Application of ASHRAE Standard 15-2010 with Respect to Multi-Evaporator Split Air-Conditioning Systems

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ABSTRACT

Multi-evaporator split air-conditioning systems, especially Variable Refrigerant Flow (VRF) systems, are receiving increased attention in the marketplace, as an alternative to water-source heat pump loops, hydronic fan-coil networks, room-by-room packaged terminal units, and even all-air systems. These systems are sometimes characterized by a significant amount of field refrigerant piping, a large number of evaporators on a common refrigerant pipe network with one condensing unit, and safety concerns related to the potential for refrigerant leaks. This paper examines the consequences and safety-based limitations imposed on such systems by ANSI/ASHRAE Standard 15-2010, explores some ambiguity not specifically covered by the Standard, attempts to assist the system design engineer on proper application of the Standard, and identifies areas where further study or research is needed to provide sound guidance on the safe application of these systems.

INTRODUCTION

Multi-evaporator split air-conditioning systems have been applied in Japan for at least two decades, and are receiving increased consideration and marketing effort in North America as a potential HVAC system choice in commercial, retail, institutional, hospitality, and multi-family residential applications [Goetzler 2007]. Indeed, the recently-renovated ASHRAE Headquarters in Atlanta, GA includes such a system in a portion of the building. HVAC system designers who once applied water-source heat pump loops, two- or four-pipe hydronic fan-coil networks, room-by-room packaged terminal units, rooftop direct-expansion (DX) units and other all-air systems, are now considering multi-evaporator split systems.

Although not all multi-evaporator split systems use variable refrigerant flow technology, the systems becoming popular today do so. These systems are commonly referred to as VRF, for variable refrigerant flow. VRF systems are a subset of multi-evaporator split systems. For the purposes of this paper, the discussion presented is equally applicable to VRF as to other multi-evaporator split systems.

The nature of a multi-evaporator split system is such that more than one evaporator is served by one common condensing unit and one common network of interconnecting refrigerant piping. Manufacturers of VRF system components advertise that as many as 40 evaporators can be included on one piping network, with up to 3100 ft (945 m) of refrigerant pipe, all connected to one condensing unit. The evaporator units may be unducted, mounted directly in the room they serve; they may be wall- or ceiling-mounted; they may be installed above a ceiling with some distribution ductwork; or they may be ducted to serve two or more rooms. It is typical to install individual evaporators room-by-room or with, at most, a few rooms served by one evaporator unit.

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Why VRF Systems May Be Selected

A list of reasons, pro and con, for selecting a multi-evaporator split-system is beyond the scope of this paper; the intent of which is simply to discuss application of Standard 15 to these systems objectively. While this paper is not a primer on VRF, some basic features and parameters of such systems are necessary for an adequate understanding of the applicability to Standard 15.

One reason VRF systems may be attractive to HVAC system designers is conduit size. The conduit (i.e., hydronic piping) of a water-based system is much smaller per unit of capacity than the conduit (i.e., ductwork) of an all-air system. In turn, the conduit (i.e., refrigerant piping) of a VRF system is smaller still than the conduit of a water-based system, for equal capacity. This is because chilled or heated water transfers energy only by accepting or giving up sensible heat from the water itself, but a multi-evaporator split system takes advantage of the phase-change of refrigerant from liquid-to-gas as a means of carrying more energy in a smaller volume. So refrigerant piping is typically smaller, and therefore consumes less ceiling cavity and/or shaft space, than does chilled water piping or ductwork for equivalent capacity.

Other reasons may be related to first cost or energy cost savings projections, or an attractive payback period, that are sometimes associated with VRF systems when compared to other system choices. For peer-reviewed literature on the pros and cons of VRF, see the April 2007 ASHRAE *Journal* [Goetzler 2007], the June 2008 ASHRAE *Journal* [Afify 2008], and the ASHRAE *Systems and Equipment* Handbook [ASHRAE 2008].

