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AERODYNAMIC PERFORMANCE OF AIR COOLED CONDENSING UNITS

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ABSTRACT

The aerodynamic performance of two condensing units is investigated herein. A testing facility was built according to AMCA 210-99 and used to evaluate the aerodynamic performance of condensing units as a whole, and also of their components: axial fans and condensers. Typical performance curves, such as fan static pressure vs. airflow rate, fan power input vs. airflow rate and condenser impedance vs. airflow rate, were obtained. The condensing unit operation point was also measured and compared to that obtained through the intersection of the fan and condenser characteristic curves. It was shown that the theoretical and measured operation points are practically the same only when the fan characteristic curve is obtained using the condensing unit wall ring.

1. INTRODUCTION

Condensing units are typically composed of one or more compressors and axial fans, and by one condenser. The condenser-fan pair, along with some accessories (filters, wall ring, etc), form the so-called ventilation system. This system needs to be appropriately designed in order to maximize the condenser heat transfer rate, to minimize the unit energy consumption and to reduce the costs.

The ventilation system is usually designed based on the fan and condenser characteristic curves furnished by the manufacturers. The intersection of these curses determines the system operation point, and consequently the airflow rate supplied and the heat released by the condensing unit.

The fan performance curves are usually supplied by the manufacturer, although they do not always represent the actual fan behavior in a condensing unit. On the other hand, the condenser characteristic curve is very rarely supplied by the manufacturer. Also, the determination of the operation point through the above mentioned curves does not consider several assembly details that can affect the system performance.

To investigate this problem an experimental test facility was built according to the specifications of the standard AMCA 210-99. The apparatus is simply an open wind tunnel where the device to be tested is placed at one of its extremities and an auxiliary fan at the other. The airflow rate is obtained from the pressure differential measured across a flow nozzle or bank of nozzles. The device under test may be either a condensing unit or one of its components (fan and condenser).

The ventilation system performance of a 3.5 HP and a 5.0 HP condensing unit was assessed in this study. The 3.5 HP and 5.0 HP units were assembled with two axial fans with wheel diameters of 350mm and 450mm, respectively. The condenser and fan characteristic curves were obtained experimentally, and when possible, compared to those furnished by the manufacturers. The resulting airflow rate from the intersection of the fan and condenser characteristic curves was compared with the measured airflow rate through the condensing unit. Details regarding the experimental apparatus and also the calibration and experimental procedures are here reported.

2. EXPERIMENTAL SETUP

The standard AMCA 210-99 establishes several methods and criteria for evaluating the aerodynamic performance of fans, in terms of airflow, static pressure, input power, efficiency and fan speed. This standard can also be used as a reference for testing passive components such as heat exchangers, filters, dampers, etc. The testing method adopted in this study, referred to as *inlet chamber setup-multiple nozzles in chamber*, may be applied to a broad range of airflow rates and dimensions of the device under test (Figure 1). The tests were performed on the suction side, meaning that the air flowed from the left to the right of the apparatus.

