Optimization of centrifugal fan position in a particular VRF outdoor unit for European market

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Abstract. Variable refrigerant flow systems (VRF), in which refrigerant flows directly through long connecting pipes into different indoor units, are nowadays having more and more market share in central air-conditioning industry. A special flat structure of air-cooled VRF outdoor unit is expected in European market due to local architecture characteristics. Centrifugal fans with forward-curved blades are suitable under this situation. However, the layout concerning the fan position is of key importance for the heat exchange performance. Four layouts with different fan position arrangement in a VRF outdoor unit supposed to reach nominal cooling capacity of 22kW were initially designed and optimized by Ansys. Air volume rate and uniformity through the fin and tube heat exchanger (FTHE) were investigated with velocity contours. When FTHE was set perpendicularly forming a triangle compact space with neighboring cabinet shells, simulations showed a better layout in which the two fans were put side by side but with one near front and the other near rear shell in the bigger empty space. Further adjustment in position details was also carried out via more simulation on selected planes and finally, air volume measurement and nominal capacity measurement were made, matched with four ducted indoor air-conditioning units, showing good performance.

1. Introduction

Since Daikin launched the first VRV air conditioner in 1982, ongoing development on air-cooled variable refrigerant volume/flow systems (VRV, also known as VRF), as shown in Fig.1 has been achieved due to its energy saving potential, design flexibility in a maximum connection of 64 indoor units to 1 outdoor unit, larger allowances for piping length and level difference [1, 2]. The needed minimum amount of refrigerant flowing to the multiple evaporators (indoor units) is controlled usually by electronic expansion valve (EEV), which enables individual climate control of air conditioning zones equipped with different indoor units [3]. So far coefficient of performance (COP) could be 146.5% higher under heat recovery mode than under cooling-only mode [4]. For air-cooled VRFs, there are two common types of outdoor units as shown in fig.2, i.e., multi-split type in which valves arranged directly on the outdoor unit in pairs (fig.2a), branch-pipes needed type with lot of welding during installation.
(fig. 2b and 2c). Besides, front air outlet type (fig. 2a and 2b) and top air outlet type (fig. 2c) are the most two common structures based on the installation environment while fan numbers are usually adjusted according to the capacity of heat exchangers.

**Figure 1.** Schematic of VRF air conditioning systems.

For European market, especially in Spain, a flat structure of air-cooled VRF outdoor unit is particularly expected in suiting the local residential architecture. Regarding the split-type air conditioners, many researches have been reported with many experiments and simulations. Simulations help people develop air-conditioning products with less cost and more efficiency, which has been a useful method in many manufacturing companies. Dilek and Ziya [5] made a thin section model for modelling indoor units of split air conditioners (SAC) for heat transfer and fluid flow analysis. Chung-Chun et. al [6] developed a theoretical thermohydraulic and condensing phase-change model for the two-dimensional indoor unit of a small-sized SAC by using ANSYS to analyze internal flow field and heat and mass transfer performance. But for the particularly flat structure of VRF outdoor unit oriented in Spain market, there are rare reports. In order to develop this kind of VRF outdoor unit with flat structure, centrifugal forward-curved blade or squirrel-cage fans of double suction type are adopted, for they are widely used in air-conditioning and ventilating systems due to the characteristics of relatively low noise and high flow rate. The fan position plays a key role in the layout of all the devices in the unit shell, such as heat exchanger (condenser in cooling mode), compressor, throttle valves and so on. The detailed influence of fan positions on the whole air flow is unclear and needs to be optimized before prototype making with much cost. In this paper, we focus on the optimization of centrifugal fan position in this structure of VRF outdoor unit, models of which were built and simulated via the help of Ansys to achieve an optimized layout with sufficient air flow rate. Based on the simulation results, a prototype was developed and basic cooling and heating performance were experimentally investigated.

**Figure 2.** (a b c) Common types of outdoor units of variable refrigerant systems.

(a) Multi-split (with several pairs of valves). (b) VRF with front air outlet. (c) VRF with top air outlet.
2. Layout scheme

In the outdoor cabinet of the kind of VRF in this paper, a rotary compressor, a fin-tube heat exchanger (FTHE) with four rows, two forward-curved multi-blade centrifugal fans, expansion valves and other devices like oil separator are usually included. The whole cabinet is of 1840mm×1360mm×620mm. Each centrifugal fan unit covered by a volute has an impeller with two parts connected by a central plate. Each impeller has 43 forward-curved blades and an electrical motor placed at one side (left side or top side in fig.3). Opposite to the motor side forming an obstructed inlet, the other side has a relatively freer air inlet. The air outlets are equipped with a rectangular duct of 1.5m outside the cabinet, respectively.

