Estimation of Lifetime CO_2 Emissions of Commercial HVAC Equipment through Building Energy Modeling

Comparison of R-32 Packaged Rooftop (RTU) Equipment Against R-410A Variable Refrigerant Flow (VRF) systems in a Small Office in California

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Executive Summary

Manufacturers, policymakers, and other industry stakeholders have for several years been preparing to phase down the refrigerant R-410A due to its high global warming potential (GWP). The environmental impact of R-410A (GWP 2,088) can be greatly reduced by alternatives including R-32 (GWP 675) and R-454B (GWP 467), however many of these alternatives are classified as A2L "mildly flammable" fluids. One short-term challenge that has arisen for manufacturers transitioning to lower-GWP alternatives relates to the disconnect between refrigerant regulations and building codes. Packaged systems like rooftop units (RTUs) are able to comply with building codes by limiting the quantity of A2L refrigerant to acceptable levels. Variable refrigerant flow (VRF) systems, which can contain several indoor units equivalent to multiple RTUs, often contain more refrigerant charge and are not compliant with current building codes for use with A2Ls. While building code updates to enable safe use of A2Ls are expected in the near future, a temporary barrier to the adoption of VRF systems may emerge if regulations phase down the use of R-410A prior to these changes.

This paper compares the environmental impacts of RTUs using R-32 against VRF systems continuing to use R-410A in equivalent circumstances. It is necessary to examine the impacts of equipment design and policy changes in a holistic manner that considers both the direct emissions of greenhouse gas (GHG) refrigerants and the indirect emissions resulting from the energy consumption of such systems. Analyses were performed on a small office building in two California locations simulated using CBECC-Com and EnergyPlus™. Simulations showed dramatic energy savings for the VRF systems relative to RTUs. While the direct refrigerant emissions of the R-410A systems were higher, most simulations showed comparable and even lower lifetime emissions for the VRF systems due to their simulated energy savings relative to the RTUs. It follows that when future VRF systems can be implemented with lower-GWP refrigerants, their lifetime emissions will be considerably lower than conventional systems.

Similar work could be repeated for a larger study of additional locations, buildings, and equipment types. The comparisons and conclusions in this report represent one set of modeling choices and assumptions to which the results are highly sensitive. Future work to expand these analyses could consider sensitivity to model assumptions and statistical distributions of system performance, refrigerant leakage rates, and other critical parameters. This work provides a comprehensive approach to compute both direct refrigerant emissions and indirect emissions from energy consumption. It also demonstrates that it is feasible, with current information and modeling tools, to rationally identify choices that minimize environmental impacts of HVAC equipment.

1. Introduction

Heating, ventilation, and air conditioning (HVAC) equipment in buildings contributes to climate change mainly through indirect emissions from electricity and fuel consumption as well as direct emissions of greenhouse gas (GHG) refrigerants. Businesses and regulatory bodies are increasingly interested in understanding these contributions to global warming because of the high carbon intensity of building heating and cooling. Rather than focusing solely on minimizing energy consumption or reducing the global warming potential (GWP) of refrigerants, the emissions of HVAC systems must be examined holistically. Well-established frameworks for these analyses include the Total Equivalent Warming Impact (TEWI), which combines the direct GHG emissions and indirect emissions due to energy consumption, as well as Life Cycle Climate Performance (LCCP), which adds detail to the former, creating an estimate of total cradle-to-grave emissions, including emissions due to manufacturing and end-of-life disposal/recycling.

In order to determine the emissions of HVAC equipment, it is necessary to estimate the frequency and quantity of refrigerant leaks as well as system energy consumption, which is highly dependent on climate and building design/operation. The magnitudes of indirect and direct emissions can vary widely depending on equipment usage and refrigerants. As such, comprehensive analyses of direct and indirect emissions are critical for correctly prioritizing system design improvements and regulatory changes that can achieve the greatest positive environmental impacts.

In this report, several illustrative scenarios of HVAC systems in California buildings are examined to demonstrate the differences in energy consumption and direct and indirect CO_2-equivalent emissions.