The Standard 15 Challenge

Standard 15 is a safety standard for refrigeration systems, specifying safe design, construction, installation, and operation of refrigeration systems; establishing safeguards for life, limb, health, and property; and prescribing safety requirements [ANSI/ASHRAE 2010]. In a multi-evaporator split-system, a leak of refrigerant from one evaporator unit in one room has the potential to leak virtually the entire refrigerant charge for the complete network of multiple evaporators and common network of refrigerant piping, into one room.

This is a key differentiator for multi-evaporator split-systems. A traditional DX split-system applied room-by-room would have one condensing unit for each evaporator, with no interconnection to other split-systems, and a leak would therefore discharge only that refrigerant contained in one individual split-system. A water-source heat pump system uses a modest amount of refrigerant in each individual heat pump unit, but the interconnecting piping between rooms carries water, not refrigerant. A multi-evaporator split-system has the potential to discharge a much larger quantity of refrigerant per leak occurrence than these other two system types.

At first glance, one might suppose that a large DX rooftop unit would have the potential to leak and disperse a large charge of refrigerant into an occupied space, similar to a multi-evaporator split-system. This is, of course, a concern which should be addressed via a careful application of Standard 15. On close examination, one will see that a large DX rooftop unit serves many rooms through a network of ducts. Should the evaporator in a rooftop unit open a catastrophic leak of refrigerant, the leaked refrigerant would be dispersed to multiple rooms via the ductwork, in some rough proportion to the overall room-by-room air distribution capacity. A multi-evaporator split system has a magnified concern because it could potentially discharge a comparable refrigerant charge all to one individual room or space.

Therefore, Standard 15 strives to ensure a safe application of refrigerant by limiting the maximum quantity of refrigerant, such that a complete discharge of refrigerant into the smallest enclosed room served by that system will not exceed an allowable limit. In other words, ensure that a worst-case leak scenario results in a concentration of refrigerant below that which is a danger to human occupants.

ITEMIZED OVERVIEW OF STANDARD 15-2010

Before going any further, the first and most fundamental question one must ask is: Does ANSI/ASHRAE Standard 15 apply to multi-evaporator split-systems? Anecdotally, in some circles there is a misconception that Standard 15 applies only to large chiller systems. In fact, Standard 15 applies “to the design, construction, test, installation, operation, and inspection of mechanical and absorption refrigeration systems, including heat pump systems used in stationary applications" [Standard 15-2010-2.2.a]. VRF and/or multi-evaporator split-systems are mechanical refrigeration systems and are therefore subject to Standard 15. In the remainder of this subsection, this paper will track through portions of Standard 15 that are specifically applicable to multi-evaporator split-systems.

System Classification

Moving further into the application of the Standard, in Section 4 one must select an Occupancy Classification from among seven choices: Institutional, Public Assembly, Residential, Commercial, Large Mercantile, Industrial, and Mixed. This selection becomes important later, when maximum allowable refrigerant concentrations are determined.

In Section 5, a Refrigerating System Classification must be selected. A multi-evaporator split-system is classified as a Direct System [ibid 5.1.1], for which the evaporator coils are in direct contact with the air being cooled. This is distinct from an Indirect System; for example, a chilled water fan-coil system in which only the water coil is in direct contact with the air being cooled. Furthermore, a multi-evaporator split-system is a High-Probability System [ibid 5.2.1] because the location of components (e.g., the evaporator) is such that a leakage of refrigerant from a failed connection, seal, or component will enter the occupied space. By definition, a Direct System is also a High-Probability System.

