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Effect of reduced evaporator airflow on the high temperature performance of air conditioners

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Abstract

Two residential sized air conditioners were tested in psychrometric rooms at reduced evaporator airflows ranging from 0 to 50% below that recommended by the manufacture of each of the units. Outdoor temperatures ranged from 35 to 49 °C. One of the units used a thermal expansion valve for flow control while the other unit used a short tube orifice. Performance of the units was quantified by the capacity, power, coefficient of performance, and sensible heat ratio. Results at 35 °C indicated that the reduction in air produced a larger drop in capacity and coefficient of performance for the orifice controlled unit than the thermal expansion valve controlled unit. The power showed less than a 4% reduction for either unit as the airflow was reduced by 50%.

Keywords: Evaporator airflow; High temperature performance; Air conditioners

1. Introduction

Low airflow across air conditioning evaporators can be caused by a number of problems: undersized ducts, dirty filters, or a dirty evaporator. Recent studies by Hammarlund et al. [1] and Modera and Jump [2] reported that low airflow in evaporators was a common problem. In a survey of 12 new homes and 66 apartments, Hammarlund et al. [1] found that only 30% of new homes and 15% of appartments met manufacturers specifications for airflow through the evaporator. Approximately 10% of multifamily residences had measured airflows that were between 40 and 50% under that specified by the manufacturers for the air conditioning units.

Low evaporator airflow would be expected to reduce the capacity and efficiency of the air conditioner. With the reduced capacity, the unit would have to run longer to meet the cooling load on the residence. Thus, the unit would run longer and operate at a reduced efficiency.

In recent years, many electric utilities have provided rebates to residential customers for purchasing high efficiency air conditioners and heat pumps. The rebates have helped increase the demand for higher efficiency air conditioning units. However, if the unit is installed with an overly restrictive duct system that produces a significant reduction in airflow across the evaporator, the purchaser of the unit may not obtain the expected efficiency and the electric utility may not obtain the expected demand reduction from the unit.

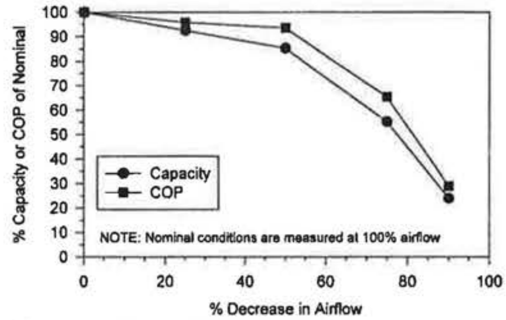

Fig. 1. Summary of reduced evaporator airflow results from Ref. [3].

A recent study [3] quantified cooling capacity and coefficient of performance (COP) as a function of a decrease in evaporator airflow (Fig. 1). For their tests, the cooling capacity decreased linearly until about 50% evaporator airflow, where it decreased much faster. For 90% airflow reduction, the cooling capacity decreased by 76%. Reductions in COP were similar to reductions in cooling capacity. The COP decreased linearly for decreasing evaporator airflow, then after 50% reduction in airflow the decrease became non-linear and the drop was greater. The main conclusion was that to maintain sufficient cooling at least 50% of the rated evaporator airflow was needed.

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A study by Chwalowski et al. [4] examined the accuracy of three computer models in predicting evaporator coil performance as a function of airflow and orientation. Because this study only focused on the evaporator and not the complete air conditioning system, it is difficult to extend their results to overall system performance when airflow is reduced.

The purpose of this research was to experimentally quantify the effect of reduced evaporator airflow on the performance of two unitary air conditioners operating under outdoor temperatures at or above design cooling conditions. One of the units used a thermal expansion valve (TXV) for flow control and the other used a short tube orifice.

Table 1
Description of the units tested for reduced evaporator airflow

Item	Unit 1	Unit 2
Manufacturer	A	B
Style	split-system	split-system
Air conditioner/heat pump	heat pump	heat pump
Nominal cooling capacity (kW)	12.3	12.3
Seasonal coefficient of performance [a]	3.72	2.93
Compressor	scroll	reciprocating
Expansion device	TXV	orifice
Evaporator airflow (m^3 min^{-1})	37.5	39.2

[a] Seasonal coefficient of performance is equal to the seasonal energy efficiency ratio (SEER) divided by 3.412.