![Diagram](image)

**Figure 3.** (a b c d) Schematic of four basic models with four different fan layouts (2D).

According to market investigation, four basic kinds of layout of the two centrifugal fans were initially designed as shown in fig.3 and fig.4 with main dimensions. From fig.3(a) to fig.3(c), the FTHE are put vertically with one end near the rear-right cabinet corner and the other end about 700mm far away from the front-right cabinet corner, forming a triangle shape with cabinet shell (about 65° angle with front wall) while the two fans are set with same intervals of 132mm but with different positions. From vertical view, in fig. 3 (a) the two fans are in the same latitude with 551mm away from the outlet on the front cabinet shell. In fig. 3(b), for fan A and fan B, the distance to the front outlet is 651mm and 351mm, respectively. In order not to obviously block the air through FTHE, the position of compressor in fig. 3 (a) is adjusted and moved near front wall based on common sense. In fig.3(c), the center line of each fan forms a 60° angle with the rear wall of the cabinet. In fig. 3 (d), the FTHE is put in the front while the two fans are put on the right side with interval of 182mm. Some other location dimensions are also given in fig.3. Fig.4 shows the 3D models corresponding to the four figures in fig.3.
3. Simulation and experiments

Under this circumstance, more air flow rate normally will bring more heat transfer. To obtain a better layout, the commercial code Fluent in Ansys was used, which has shown much feasibility in dealing with turbulent features in the last few decades. A simplified mechanical model was built with main parts of the cabinet itself, compressor, FTHE, two centrifugal fans, inlet duct and diffuser outlet duct to investigate the influence of the fan position and the layout on the air flow field. The effects of the shell thickness including the air tunnel of inlet and outlet, motor shaft, connecting pipes inside, drain tray, and some other small assembly parts were not modelled because of the need for simplification and higher mesh quality. Preprocessor Mesh was used to develop meshes and skewness was chosen as mesh metric. The FTHE was assumed to be a porous media for decreasing the nodes. Different types and sizes of grids were generated as prismatic, hexahedral meshes were mainly applied to the model. The details of the numerical grid, which involved 20681140 prismatic elements and 10995049 nodes with average skewness of 0.1882, are generally and partly shown in Fig. 5. The flow was assumed fully developed when leaving the inlet and outlet ducts. Boundary conditions were determined according to testing conditions for rated cooling capacity of air conditioners. In this simulation study, the standard $k - \varepsilon$ model with standard wall functions was chosen in the fluent solver because of its numerically robust and wide capability. The SIMPLEC numerical algorithm was adopted for the pressure and velocity coupling. The pressure parameter was discretized by second-order central difference and the parameters of $k$ and $\varepsilon$ were discretized by second-order upwind scheme.

Figure 6 and Fig.7 shows the velocity contour of the whole model and the velocity contour on the two sides of FTHE under different four layout models with 800 rpm for each fan, plotted in a 0 to 30 m/s scale. From fig.6, air flows relatively smoothly from the inlet duct, suctioned by the two centrifugal fans through the FTHE and then flows out through each outlet duct. For layout I, generally much air flows into fan B from the side near FTHE while less air flow into fan A. Air flows apart between fan A and fan B for the small interval space and motor B brings some obstruction for fan A. For layout II (-0),
compared with layout I, more air goes into fan A through its free inlet, but only a little air goes into fan B through its motor side. For layout III, fan A and fan B shares an outlet and airflow needs to turn around after leaving the fan outlet and then gathers on just one side of the outlet duct. Besides, there is a big vortex at the corner of the cabinet and smaller vortexes near the separating plate between fan A and fan B. For layout IV, the air into the FTHE is not by suction but by pure blowing. Under this structure, fan A and fan B can suction a lot of air with less fluid resistance than the former three structures. However, the air flow gets blocked in the cabinet after leaving each fan outlet and hard to get through FTHE.

Figure 5. (a b ) Mesh grid in the model.