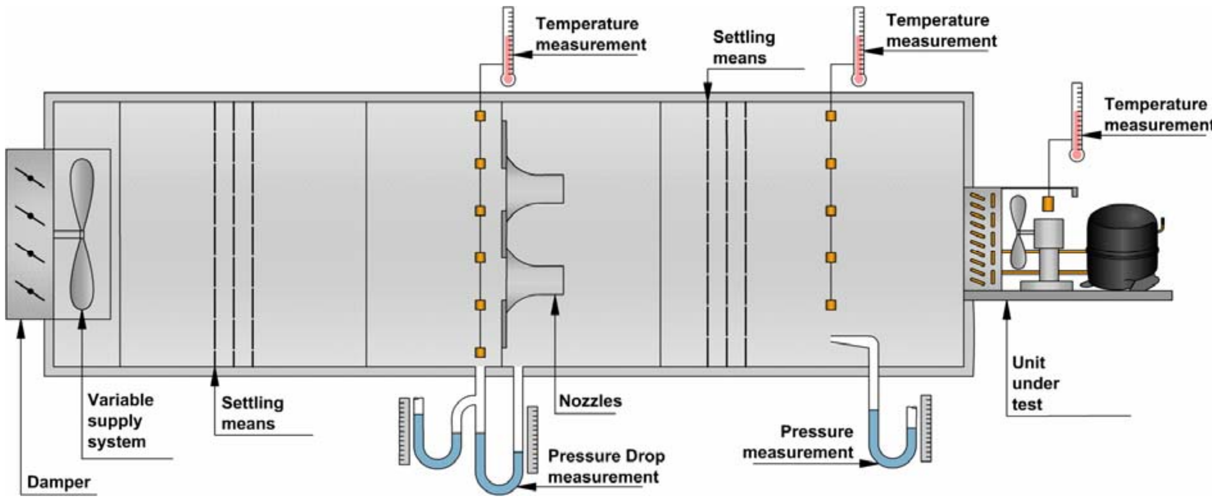

Figure 1: Diagram of the test apparatus

An auxiliary variable speed fan (Figure 2) and a damper were used to control the airflow rate through the test setup. The auxiliary fan was designed to provide sufficient pressure at the desired airflow rate to overcome losses through the test facility. Flow settling means were installed downstream of the auxiliary fan to provide the required airflow pattern through the nozzle bank. The airflow rate was calculated from measurements of the pressure differential across the nozzle bank, installed almost in the middle of the test setup. The number and size of the nozzles varied according to the airflow rate. Figure 3 shows a nozzle bank with four nozzles. Settling means were also installed downstream of the nozzle bank to ensure a uniform airflow at the entrance of the test section, placed in the right extremity of the test setup. Figure 4 shows a picture of the test setup.

Figure 2: Auxiliary fan

Figure 3: Nozzle bank

Figure 4: Test setup

The air pressures were measured by pressure transducers and also by manometers of oil column. The air temperatures were measured by type T thermocouples. The fan power, current and voltage were measured by specific transducers and the fan speed by a digital stroboscope. The output signals from the transducers and thermocouples were recorded through a computerized data acquisition system. The test setup has a length of 4.2m and a squared cross-section of 1.6m. The airflow rate operation band ranges from 110m^3/h to 10300m^3/h.

3. METHODOLOGY

The device under test was firstly assembled in the test section, located in the discharge side of the test setup (Figure 1). Both the auxiliary and the test fan were then switched on. After start-up a time period of approximately 30 minutes was needed for the fan motor temperature and speed reach equilibrium conditions. In each case, test data were taken after a stabilization time of 8 minutes. The pressure, temperature and electric parameter readings were taken every 2.5 seconds, during a time interval of 2 minutes. The fan speed was measured only once during the test period. Relative humidity and atmospheric pressure measurements were also made for each test. The pressure, temperature, fan speed and input power were directly measured by their respective transducers. The airflow rate and the fan efficiency were indirectly obtained through a set of formulae, described in detail in the standard AMCA 210-99.

4. RESULTS AND DISCUSSION

4.1 Repeatability tests

Figure 5 illustrates the performance curves of static pressure and input power versus airflow rate obtained from two different tests, both performed with the 450mm fan, using a nozzle of 10”. The results plotted in Figure 6 were obtained from three different tests, all performed with the 350mm fan, using a combination of three nozzles (4”, 4.5” and 5.5”). Figures 7 and 8 show the 450mm and 350mm fans mounted in the discharge side of the test setup, respectively.

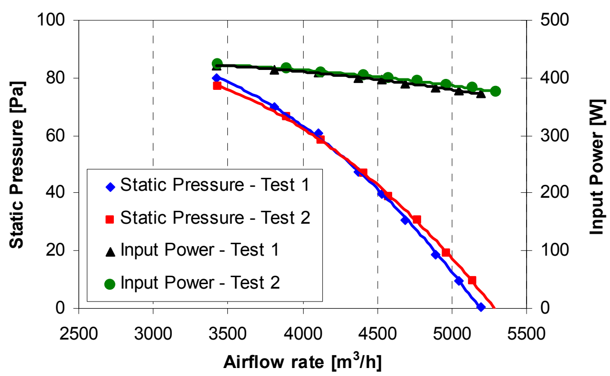

Figure 5: Static pressure and input power vs. airflow rate (450mm fan)

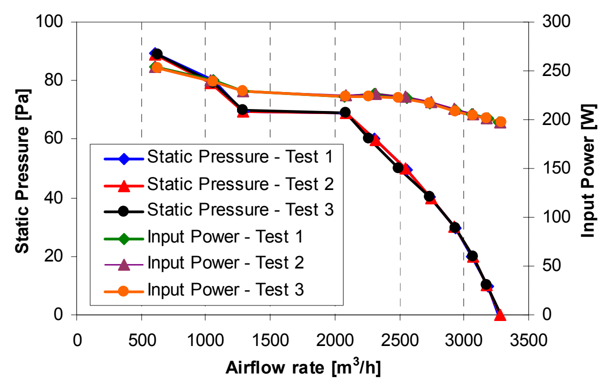

Figure 6: Static pressure and input power vs. airflow rate (350mm fan)