Refrigerant Concentration Limit

At this point, one must refer to ANSI/ASHRAE Standard 34 [ANSI/ASHRAE 2010] to determine the maximum allowable refrigerant concentration, or RCL (refrigerant concentration limit) and other pertinent safety classifications. By far the most common refrigerant for building VRF systems is R-410A, and that refrigerant will be followed throughout this paper. Obviously, the specific refrigerant must be verified and the appropriate factors determined for that refrigerant. Per Standard 34 Table 2, the safety classification of R-410A is Group A1 (meaning non-flammable and non-toxic). Note however, that even though R-410A is non-flammable and non-toxic, it remains a potential danger to occupants if released in excessive quantities due to its ability to displace oxygen and in the extreme case, the possibility of occupant death by asphyxiation. Therefore, Standard 34 Table 2 has established a maximum refrigerant concentration limit for R-410A of 25 pounds of refrigerant per 1000 cubic feet of room volume (390 g/m^3).

Note that refrigeration systems containing not more than 6.6 lb (3 kg) of refrigerant are exempt from the requirements of Standard 15 [Standard 15-2010-7.2.a]. This will typically exempt residential refrigerators, small packaged-terminal air-conditioners (PTACs), and the like, but most multi-evaporator split-systems can be expected to exceed 6.6 lb (3 kg) of refrigerant. So we continue forward.

Now the Occupancy Classification found earlier becomes important. For Institutional Occupancies (e.g. hospitals), the maximum refrigerant concentration is cut in half [ibid 7.2.1], effectively changing the maximum refrigerant concentration limit for R-410A to 12.5 lb/1000 ft^3 (195 g/m^3) in that classification.

The volume of the smallest individual space(s) served by an individual evaporator unit, or the smallest individual room through which refrigerant piping is installed, is used to determine the maximum potential refrigerant concentration in the event of a leak [ibid 7.3]. If that room has permanent opening(s) to adjacent room(s), the combined room volumes may be used [ibid 7.3.1]. The space above a suspended ceiling is not considered as part of the room volume [ibid 7.3.2.2]. If any proposed multi-evaporator split-system would exceed the allowable concentration limit of 25 lb/1000 ft^3 (390 g/m^3), or 12.5 lb/1000 ft^3 (195 g/m^3) if institutional, for the smallest space(s) served by an individual evaporator unit or crossed by refrigerant piping, then that system violates Standard 15.

Field Refrigerant Piping

There are several other important provisions with respect to refrigerant field piping. Joints in field refrigerant piping must be left exposed for visual inspection prior to being insulated or enclosed [ibid 8.9]. Piping must be 7'3" (2.2 m) or greater above the floor of any passageway [ibid 8.10.1], and shall not be installed in any elevator or dumbwaiter shaft, an enclosed public stairway, stair landing, or means of egress [ibid 8.10.2]. Finally, trapping of liquid refrigerant subject to hydrostatic expansion due to closing of isolation valves must be addressed by pressure relief devices and/or engineering controls [ibid 9.4.3].

Finally, the required design working pressure for both the piping and the system components must be based on not less than 122ºF (50 C) [ibid 9.2.1.c], which in the case of R-410A is 450 psia (3.1 MPa) [ASHRAE 2009]. At these pressures, a refrigerant discharge from even a small leak can occur very quickly, and the integrity of field piping joints is critical. R-410A is a colorless, odorless, tasteless gas, so occupants likely receive no warning upon accidental discharge of refrigerant.

The Machinery Room paragraphs [Standard 15-2010-8.11 and 8.12] are not discussed in this paper. While the Machinery Room provisions (e.g. refrigerant leak detection and alarms, increased ventilation) will be familiar to experienced designers of large chiller systems, Machinery Rooms are not an applicable compliance path for VRF systems. Machinery Rooms are not to be occupied by anybody other than authorized personnel [ibid 8.11.8], effectively defeating any attempt to use a Machinery Room compliance path in a VRF system serving an occupied building.