2. Experimental apparatus

The experimental apparatus consisted of psychrometric rooms, indoor and outdoor test sections, the test air conditioners, installation and data acquisition. The air conditioning units were tested in psychrometric rooms (Fig. 2) that could control temperature and humidity for both indoor and outdoor sections of the unit. Dry-bulb and wet-bulb temperatures were maintained within ± 0.2 °C during steady-state operation.

Electric resistance heaters and chilled water coils were used to maintain room temperature. Reheat in each room was provided by four banks of 9.9 kW electrical strip heaters in the conditioning ductwork. The cooling coil was supplied with a chilled water ethylene glycol solution from a chiller. A 3.8 m^3 chilled water thermal storage tank was mounted in the system to stabilize the chilled water temperature, while reducing the cycling of the chiller. Steam from a boiler and dehumidification coils were used to add or remove humidity in the rooms.

The indoor test section consisted of the indoor airflow chamber and the indoor coil corresponding to the specific test performed. The air flow chamber fan drew conditioned air from the indoor room through the indoor test section. An adjustable damper was used to maintain the amount of airflow specified for each unit. After leaving the airflow chamber, conditioned air was routed back into the indoor room. The outdoor room section is basically the condensing unit (compressor and outdoor coil). The conditioned outdoor air entered the outdoor coil and was exhausted by the unit fan back into the room.

Two air conditioning units were used (Table 1). One was a split-system heat pump with a scroll compressor and a thermostatic expansion valve (TXV). The other unit was a split-system heat pump with a reciprocating compressor and a short tube orifice as the expansion device. The TXV controlled unit had a higher seasonal coefficient of performance (3.72) than the orifice controlled unit (2.93). The efficiency of the orifice controlled unit would just meet minimum efficiency requirements in the United States, while the TXV controlled system would be in the middle of the efficiency range of units sold in the US.

The instrumentation for all tests was divided into air-side and refrigerant-side measurements. The air-side temperature

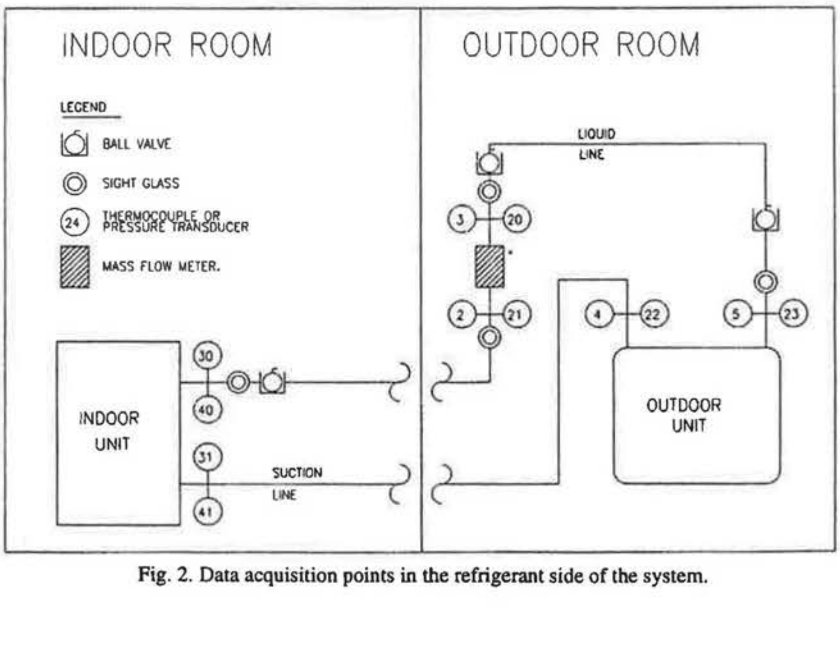

Fig. 2. Data acquisition points in the refrigerant side of the system.

measurements for both the inlet and outlet of the indoor coil unit were made using 12-element type-T thermocouple grids. Wet-bulb sensors were also used for both the inlet and outlet of the indoor coil unit. For the outdoor unit, the only air-side temperature measured was the inlet air temperature. It was measured with a single type-T thermocouple located in the sampling duct surrounding the outdoor unit.