Figure 7 shows the velocity contour of the two sides of FTHE under different four layout models also with 800 rpm for each fan, plotted in a 0 to 3.5 m/s scale and 0 to 3.0m/s scale for air-in side and air-out side, respectively. From fig.7, it can be seen the obvious difference of the air flow field and uniformity. In the former three layouts, the air velocity has lower value near the left end for both air-in and air-out sides of FTHE and uniformity is also worst in this zone with a hip joint shape on air-in side. Although the velocity contours for the former three layouts are partly similar, the general air velocity for layout II (-0) is the highest and the uniformity is the best. For layout IV, there is almost no air flow through FTHE near the fans and thus shows a big green zone. With 800 rpm for each fan, the calculated air flow rates were 7616m$^3$/h, 8216m$^3$/h, 7158m$^3$/h, 6631m$^3$/h, respectively. The layout II (-0) was so far the best. More improvement in the position details was made based on layout II (-0).
Figure 6. (a b c d) The whole velocity contour of four models with different layouts.
Figure 8 shows the two modified models based on layout II (-0), in which the relative position of fan A and fan B are adjusted. Compared with layout II (-0), fan A is put farther from the front air outlet while fan B is nearer with respective distance of 721mm and 281mm in the layout II (-1). For layout II (-2), fan B is put 301mm away from the front air outlet. The whole velocity contours of the two modified models, i.e., layout II (-1) and layout II (-2), are shown in fig.9. From fig.6(b) and fig.9 the air velocity contour out of the two fans is more balanced in layout II (-1) and layout II (-2) with more air flowing into fan A. The velocity contour of the two sides of FTHE in the two modified models are given in fig.10 plotted in a 0 to 3.5 m/s scale. Although the difference between them is not obvious, the air flow is a little more uniform for layout II (-1) with more equal distribution. The calculated air flow rates are 8216m$^3$/h, 8311m$^3$/h, 8223m$^3$/h for layout II (-0), II (-1) and II (-2), respectively. This allows small deviations in the longitude direction during real production.
In order to know more about layout II (-1), several selected planes with fan A at 750 rpm and fan B at 730 rpm are shown in fig.11 to see the inlet flow structure. The air flow near left part of each fan is less than the right part, with big vortexes in the shared outlet duct, especially for fan B, because each fan has its motor on the left side increasing the flow resistance. The air behind fan B is suctioned by fan A in priority and thus less air flows into the left (top) side of fan B as shown in fig.11 (c) with most of the displayed zone in blue. The calculated general air volume rate was 7041 m$^3$/h when fan A and fan B was set respectively at 750 rpm and 730 rpm. From the general performance including air flow rate and flow uniformity, the layout II (-1) was adopted for the next prototype making as shown in fig.12.

**Figure 7.** (a b c d ) The velocity contour of the two sides of FTIHE with different layouts

**Figure 8.** (a b ) Two modified models based on layout II (-0) with main dimensions.

**Figure 9.** (a b ) The whole velocity contour of two modified models of layout II (-0).
Figure 10. (a b) The velocity contour of the two sides of FTHE in the two modified models.

(a) The velocity contour of XY plane at left part of fan A
(b) The velocity contour of XY plane at right part of fan A.
The prototype of VRF with layout II (-1), supposed to have nominal cooling capacity of 22kW was then employed in the air volume measurement and capacity measurement (cooling and heating) as well. In the experiments, taking the vibration and balance of the two motor currents, one of the fan speeds was set 20 rpm slower than the other. When fan A and fan B were set at (800+780) rpm and (750+730) rpm, the measured air volume rate of the VRF outdoor unit prototype was 9170 m$^3$/h and 8090 m$^3$/h respectively, higher than the simulated ones, which met the design requirements. In the capacity measurement, the VRF outdoor unit matched four ducted indoor air-conditioning units and reached 21.92kW and 23.60kW for cooling and heating, respectively.

4. Conclusions
To develop a flat structure of an air-cooled VRF outdoor unit with nominal cooling capacity of 22kW, four layouts of two centrifugal fans with different relative positions were initially designed and then the
influence of different layouts on the whole air flow volume rate and uniformity on the heat exchanger were mainly investigated by simulations using commercial code Ansys. A better layout was selected and optimized by adjusting the position in the longitude direction. In a flat cabinet of 1840x1360x620mm, a fin-tube heat exchanger was put vertically forming an angle of 65° with front shell wall and constructed a triangle space with the two perpendicular cabinet walls. In this compact space, compressor, EEV, oil separator and other devices were arranged. Two centrifugal fans with forward-curved blades were put side by side in the left wide space with matching volutes. The left fan A was near the rear wall while fan B was near the front one. Based on simulations, a prototype device was successfully built and experimentally investigated. By using this layout, the calculated and measured air volume rate were over 7000 m$^3$/h and 8000 m$^3$/h. Matching four ducted indoor air-conditioning units, the prototype VRF reached cooling and heating capacity of 21.92kW and 23.60kW, respectively.

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References
[1] Geun Young Yun, Je Hyeon Lee, Han Jun Kim, Development and application of the load responsive control of the evaporating temperature in a VRF system for cooling energy savings, Energ. Buildings, 2016, 116: 638-645.
[2] Hanlong Wan, Tao Cao, Yunho Hwang, Saikee Oh, A review of recent advancements of variable refrigerant flow air-conditioning systems, Appl. Therm. Eng., 2020, 169: 114893.
[3] Ali Alahmer, Sameh Alsaqoor. Simulation and optimization of multi-split variable refrigerant flow systems. Ain Shams Eng. J., 2018, 9 (4): 1705-1715.
[4] Y. Joo, H. Kang, J.H. Ahn, et al., Performance characteristics of a simultaneous cooling and heating multi-heat pump at partial load conditions. Int. J of Refrig, 2011, 34: 893–901.
[5] Dilek Kumlutas, Ziya Haktan Karadeniz, Funda Kuru. Investigation of flow and heat transfer for a split air conditioner indoor unit. Appl. Therm. Eng., 2013, 51: 262-272.
[6] Chung-Chun Tsao, Yang-Cheng Shih, Chun-Hsiung Lin, et al., Thermohydraulic and condensing phase-change analysis within the indoor unit of a split-type air-conditioner. Case Stud. Therm. Eng., 2020, 21:100714.