EXAMPLE CALCULATIONS

Before performing some example calculations of maximum allowable refrigerant concentrations, it may be helpful to understand some real-world operating parameters of VRF systems. As stated previously, most manufacturers of VRF system components surveyed for this paper are advertising R-410A as the applied refrigerant. Published catalog data from those manufacturers show a typical factory refrigerant charge of 2 to 3 lb per nominal ton of capacity (0.3 to 0.4 kg/kW), with an additional 1 to 3 lb per nominal ton (0.1 to 0.4 kg/kW) contained in the field piping (subject to routing and layout), for a practical range of 3 to 6 lb per ton (0.4 to 0.8 kg/kW). Operating pressures are relatively high, making the integrity of field pipe joints very important.

At one actual installed-and-operating VRF system this author viewed in preparation for this paper, serving a commercial office, the condensing unit nameplate was stamped 478 psig (3.3 MPa) high side and 320 psig (2.2 MPa) low side; 550 psig (3.8 MPa) test pressure; 96 MBh (28 kW) capacity; 23.4 lbs (10.6 kg) factory charge of R-410A. The network features three parallel field pipes (liquid, suction, and hot gas) because the system is designed as a heat pump network, allowing some zones to be in heat mode while other zones are in cooling mode. A technician knowledgeable about the system displayed receipts showing the total charge was 40-41 lbs (18-19 kg) when field piping is included.

Example 1: Commercial Office Building

The commercial office example initiated above is continued here. A simplified sketch of the layout is found in Figure 1. A total of seven (7) evaporator units, one installed within each room plus the corridor, are networked with one condensing unit. In this example, it appears the smallest individual room with its own wall-mounted VRF evaporator is 12 feet by 8 feet, with a 9 foot ceiling, for a volume of 864 cubic feet (24.5 m^3). Therefore, the maximum allowable refrigerant charge of R-410A is 0.864 x 25 = 21.6 pounds (9.5 kg). Since the actual system charge is 40-41 lbs (18-19 kg), it appears this installation does not comply with Standard 15.

But wait – if the room includes permanent openings to adjacent rooms, those rooms may be combined in the allowable concentration calculation [ibid 7.3.1]. Does the room's door qualify as a permanent opening? It seems reasonable to assume the door will be closed at least some of the time, making the opening non-permanent. If one imagines this room is a human resource director's office, for example, it must be assumed that room doors will be occasionally closed for private conversations, and other permanent openings between rooms (such as air transfer grilles) will not be permitted for reasons of crosstalk.

Does that mean VRF absolutely cannot be applied in this commercial office? Not necessarily. One option, for example, would be to remove the smallest office from the VRF system and condition it with a separate unit, perhaps a PTAC, and route the VRF refrigerant pipe so that it does not pass through the smallest office. Then the next-smallest office becomes the critical room, at 12 feet by 12 feet, with a 9 foot ceiling, for a volume of 1296 cubic feet (36.7 m^3). Therefore, the maximum allowable refrigerant charge of R-410A is 1.296 x 25 = 32.4 pounds (14.3 kg). With the reduced capacity found by eliminating one evaporator unit, along with a re-optimized piping layout, perhaps the total system charge will now be in compliance. Another allowable "fix" would be to use two completely separate VRF systems, one for each side of the corridor.