The refrigerant-side measurements consisted of temperature and pressure measurements throughout the refrigerant lines. With the exception of the refrigerant flow rate, the outdoor unit power and the airflow differential pressure, the rest of the measurements were temperatures (dry-bulb, wet-bulb, dew point) and pressures. Temperature probes were inserted as close to the centerline of the copper tubing as possible. At each point that a temperature measurement was taken, pressure transducers were used to measure refrigerant pressure. A standard T-connection was put into the refrigerant line and a ball valve attached to one end. The pressure transducer was then attached to the valve. Refrigerant mass flow was measured with a Coriolis-type mass flow meter.

The data acquisition system converted signals coming from all the sensors in the indoor and outdoor rooms into temperatures, pressures, flow rates, or power. A data logger was used to collect data from the testing apparatus. The logger was linked to a computer where the data were visually displayed during testing. Once a test was complete, the data were transferred to another computer for processing. A total of 22 channels was monitored during testing. Each channel was scanned by the logger at 30 s intervals. Since each test was 20 min long, a total of at least 40 time intervals was recorded for each test. The data acquisition test points are listed in Table 2.

Table 2
Test points for the data acquisition system

Channel	Sensor type	Channel description
0	mass flow meter	refrigerant flow rate
1	Watt transducer	system power
2	pressure transducer	mass flow exit pressure
3	pressure transducer	mass flow inlet pressure
4	pressure transducer	condenser suction line pressure
5	pressure transducer	mass flow inlet temperature
20	thermocouple	mass flow exit temperature
21	thermocouple	condenser suction line temperature
22	thermocouple	condenser liquid line temperature
23	thermocouple	condenser inlet air temperature
24	thermocouple	evaporator inlet wet-bulb temperature
30	thermocouple	evaporator exit wet-bulb temperature
31	thermocouple	evaporator suction line temperature
32	thermocouple	airflow chamber temperature
33	wet-bulb sensor	evaporator inlet wet-bulb temperature
35	wet-bulb sensor	evaporator exit wet-bulb temperature
36	thermocouple grid	evaporator exit dry-bulb temperature
38	thermocouple grid	evaporator inlet dry-bulb temperature
40	pressure transducer	evaporator liquid line pressure
41	pressure transducer	evaporator suction line pressure
43	dewpoint sensor	dewpoint temperature
44	differential pressure	airflow differential pressure

The parameters used to describe the performance of the air conditioners were net capacity, COP, power consumption and sensible heat ratio (SHR). The net capacity was calculated by multiplying the indoor air mass flow rate by the enthalpy change of the air through the indoor fan coil unit. The capacity was the net capacity because it included both the cooling provided by the evaporator and heat gain of the indoor fan. The air enthalpies were determined from the temperature and humidity measurements upstream and downstream of the indoor fan coil unit. The air mass flow rate was determined by dividing the volumetric flow rate of the air by the specific volume of the air at the airflow chamber. The evaporator inlet and exit temperatures and pressures were used to determine the values for enthalpy. Refrigerant temperatures and pressures at the evaporator and expansion device were used to determine the values for superheat and subcooling, respectively. The COP was calculated by dividing the net capacity by the total power measurements.

3. Experimental procedure

Each of the tests was steady state, with data averaged over a 20 min interval. The conditions entering the indoor coil remained at 26.7 °C dry-bulb (DB) and 19.4 °C wet-bulb (WB) temperatures for all tests. These conditions are those used in the Air Conditioning and Refrigeration Institute test procedures [5]. The outdoor temperatures ranged from 35 to 48.9 °C.

The installation of each unit was the same. The first step was to install both the indoor and outdoor units. The next step was to set up all sensors and attach all of the refrigerant lines. A vacuum was pulled on the system and the unit leak tested. Refrigerant was weighed and added to the system according to manufacturers' recommendations. The refrigerant used for all tests was Refrigerant-22. The unit was charged when the indoor room was at 26.7 °C DB and 19.4 °C WB, and the outdoor room was at 35 °C DB temperature. After the unit was charged, the airflow through the evaporator was set to its recommended setting by using a damper on the indoor airflow chamber. Adjusting the flow using a damper would keep a relatively constant static pressure on the indoor blower, which would simulate the type of problems found in field installations (dirty/blocked filters or undersizing the duct system). Table 3 shows the combination of airflow rates and outdoor conditions used for these tests.