It can be observed that both the static pressure and the input power performance curves from the different tests are very close to each other. For the 450mm fan the maximum airflow rate deviation was found under the free delivery condition and it was less than 2% (Figure 5). For the 350mm fan the airflow rate deviations between the three tests were kept below 1% (Figure 6). The maximum input power deviation was also found under the free delivery condition and it was less than 2% for both fans. The repeatability tests were therefore considered very reasonable, since the deviations were found to be in the same range as the airflow rate (±2%) and power input (±1%) experimental uncertainties.

Figure 7: 450mm axial fan

Figure 8: 350mm axial fan

4.2 Validation study

The results obtained with the 450mm fan were compared to those obtained by the IPT (Institute of Technological Researches of the State of São Paulo) using the same fan. The test facilities of the IPT are accredited by the DKD (Deutscher Kalibrierdienst), an association of calibration laboratories of industrial firms and research, inspection and testing institutes in Germany.

Figure 9 shows the 450mm fan mounted in the suction side of the IPT test setup. It is worth mentioning that in both test setups the fan was tested with the same wall ring. Figure 10 compares the performance curves of static pressure and input power versus airflow rate obtained by the IPT with those obtained in this study (POLO). A small deviation between the POLO and IPT performance curves can be observed in Figure 10. The maximum airflow rate deviation, for a static pressure range from 0 to 70 Pa, was found to be 6%. This value is slightly above the combined airflow rate measurement uncertainties of the POLO (±2%) and IPT (±2%) test facilities. The maximum input power deviation was found to be 5% for an airflow rate range from 3500 to 5200 m^3/h. This deviation can be explained by the airflow rate and also by the power input measurement uncertainties of the POLO (±1%) and IPT (±2%) test apparatuses. It should be noted that no comparisons were made below 3500m^3/h, due to the excessive heating of the fan motor and to the consequent fan on-off operation.

The small deviations between the IPT and POLO results may be due to differences between the test apparatuses and also to the test procedures. For instance, the fans were tested in the suction side in the POLO setup, but tested in the discharge side in the IPT setup. The time required to reach the initial steady-state conditions was also different, 30 minutes and 10 minutes in the POLO and IPT test procedures, respectively.

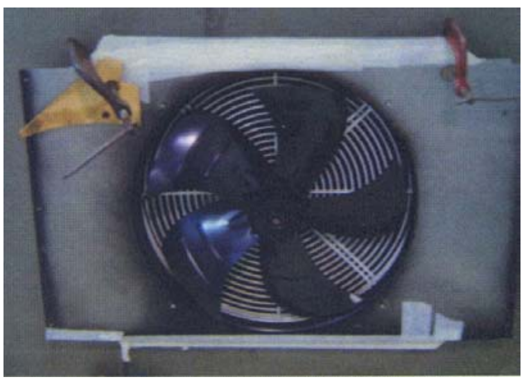

Figure 9: 450mm axial fan (IPT)

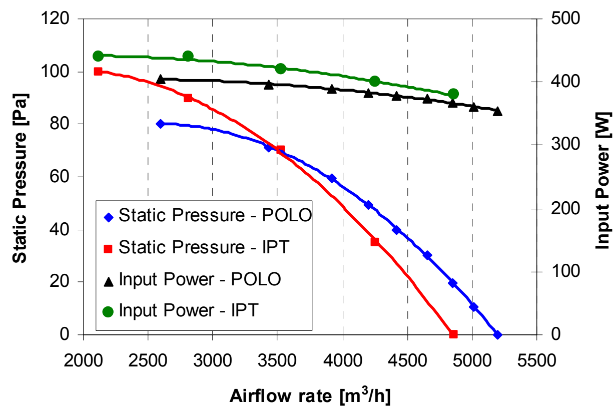

Figure 10: Static pressure and input power vs. airflow rate

4.3 Comparison with the manufacturer's data

The characteristic curves obtained with the 450mm and 350mm fans were also compared to the manufacturer's data. It is worth emphasizing that the wall rings used by the manufacturer are not exactly the same as those used in this study. Figures 11 and 12 show a comparison between the performance curves supplied by the manufacturer and those obtained in this study for the 450mm and 350mm fans, respectively.