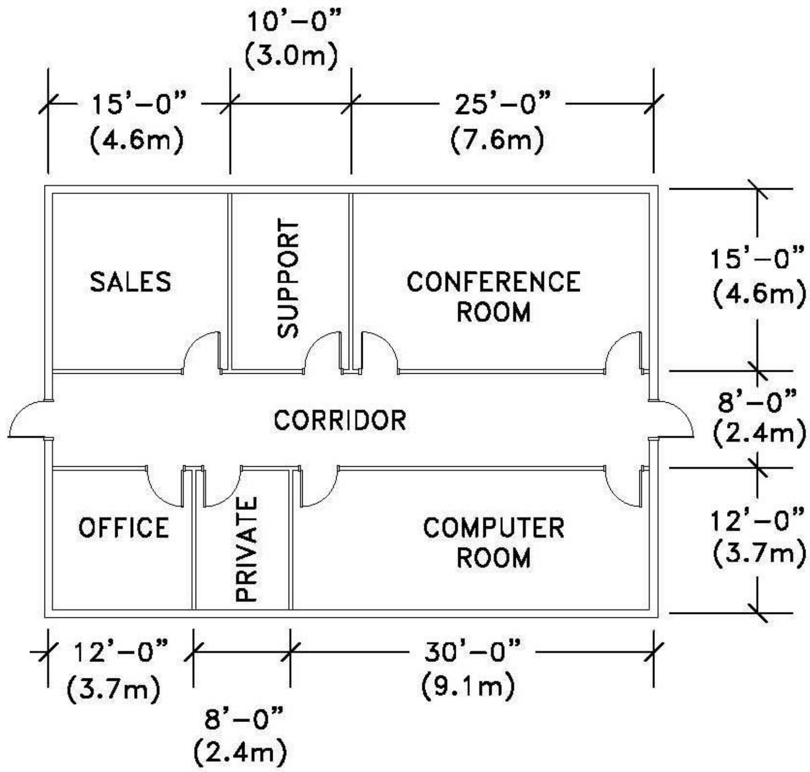

Figure 1 *Example 1 commercial office layout. A total of seven (7) evaporator units, one installed within each room plus the corridor, are networked on one common refrigerant circuit with one condensing unit.*

Example 2: Hotel Guest Rooms

Next, imagine a guest room floor of a hotel featuring a series of identical units 18 ft x 14 ft with an 8.5 ft ceiling, for a volume of 2142 ft^3 (60.6 m^3). A VRF system using R-410A is proposed, with one 0.75-ton (2.6 kW) evaporator per guest unit. What is the maximum refrigerant capacity permitted in one VRF network? The allowable concentration limit is 25 lb/1000 ft^3 (390 g/m^3) for R-410A in a Residential occupancy. Therefore, the maximum permissible total refrigerant charge is 53.5 lb (23.6 kg) including the condensing unit, all evaporator units, and the field refrigerant piping.

However, system designers must consider the ramifications of the washroom in a typical hotel guest room. The exact wording of Standard 15 reads *the volume used to determine the refrigerant quantity limits for refrigerants in Section 7.2 shall be based on the volume of space to which refrigerant disperses in the event of a refrigerant leak* [ibid 7.3]. Some hotel guests close the door of their washroom before going to bed. Indeed, some hotel properties suggest exactly that to their guests, to minimize disturbance from plumbing noise that may tend to travel room-to-room via the washrooms. With the washroom door closed, the volume of space to which the refrigerant may disperse does not include the washroom. Formal Interpretation IC 15-2007-2 confirms that the washroom of a typical hotel guest room cannot be counted in the room volume calculation. If, for example, the washroom is 6 ft x 9 ft, the effective volume of the guest room proper is reduced to 1683 ft^3 (47.7 m^3). In this case, the maximum permissible total refrigerant charge is 42.1 lb (18.6 kg) including the condensing unit, all evaporator units, and the field refrigerant piping.

In practical terms, how many guest rooms can be networked together on one VRF network? That, of course, depends on the total refrigerant charge in the system. For illustrative purposes only, let us assume an R-410A refrigerant charge of 4.5 lb per ton (0.6 kg/kW). In the example above, a 42.1 lb (18.6 kg) refrigerant charge limit would allow a 9.3 ton (32.8 kW) system, or a maximum of 12 rooms with a 0.75-ton (2.6 kW) capacity requirement per guest room. The actual capacity, room size, load calculation, refrigerant type and refrigerant charge of each system must be evaluated on a case-by-case basis and may differ from this example, but these figures are provided to lend some order-of-magnitude to the discussion.