4. Results

4.1. Capacity and sensible heat ratio (SHR)

The capacity of the unit was based on the measurements on the air-side of the evaporator. The sensible heat ratio is defined as the sensible capacity divided by the total capacity. Like the total capacity, this measurement was made on the

Table 3
Reduced evaporator airflow tests performed

Airflow reduction (%)	Outdoor room air temperature (°C) [a]			
0	35.0	37.8	43.3	48.9
10	35.0			
20	35.0			
35	35.0	37.8	43.3	48.9
50	35.0			

[a] Indoor room conditions: 26.7 °C DB, 19.4 °C WB.

air-side of the evaporator. Fig. 3 shows a plot of the total capacity and SHR for the unit with the TXV controlled expansion. With the exception of the small increase as the airflow was decreased by 10% from its rated value, both the capacity and SHR dropped as airflow decreased. The increase in capacity could indicate that the manufacturer's recommended airflow was slightly less than the optimum for this condenser/ evaporator combination. Overall, the capacity dropped from 11.3 kW at rated airflow to 9.6 kW for a 50% reduction in airflow. For the same drop in airflow, the SHR decreased from 0.72 to 0.67.

For a fixed amount of airflow through the evaporator, the capacity decreased with increasing outdoor temperature (Fig. 4). The drop in capacity as a function of temperature was relatively constant. For example, the 35% drop in airflow at 35 °C produced a 0.73 kW drop in capacity while the same percentage drop in airflow at 48.9 °C produced a 0.71 kW. However, the percentage drop in capacity at 49 °C was larger (7.1% versus 6.4%) than at 35 °C because the total capacity of the base case was smaller at the higher temperature.

As was the case for the TXV unit, the capacity of the orifice unit dropped as airflow decreased from the rated airflow (Fig. 5). At 35 °C, the capacity dropped from 12.2 kW at the rated airflow to 9.2 kW at 50% reduced airflow. For the orifice unit, the capacity started higher than for the TXV unit. However, the capacity dropped more sharply with reductions in evaporator airflow than it did for the TXV unit.

Fig. 6 shows the plot of capacity as a function of outdoor temperature for both the base airflow case and the 35% reduced airflow case. The drop in capacity was largest at 35 °C and decreased as the temperature increased. The drop in capacity at 35 °C was 1.86 kW compared to 1.12 kW for 48.9 °C. These represented percentage drops of 15.3 and 11.9% at 35 and 48.9 °C, respectively. Thus, the orifice controlled unit produced larger drops in capacity throughout the outdoor temperature range covered in this study. This could possibly be expected because the orifice was a fixed area expansion device and was not able to adjust to conditions as can the TXV.

4.2. Power consumption and coefficient of performance (COP)

The coefficient of performance is the total capacity of the unit divided by its total power input. For the TXV unit, the COP showed similar trends to the capacity curves (Fig. 7). The COP dropped from 2.92 to 2.51 as the airflow dropped from 100 to 50% of the rated value. The power showed a small decrease from 3.89 to 3.80 kW.

The drop in COP for a given drop in airflow (35%) showed little variation as the outdoor temperature decreased (Fig. 8). For example, at 35 °C, the COP dropped by 9.4% compared to 9.3% at 49 °C.

The power and COP for the orifice unit showed similar trends to those of the TXV unit (Fig. 9). The power decreased slightly from 4.02 kW at 100% rated airflow to 3.86 kW at 50% airflow. The COP dropped from 2.69 to 2.10. The drop in COP was higher for the orifice unit than for the TXV unit for the same conditions. These trends with COP were consistent with the larger drops in capacity measured with the orifice unit.

Fig. 10 shows the orifice controlled system COP for two airflow conditions (0 and 35% reduced flow) for outdoor temperatures ranging from 35 to 48.9 °C. As with the capacity for the orifice controlled unit, the largest drop in COP

