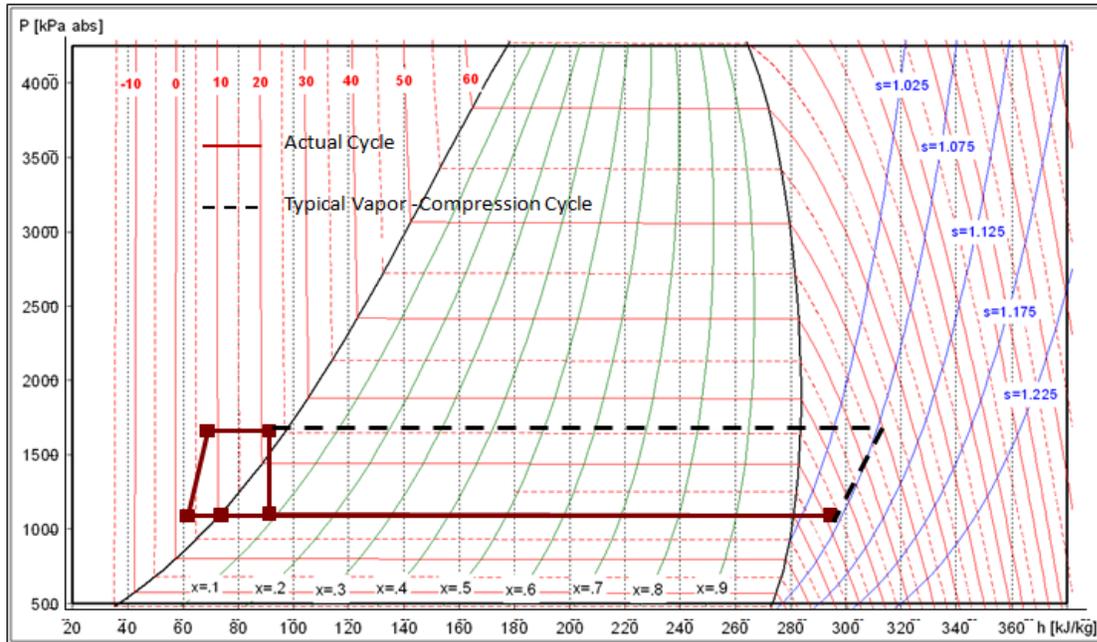
Figure 4: Representative Pressure-Specific Enthalpy of Test Cycle

In order to ensure that the test facility was not influencing the results of the testing, pressure drop measurements in the additional segments of each circuit was measured using single phase nitrogen at equal mass flow rates. Specifically, these measurements were made between the evaporator exit of each circuit and the ball valves at the exit of the mass flow meters. The results of these pressure drop measurements for several flow rates are shown in Figure 5. The similarities in pressure drop at each mass flow indicated that flow resistance in the additional parts of the facility are nearly the same for each circuit and likely have little influence on the distribution of the two-phase refrigerant flow through the evaporator.
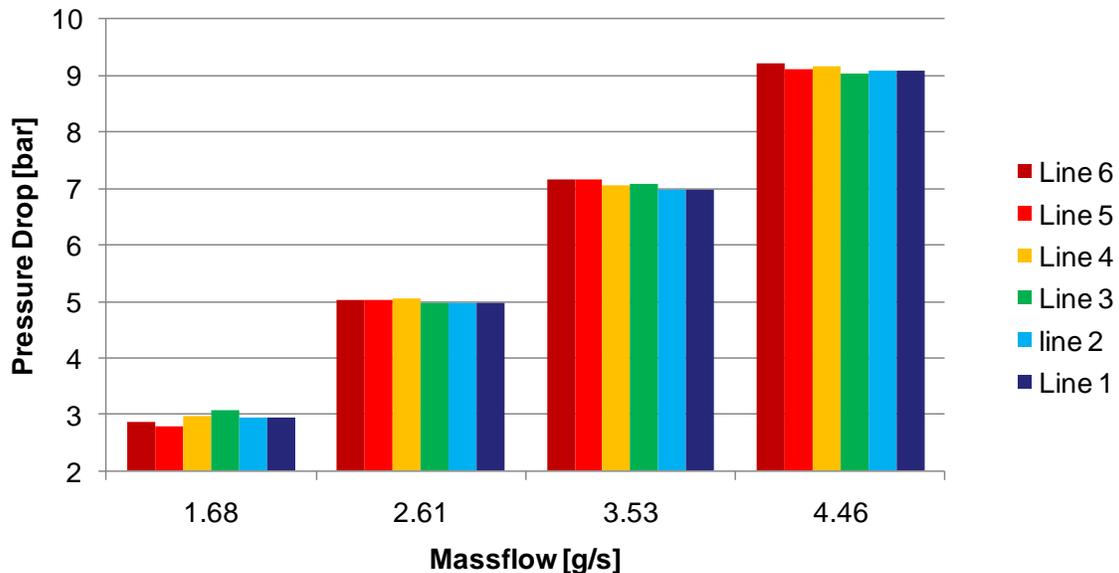
Figure 5: Individual Circuit Pressure Drops using Single Phase Nitrogen at Various Mass Flow Rates