It can be seen that the airflow rate deviation for the 450mm fan was 18% for the free delivery condition and 32% at a pressure of 90 Pa (Figure 11). Once again, no comparisons were made below 3500 m^3/h for the same reason given in section 3.2. Figure 12 shows that the difference between the static pressure performance curves for the 350mm fan was lower than that obtained with the 450mm fan. For instance, for the free delivery condition the airflow rate deviation was 8% and 18% for the 350mm and 450mm fans, respectively. In terms of input power the deviations were always kept below 5% for both fans.

The differences between the POLO and the manufacturer's performance curves are very likely due to differences in the wall rings used during the tests. The manufacturer used the original wall rings of the 450mm and 350m fans, but the tests reported herein were performed using the wall rings of the 3.5HP and 5.0HP condensing units, whose dimensions are quite different from those used by the manufacturer. For instance, the wall ring collar of the 5.0HP condensing unit is 19mm, while the wall ring collar of the 450mm fan is 100mm. The same level of difference between the wall ring collars was found for the 350mm fan, with 80mm for the fan itself and 29mm for the 3.5HP condensing unit.

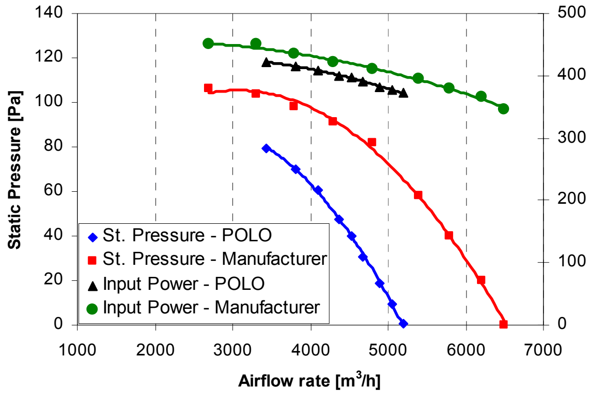

Figure 11: Static pressure and input power vs. airflow rate (450mm fan)

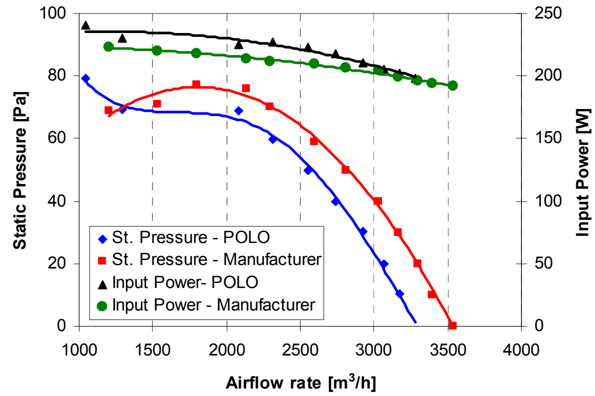

Figure 12: Static pressure and input power vs. airflow rate (350mm fan)

4.4 Determination of the operation point

In this section, the airflow rate determined through the intersection between the fan and condenser characteristic curves was compared to that obtained from the tests with the condensing unit. The performance of two fans working in parallel was also compared to the individual performance of the fans.

Figures 13 and 14 show the 450mm and 350mm fans mounted in the discharge side of the test setup, respectively. Figure 15 displays the static pressure versus airflow rate performance curve measured with two 450mm fans, working in parallel. Figure 15 also shows the theoretical performance curve obtained through the parallel association of two 450mm fans. From this figure it becomes clear that the individual performance of the fans was not affected by their parallel association. Figure 16 presents a similar comparison, but for the 350mm fan. It can be observed that, in this case, the individual performance of the fans was slightly affected by the parallel association. The maximum airflow rate deviation was found to be 7% for the free delivery condition.

Figures 17 and 18 illustrate the heat exchangers of the 3.5 HP and 5.0HP condensing units, mounted in the discharge side of the experimental apparatus, respectively. Figure 19 shows two different operation points for the 5.0HP condensing unit. The 7691 m^3/h point was obtained using the condenser and fan performance curves measured in this study. On the other hand the 8918 m^3/h point was obtained using the fan characteristic curve supplied by the manufacturer. In both cases the fan performance curve was obtained through the parallel association of two similar fans. As mentioned above, the results of this study were obtained using the wall rings of the condensing units, while the manufacturer's data were obtained using the original fan wall rings. Figures 19 and 20 show that the measured airflow rates of the 5.0HP and 3.5HP condensing units were 16% and 5% less than that obtained using the fan manufacturer's data, respectively.