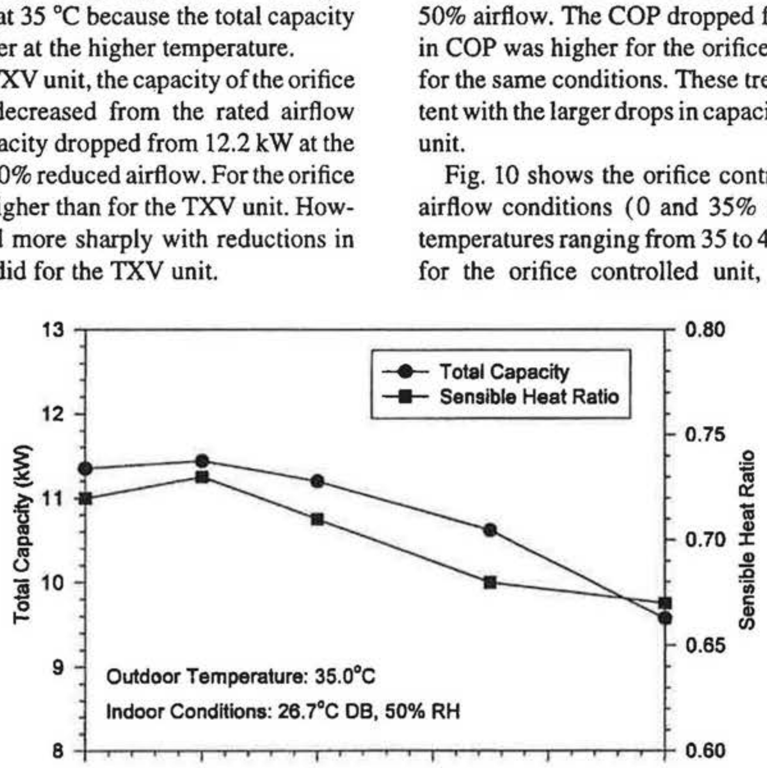

Fig. 3. The TXV controlled system total capacity and SHR at 35 °C outdoor temperature for various evaporator airflow conditions.

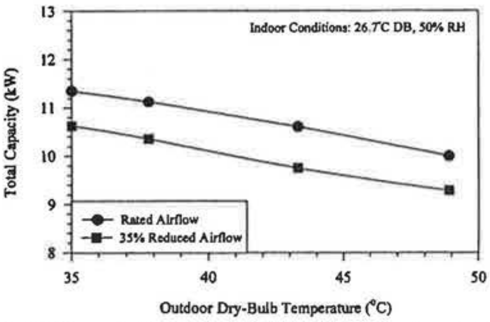

Fig. 4. The TXV controlled system total capacity at various outdoor temperatures and evaporator airflow conditions.

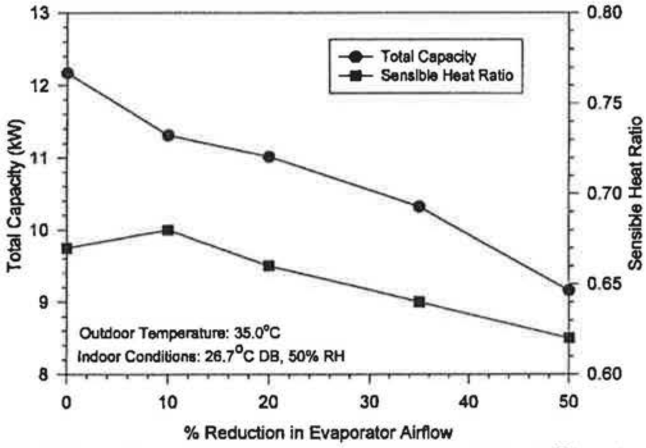

Fig. 5. The orifice controlled system total capacity and SHR at 35 °C outdoor temperature for various evaporator airflow conditions.

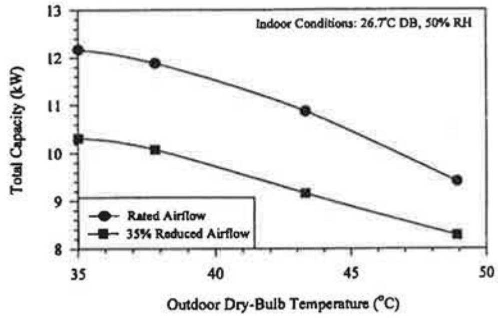

Fig. 6. The orifice system total capacity at various outdoor temperatures and evaporator airflow conditions.