Example 3: Hospital Patient Rooms

Finally, consider the application of VRF in the in-patient room wing of a hospital. At first glance, this may seem to be very similar to the hotel room example above. Indeed, for illustrative purposes, let us assume a standard patient room with washroom, identical in all dimensions and system characteristics to the hotel room in Example 2. Recall that for Institutional occupancies, the maximum allowable refrigerant concentration is reduced by 50%. Therefore, the results found for the hotel room example must all be cut in half for a hospital patient room: The maximum permissible total refrigerant charge would be 21.0 lb (9.3 kg) with the washroom volume is deducted, and the maximum number of rooms on one network would similarly be halved to 6 instead of 12.

Table 1. Summary of Three Example Calculations

Example	"Critical" Room Size		Maximum RCL Factor For R-410A from Std 15		Net System RCL	
Number	(ft^3)	(m^3)	(lb/1000 ft^3)	g/m^3	lb	kg
1a	864	24.5	25	390	21.6	9.5
1b	1296	36.7	25	390	32.4	14.3
2	1683	47.7	25	390	42.1	18.6
3	1683	47.7	12.5	195	21.0	9.3

Notes: Example 1a corresponds to the Commercial Office Building in the narrative.
Example 1b is similar to 1a with the smallest (critical) room removed from the VRF System.
RCL is the Refrigerant Concentration Limit.

AMBIGUITIES

A key component of the above example calculations is the determination of the volume of the smallest occupied space not connected to other spaces through permanent openings [ibid 7.3.1]. If two or more rooms are connected by permanent openings, the volume of those rooms may be combined to find the maximum refrigerant limit. A potential ambiguity exists in the evaluation of what constitutes a permanent opening. Standard 15 is a performance-based standard and not a design guide, so a good deal of engineering judgment is left to the system designer.

Does an undercut door or a transfer opening qualify as a permanent opening? If so, how large an undercut or transfer opening would be needed to qualify? These questions are not specifically addressed in Standard 15, neither to affirm nor disqualify. Clearly, undercut doors or transfer openings would *eventually* permit a large leak of refrigerant in one small room to disperse to adjacent rooms. However, without detailed study or modeling, we do not know that this will occur quickly enough to protect the safety of the room's occupants. Keep in mind that the driving force expelling R-410A from a ruptured refrigerant pipe is on the order of 450 psi (3.1 MPa), but the driving force pushing transfer air under a door or through a transfer opening is five or six orders-of-magnitude less. Ceiling-mounted transfer ducts are even more suspect, since most commonly-used refrigerants are heavier than air.

Another ambiguity is found where a separate ventilation system exists in spaces served by a VRF system. Since VRF evaporator units are often 100% recirculating type, a separate system for delivering outdoor air, such as a DOAS (dedicated outdoor air system) is often provided. An example is a hotel room continuously ventilated with outdoor air in the sleeping area and exhaust air in the washroom. How quickly would a refrigerant leak be diluted to safe levels? What if the washroom door is closed? Standard 15 offers some performance-based guidance, saying *Where a refrigerating system . . . is located within . . . an occupied space served by a mechanical ventilation system, the entire air distribution system shall be analyzed to determine the worst case distribution of leaked refrigerant* [ibid 7.3.2]. Might the "worst case" be interpreted as the case where the ventilation system is accidentally or intentionally turned off, such as for maintenance or a broken fan belt? Formal Interpretation IC 15-2007-3 confirms that dilution by supply and/or exhaust air ventilation should not be considered.

Finally, this author is anecdotally aware of another proposed approach to refrigerant leak management, one that involves the installation of automatic shutoff valves within the field refrigerant piping. In conjunction with refrigerant leak detectors, the intent is to isolate a leak to one piping segment and limit the quantity of refrigerant that can be leaked between valves to a quantity below the allowable concentration limit. This approach is not permitted by Standard 15. The only place within Standard 15 that refrigerant leak detectors are addressed is within the Machinery Room compliance path, and we have already discussed that the Machinery Room compliance path is not appropriate for occupied space. Furthermore, trapping of liquid refrigerant subject to hydrostatic expansion due to closing of isolation valves must be addressed by pressure relief devices and/or engineering controls [ibid 9.4.3.1]. Multiple pressure relief devices within the building could themselves become leak points or source of maintenance needs.