The refrigerant mass flow and superheat distribution was quantified with both the baseline valve and distributor combination as well as the new integrated valve and distributor at four separate conditions, shown in Table 1. These test conditions were based upon AHRI 210/240 ambient test conditions and typical air flow and refrigerant characteristics in an air conditioning system operating at these conditions. Humidity was removed from the test conditions in attempt to decouple the refrigerant flow distribution and latent loading of the coil. To achieve this, the wet bulb temperature of the air was kept below the suction saturation temperature of the refrigerant.

Table 1: Test Conditions

Test Condition	Dry Bulb (°C)	Wet Bulb (°C)	Air Flow Rate (SCFM)	Suction Pressure (kPa)	Liquid Pressure (kPa)	Suction Temperature (°C)	Liquid Temperature (°C)	Average Exit Superheat (K)
A (Steady Wet)	26.7	<9.9	1200	1080	2544	9.9	38.3	6
B (Steady Wet)	26.7	<9.9	1200	1080	2137	9.9	31.1	5
C (steady Dry)	26.7	<3.6	1050	894	2137	3.6	31.1	6
F (Steady Wet)	26.7	<10.7	650	1108	1558	10.7	21.7	9

3. RESULTS AND DISCUSSION

As discussed earlier, the refrigerant mass flow rate and exit superheat were measure independently for each circuit. An example of these results for the baseline valve and distributor at assembly Condition B are shown in Figure 6. As a reference the A-coil being test was oriented with the coil horizontally, circuits 1-3 were in the bottom slab and 4-6 in the top slab of the coil. In this example, the first three circuits are preferentially fed with refrigerant compared to the circuits in the top slab of the A-coil. In fact all circuits in the upper slab receive less refrigerant flow than any circuit in the bottom slab of the coil. The distribution of the superheat between the circuits in the inverse of the distribution of the refrigerant mass flow. This seems to be a logical result, as the circuits with less refrigerant flow should have a correspondingly higher exit superheat, if the loading on the air side is the same.
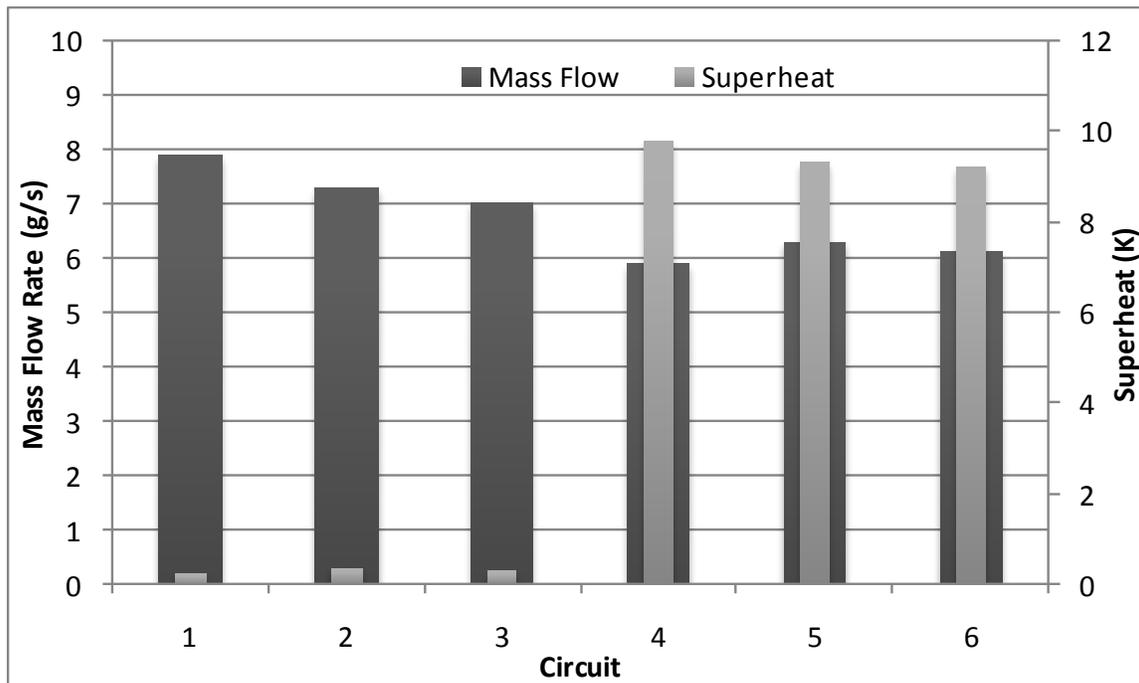
Figure 6: Individual Circuit Mass Flow and Superheat Distributions With Baseline Valve and Distributor at Condition B