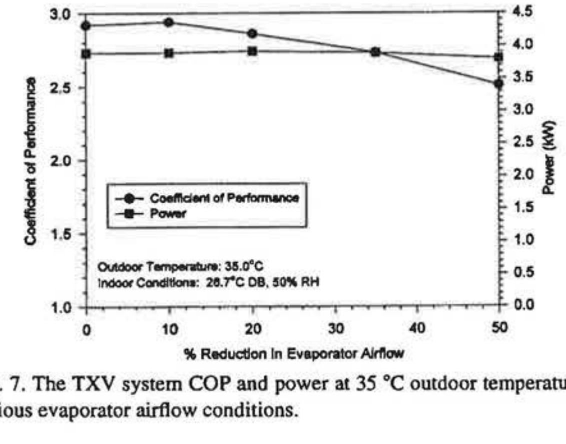

Fig. 7. The TXV system COP and power at 35 °C outdoor temperature for various evaporator airflow conditions.

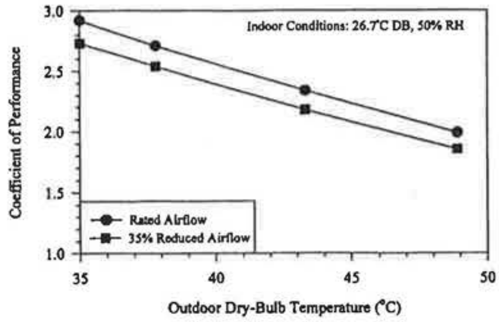

Fig. 8. The TXV system COP at various outdoor temperatures and evaporator airflow conditions.

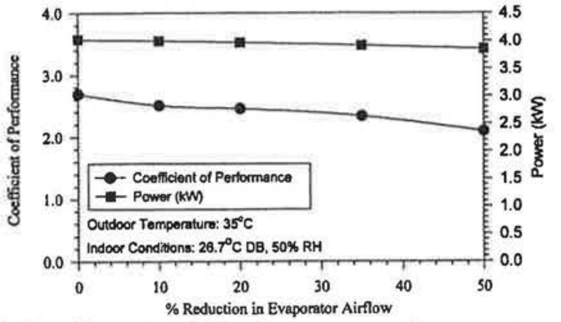

Fig. 9. The orifice system COP and power at an outdoor temperature of 35 °C for various evaporator airflow conditions.

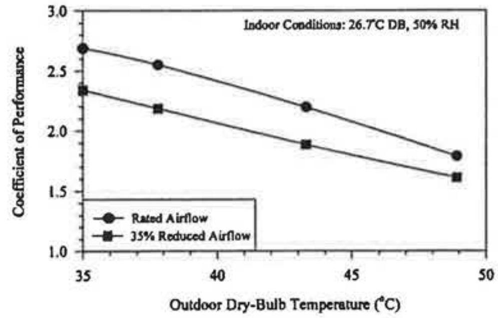

Fig. 10. The orifice system COP at various outdoor temperatures and evaporator airflow conditions.

occurred at 35 °C. The drop in COP was 13.2% at 35 °C compared to only 9.8% at 49 °C.

4.3. Other variables

One measure often used in the field as a check on performance is the temperature differential across the evaporator. The temperature differential across the evaporator increased with reduced airflow (Fig. 11). For an outdoor temperature of 35 °C, reducing the evaporator airflow increased the temperature drop across the cooling coil from 11.7 °C at normal flow to 15.8 °C at 50% reduced airflow.

Results were very similar for the orifice unit (Fig. 12). For that unit, the temperature drop across the evaporator also increased with reduced airflow. For an outdoor temperature

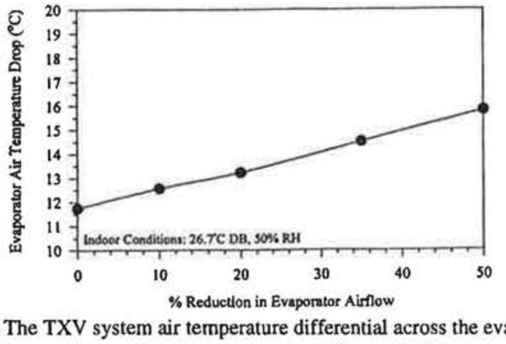

Fig. 11. The TXV system air temperature differential across the evaporator at 35 °C temperature and various evaporator airflow conditions.

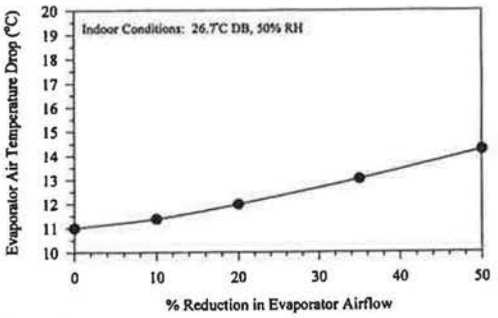

Fig. 12. The orifice system air temperature differential across the evaporator at 35 °C temperature and various evaporator airflow conditions.