FURTHER RESEARCH

To assist design engineers applying Standard 15 and AHJs (authorities having jurisdiction) enforcing Standard 15, some further research may be helpful to give better definition to issues identified above as Ambiguities. For example, it seems advisable to further define whether an undercut door or a transfer opening qualifies as a permanent opening between rooms, and if so, how large would it need to be, and how high or low it should be installed. By conducting sophisticated computer modeling using CFD (computational fluid dynamics), many different leak scenarios can be studied, using varying leak rates, varying transfer opening sizes, varying transfer opening locations with respect to floor level, and so forth. Further CFD models could predict dilution rates for refrigerant leaks in separately-ventilated spaces, such as those served by both a VRF system and a DOAS system.

Subsequent to publication of this paper, this author plans to prepare and submit one or more RTARs (Research Topic Acceptance Requests) through proper ASHRAE channels, proposing research for the above.

CONCLUSIONS

First and foremost, this paper is intended to serve as a reminder that careful application of ANSI/ASHRAE Standard 15 is necessary when designing multi-evaporator split air-conditioning systems, because of the risk of significant refrigerant leaks in small spaces. Just because a VRF system manufacturer advertises that a large number of evaporators (up to 40 by some) and lengthy field piping can all be interconnected on one refrigerant piping network, does not mean that one should do so without first reviewing the safety provisions of Standard 15. Yet within the bounds of Standard 15, we have seen that multi-evaporator split air-conditioning systems can be properly selected, designed, installed, and operated.

In the Introductory section, it was observed that one attraction of VRF is the small size of the refrigerant piping – smaller than hydronic piping and much smaller than ductwork for equivalent capacity. This inherent advantage also provides a challenge, however. Operating pressures are relatively high, making critical the integrity of all the field joints. Some systems use three refrigerant pipe mains to provide a heat pump option, increasing the number of potential leak points. Routing is also key, since refrigerant piping must not be installed within a means of egress, and it is advantageous not to route refrigerant piping through smaller enclosed rooms with a lower maximum refrigerant concentration level.

It was also observed that energy cost projections, first cost projections, or an attractive cost-payback period, are other reasons for choosing VRF. After reviewing the three example calculations in this paper, we have seen there may be some limitations on system size in order to avoid exceeding the maximum allowable refrigerant concentration level. In some cases, it may be necessary to use more, smaller, separate VRF system networks in lieu of one larger system; or it may be advantageous to remove small enclosed rooms from a VRF system and serve those rooms using a different HVAC system choice. All of these options have a cost associated with them, and it is recommended that the HVAC system design engineer evaluate all of these factors carefully.

Finally, it is clear that some ASHRAE Research would be helpful in better defining critical volume calculation parameters. How to treat an undercut door or a transfer opening, or a parallel ventilation system, really need further exploration. Until then, the safest and most conservative approach is not to rely on undercut doors, transfer openings, or parallel ventilation systems as a path to compliance.

Although this paper focuses on ANSI/ASHRAE Standard 15, most of the key provisions addressed herein are also found, with essentially similar wording, in Chapter 11 of the 2009 *International Mechanical Code* [ICC 2009]. Therefore, it should be anticipated that the safety-based limitations and restrictions on multi-evaporator split air-conditioning systems highlighted herein will be Code-enforced in much of North America.

DISCLAIMER

While the author is a member of SSPC 15, the opinions and interpretations of Standard 15 offered in this paper are those of the author alone, and shall not be construed as official interpretations by the SSPC 15 committee. The summary of Standard 15 in this paper is specific to multi-evaporator split air-conditioning systems, and many portions of the Standard not specific to these systems were not discussed for reasons of brevity. All numerical examples provided in this paper are for illustrative purposes only; every building is unique and must be studied with respect to Standard 15 on a case-by-case basis.

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