Results at the same condition, Condition B, with the integrated valve and distributed are presented in the same fashion in Figure 7. In these results the refrigerant flow distribution appears to be less preferential to the circuit on the bottom slab of the A-coil. In fact, these results do not appear to indicate any preference between slabs as there are circuits in each slab that are both lower and higher in mass flow than the average. The superheat still appears to exhibit a maldistribution between the slabs. Circuits 4 and 5 actually have both the highest refrigerant mass flow rate and the highest exit superheat. Upon further examination it was noticed that the exit lines of each circuit header and the common suction header (which was left in place for this testing) presented a significant air blockage on the bottom slab of the heat exchanger. This was further confirmed through measurements of the air velocity distribution at the exit of the heat exchanger. This serves as demonstration that while some technologies may make the fluid distribution on one side of the heat exchanger optimum, it is important to ensure that the performance of both streams has been optimized with consideration of the other.
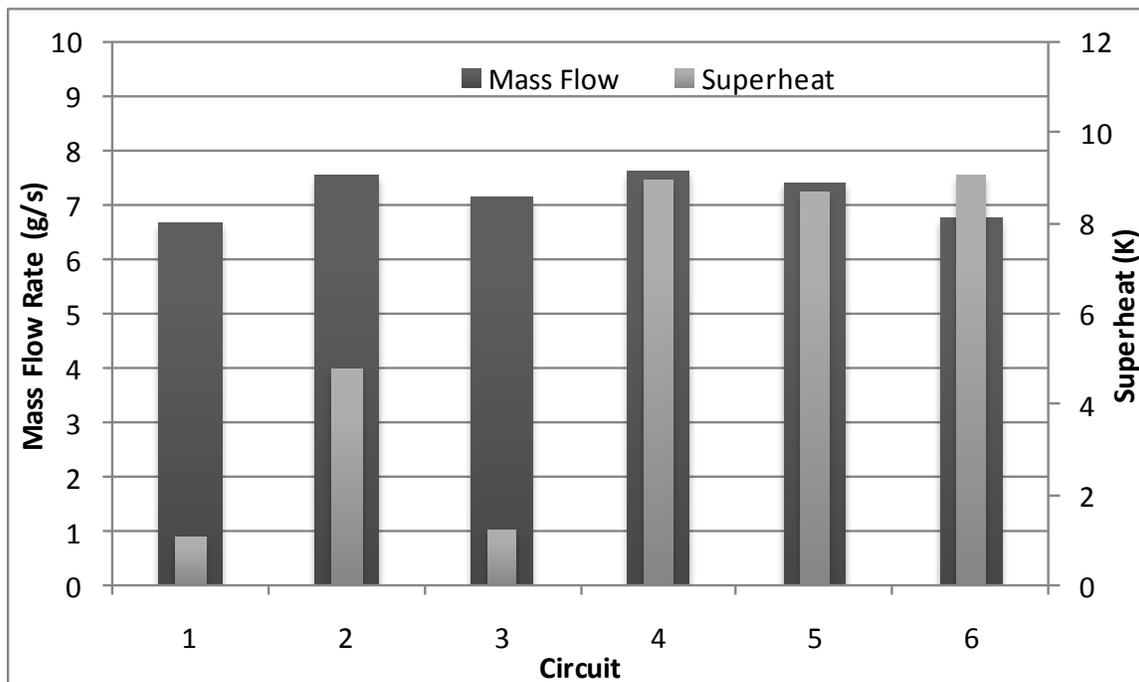