Table 4
Subcooling at the TXV for various airflow conditions

Airflow reduction (%)	Superheat (°C)
0	8.3
10	7.2
20	7.3
35	7.3
50	4.6

of 35 °C, reducing the evaporator airflow increased the temperature differential across the evaporator from 11.0 to 14.3 °C at 50% reduced airflow.

For the TXV controlled system, the TXV maintained a relatively constant evaporator superheat as the airflow decreased. The subcooling before the TXV decreased from 8.3 °C at rated flow conditions to 4.6 °C at 50% rated airflow conditions (Table 4). For the orifice controlled system, there was a drop in both the subcooling before the orifice and the superheat leaving the evaporator (Fig. 13). While the superheat started at 3.7 °C for rated airflow conditions, it dropped to near saturated conditions for a 50% reduction in airflow. Further reductions in airflow could have introduced liquid into the compressor.

4.4. Summary comparison

Capacity and COP peaked at 10% reduced airflow, and then decreased to the 50% reduced airflow for the TXV unit

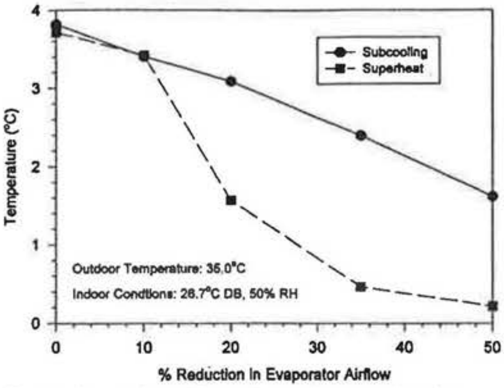

Fig. 13. Subcooling at the orifice and superheat leaving the evaporator of the orifice controlled system for various airflow conditions.

Table 5
Summary of performance degradation due to reduced evaporator airflow at 35 °C for the TXV controlled system

Airflow reduction (%)	Change in variable (%)		
	Capacity	COP	Power
10	+0.8	+0.8	0.0
20	−1.3	−1.8	+0.6
35	−6.4	−6.3	0.0
50	−15.7	−13.8	−2.3

Table 6
Summary of performance degradation due to reduced evaporator airflow at 35 °C for the orifice controlled system

Airflow reduction (%)	Change in variable (%)		
	Capacity	COP	Power
10	−7.1	−6.5	−0.6
20	−9.5	−8.4	−1.2
35	−15.3	−13.2	−2.3
50	−24.8	−21.8	−3.7

(Table 5). Both capacity and COP decreased with increasing outdoor temperature. Total power consumption increased linearly with increasing outdoor temperature. Power consumption was relatively constant with changes in evaporator airflow.

Capacity and COP decreased with reductions in evaporator air supply for the orifice controlled unit (Table 6). The highest capacity and COP were at the nominal rated airflow conditions. In general, reductions of evaporator airflow caused a larger degradation in cooling performance for the orifice unit than for the TXV unit. For example, at 35 °C outdoor temperature, a 50% reduction in evaporator airflow caused a 24.8% drop in capacity for the orifice unit compared to a 15.7% drop for the TXV unit.

5. Conclusions

This study has quantified the decrease in cooling performance caused by reduced airflow though the evaporator. While the performance drops were measurable, both the system with TXV and orifice controls required a drop in airflow of at least 20% before a 10% drop in COP was observed. The impact on the orifice controlled system was much greater than with the TXV controlled system. Even with a 35% reduction in airflow, the TXV system only showed a 6.3% drop in efficiency.

The better performance of the TXV unit would appear to indicate a benefit of this type of refrigerant flow control over the less expensive orifice control for airflow conditions that differ from the nominal design. Another study [6] relating to refrigerant charge also found better off-design performance of TXV controlled systems relative to orifice flow control. Because this study only used one TXV and one orifice controlled system, one must use caution in generalizing the differences between the performance of the TXV and orifice controlled system. More units should be tested to see if the trends in this study are inherent to systems with different flow control devices.

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