Figure 13: 450mm fans in parallel

Figure 14: 350mm fans in parallel

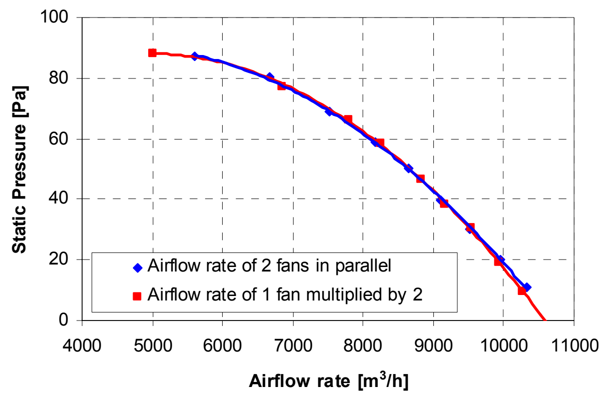

Figure 15: Static pressure versus airflow rate (450mm fan)

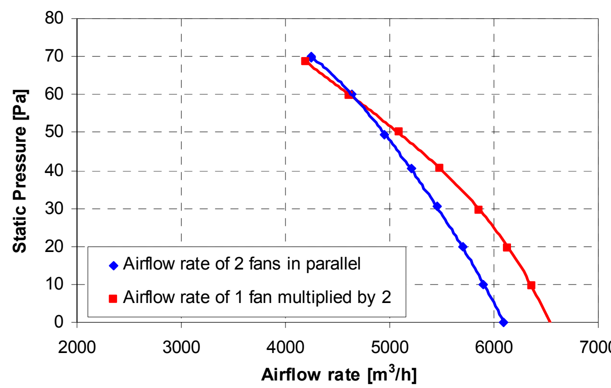

Figure 16: Static pressure versus airflow rate (350mm fan)

Figure 17: 3.5HP condenser

Figure 18: 5.0HP condenser

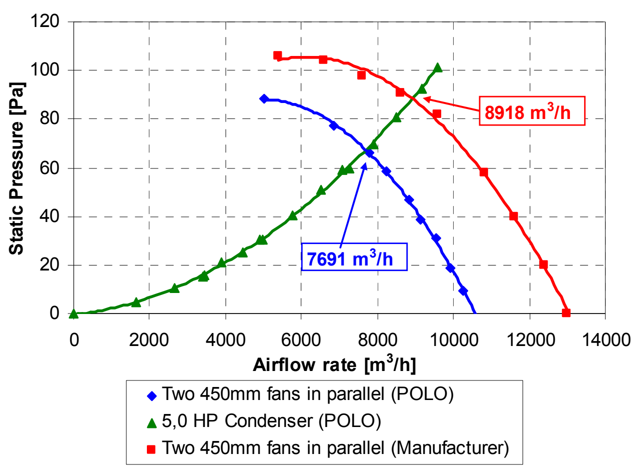

Figure 19: Operation point (5.0HP unit)

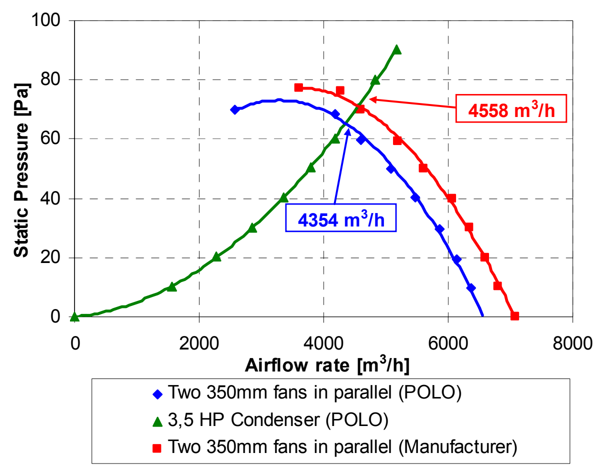

Figure 20: Operation point (3.5HP unit)

During the operation of a condensing unit the air is suctioned and discharged at atmospheric pressure. In other words, the pressure difference across the unit is null. Due to the difficulty in maintaining the pressure differential exactly at the zero value during the tests, five tests were performed using pressure differences in the range of -20 to 20Pa. The condensing unit operation point was then determined from a linear fitting using the measured data points. Figures 21 and 22 show the 5.0HP and 3.5HP condensing units, mounted in the discharge side of the test setup, respectively. Figures 23 and 24 illustrate the operation point for each of the condensing units.