Figure 7: Individual Circuit Mass Flow and Superheat Distributions With Integrated Valve and Distributor at Condition B

Figure 8 presents a relative comparison of both the baseline valve and distributor assembly and the integrated valve and distributor (labeled EDEV). These results are presented in the form of percentage point deviation from the average in each circuit. As an example, for this six circuit heat exchanger, if a single circuit were to receive no refrigerant flow it would be reported as -16.67%. These results show that the baseline configuration experiences deviations of up to three percentage points, while the largest deviation seen in the case where the integrated valve and distributor are used is just over one percentage point. In all circuits the absolute deviation from the average is less using the integrated expansion valve and distributor than with the baseline configuration. It should be noted that in all but two of the circuits, the direction of the deviation (over fed or underfed) is changed when the integrated valve and distributor were used. The circuits in which the direction of the deviation where not changed were the circuits that received refrigerant flow closest to the average.
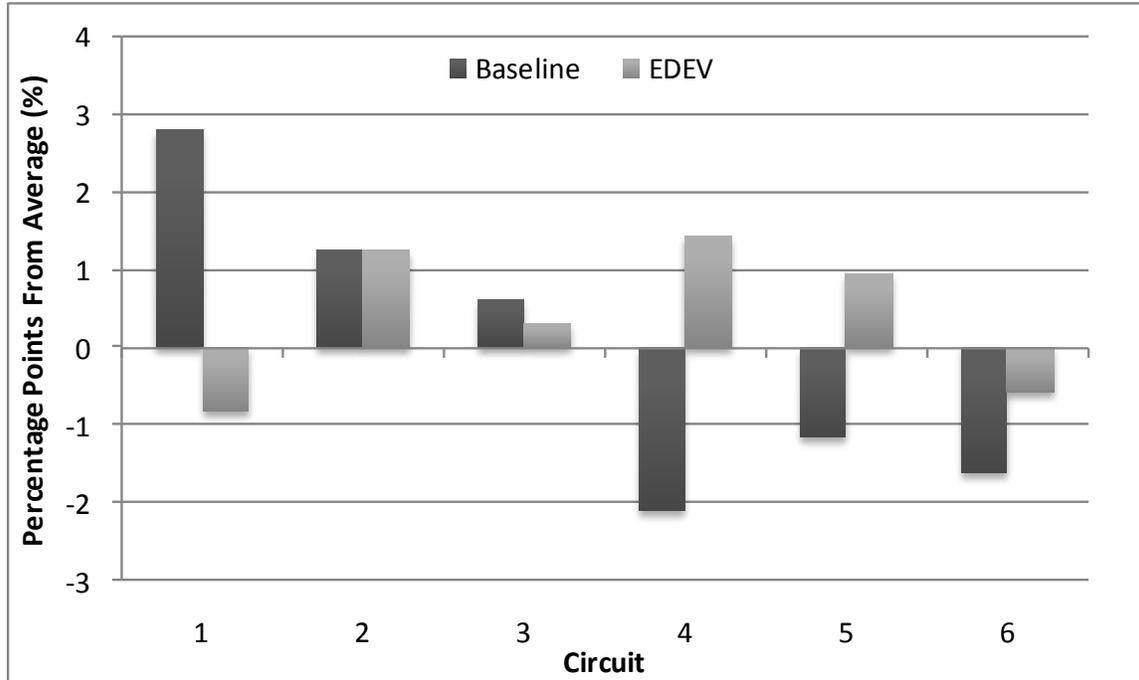
Figure 8: Deviation from Uniform Mass Flow Distribution in Both Configurations at Condition B

In order to comparatively evaluate the distribution results at the various conditions evaluated, they are presented in terms of a coefficient of variation. Bowers *et al.* (2006) used this approach in quantifying the refrigerant distribution in the manifolds of microchannel evaporators. The coefficient of variation (CV) is a measure of relative scatter with respect to the mean. Here the mean is considered to be exactly uniform refrigerant mass flow distribution among all six circuits. Thus, for present concerns the CV measures how far the liquid flow distribution deviates from completely uniform distribution. The use of the coefficient of variation allows the flow rates in each circuit to be reduced to one point that tells how uniformly the flow is distributed. This allows for a more compact analysis of the effect of varying test parameters. Equation (1) shows the equation used to obtain the CV. Where the standard deviation is gained from Equation (2) and the mean mass flow rate is defined as shown in Equation (3). The CV is bounded by 0 and the square root of the number of circuits, 2.449 for the six circuit evaporator evaluated here. The boundary value of 0 represents completely uniform distribution while the boundary value of $\sqrt{6}$ represents the entire refrigerant mass flow rate entering only one circuit.

$$CV = \frac{\sigma}{\bar{\dot{m}}} \tag{1}$$

$$\sigma = \sqrt{\sum_{i=1}^{n} \frac{(\dot{m}_i - \bar{\dot{m}})}{n}} \tag{2}$$

$$\bar{\dot{m}} = \frac{\sum_{i=1}^{n} \dot{m}}{n} \tag{3}$$

The results of the distribution, in the form of the coefficient of variation described above, in all the conditions evaluated are presented in Figure 9 for both the baseline valve and distributor configuration as well as the integrated valve and distributor. In general, as the tests progressed from A to F, the refrigerant flow distribution becomes less uniform, as evidenced by the higher CV value. In general this CV trend was not as strong for the integrated valve and distributor. These results show that the integrated valve and distributor consistently provided more uniform refrigerant flow distribution when comparing individual circuit deviation to the mean. It should be noted that this is the case over a large range of mass flow rates. The largest mass flow rate observed during this testing was