Figure 21: 5.0HP condensing unit

Figure 22: 3.5HP condensing unit

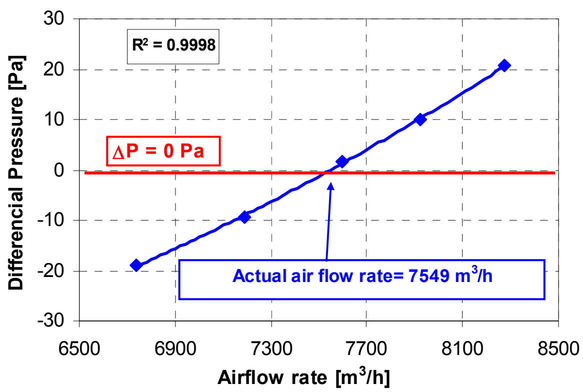

Figure 23: Operation point (5.0HP unit)

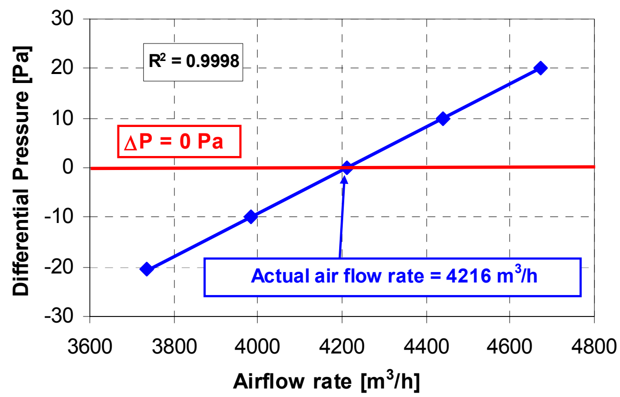

Figure 24: Operation point (3.5HP unit)

The airflow rate obtained from the tests with the 5.0HP condensing unit (7549m^3/h) was 2% lower than that obtained through the intersection of the fan and condenser characteristic curves obtained in this study (7691m^3/h) and 18% lower than that obtained through the manufacturer's data (8918m^3/h). In a similar way, the airflow rate of the 3.5HP condensing unit (4216m^3/h) was 3% lower than that obtained in this study (4354m^3/h) and 8% lower than that obtained through the manufacturer's performance curves (4558m^3/h). This indicates that the condensing unit airflow rate can be inferred from the intersection of the fan and condenser characteristic curves, but remembering that the fan must be tested with the condensing unit wall ring.

5. CONCLUSIONS

The performances of the ventilation systems of two condensing units were experimentally evaluated in this study. An experimental apparatus was built under the specifications of the standard AMCA 210-99 and used to evaluate the aerodynamic performance of the condensing units as a whole and also of their components: axial fans and condensers. Typical measured variables were the airflow rate, static pressure, fan speed and input power.

The main conclusions of this study are summarized below:
(1) The test repeatability was quite good since the deviations were found to be in the same range as the experimental uncertainties;
(2) The 450mm fan characteristic curves measured here were compared to those measured by the IPT. The deviation between the static pressure performance curves was slightly higher than the experimental uncertainties. Such a deviation may be due to differences in the test apparatuses and in the experimental procedures;
(3) The 450mm fan airflow rates measured in this study were significantly lower than those supplied by the manufacturer. Such a performance loss is probably due to differences in the fan wall rings used here and by the manufacturer;
(4) The individual behavior of the fans was only slightly affected by their parallel association. This means that the airflow rate of a parallel fan arrangement can be estimated through multiplying the airflow rate of a single fan, for a given pressure, by the number of fans;
(5) The condensing units airflow rates were very similar to those obtained through the intersection of the fan and condenser characteristic curves measured in this study. This means that the condensing unit airflow rate may be

obtained through the intersection method, but remembering that the fan must be tested with the condensing unit wall ring;
(6) The operation airflow rate obtained using the fan characteristic curve supplied by the manufacturer was 18% higher than the 5.0HP condensing unit measured airflow rate. This indicates that the condensing unit design criteria could be missed because the fan data used was too optimistic.

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