approximately three times the lowest.. The results indicate that significant improvement in refrigerant flow distribution in multi-circuited round-tube-plate-fin heat exchangers can be achieved through implementation of the an integrated expansion valve and distributor.
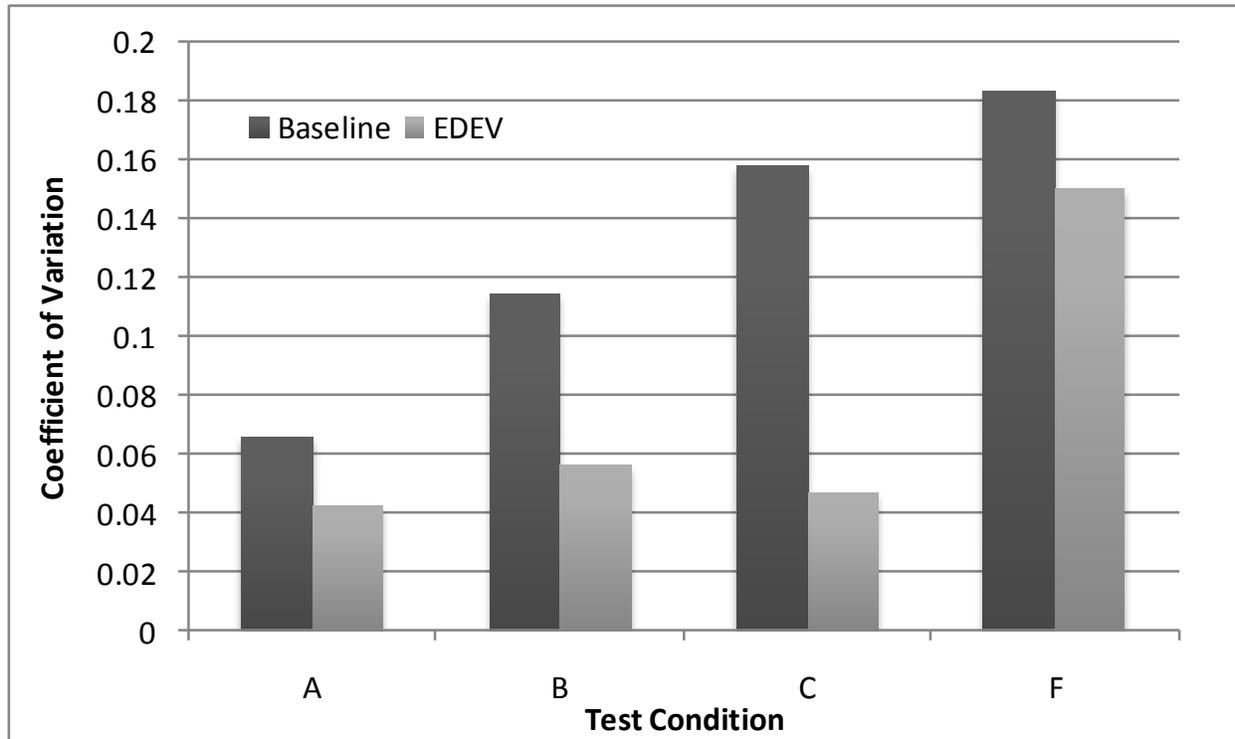
Figure 9: Quantification of Distribution for Baseline Valve and Integrated Valve and Distributor

4. CONCLUSIONS

Managing the refrigerant flow distribution in a multi-circuited evaporator in a way that optimizes the air side loading of the coil is crucial to obtaining the best performance. The work presented here demonstrated the possibility for improvements in the refrigerant flow distribution in such heat exchangers using a novel electronic expansion device. In addition to a more uniform refrigerant flow distribution this technology has the potential to increase the compactness of systems and provide a smaller bill of materials. A secondary finding of the above work also indicates that it is important to continually improve the interaction between both fluid streams of the heat exchanger. While not investigated in this work the technology examined also contains a built in check valve that would make the technology attractive for application in reversible systems.

NOMENCLATURE

CV	Coefficient of Variation	(-)
$\dot{m}$	Mass Flow Rate	(g/s
$\bar{\dot{m}}$	Average Mass Flow Rate	(g/s)
n	Number of Circuits	(-)
σ	Standard Deviation	(-)

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