2. Background

2.1. Energy Modeling

In many HVAC systems, the indirect emissions due to energy consumption far exceed emissions from other sources. Energy consumption also requires the greatest modeling detail to accurately predict. Building Energy Modeling (BEM) refers to the simulation of whole buildings using physics-based software models that contain numerous details pertaining to building construction, equipment, occupancy, and controls. California Building Energy Code Compliance for Commercial/Nonresidential Buildings (CBECC-Com) is a program developed by the California Energy Commission (CEC) to evaluate commercial building compliance with energy codes. CBECC-Com uses the US Department of Energy's EnergyPlus™ software to perform building model simulations. Since EnergyPlus™ is a highly developed tool and CBECC-Com has been thoroughly vetted by CEC, this framework is used throughout this report to simulate representative California commercial buildings.

2.2. HVAC Systems

For the purposes of this analysis, two main system types are of interest: 1) "conventional" systems, which are single-zone packaged rooftop units (RTUs), and 2) Variable Refrigerant Flow (VRF) systems. Conventional RTUs consist of a packaged system of a compressor, a condenser, and an evaporator, usually located on a building roof where heated or cooled air is provided to the conditioned space via ducted air. In contrast, a VRF system consists of one outdoor unit serving numerous indoor units that are connected via refrigerant piping. The use of a variable-speed compressor and multiple indoor units can allow for better modulation of capacity and higher part-load efficiency than conventional equipment.

The high GWP of common refrigerants has been a focus for manufacturers and regulators seeking to reduce emissions from HVAC systems. R-410A is the most common refrigerant found in air conditioning and heat pump systems at present; its GWP is 2,088 times the warming potential of CO_2 (Forster et al., 2007)[*]. Near-term policy and equipment design changes will soon begin the phase down of R-410A and replacement with lower-GWP alternatives which may include R-32, R-454B, R-466A, and others. These alternatives have significantly lower GWPs than R-410A, however refrigerant selection requires balancing several tradeoffs between GWP, safety, and material compatibility. Most R-410A alternatives including R-32 and R-454B are classified as A2L "mildly flammable" and while R-466A is classified as A1 "non-flammable", manufacturers have expressed concerns regarding its chemical stability and material compatibility (Bitzer, 2020; Kujak, 2020).

At present, many building codes prevent the use of VRF systems with A2L refrigerants given that VRF's are direct systems (where refrigerant has the potential to leak into the occupied space) and the charge amount can exceed the currently allowable limit. The combination of an R-410A phase down and building codes that prevent the use of A2L refrigerants could significantly hinder the implementation of VRF systems. It is thus important to consider the

[*] US EPA and California ARB use GWP values from IPCC's AR4 report rather than the more recent AR5 report. For consistency those values are used throughout this report. R-410A and R-32 have GWPs of 1,924 and 677, respectively in the AR5 report (Myhre et al., 2013).

temporary environmental impact of continuing to use R-410A VRF systems until the building and safety standards are better aligned.

In this report, two basic scenarios are examined to compare their environmental impacts: 1) the use of conventional RTU equipment with lower-GWP (675) R-32, versus 2) a VRF system using the currently used, higher-GWP (2,088) R-410A. The intent is to understand both the general differences in energy performance and, more importantly, the life cycle impact of both direct and indirect emissions.

3. Methodology

3.1. CBECC-Com Modeling

Work was performed using CBECC-Com 2019 1.2, which was the latest version approved by the CEC at the time of analysis. Simulations were performed on the small office sample building model included by default with the installation of CBECC-Com (BEES, 2020). This prototype building model is based on the DOE reference building (Deru et al., 2011) of the same name but includes several modifications relevant to current California energy code compliance. The building model represents a single-story office with five conditioned zones totaling 5,500 ft^2 plus an unconditioned attic.

To model the VRF system, the default model "021013-OffSml-VRFSys.cibd19" was used; in order to scale equipment capacities with different simulated climate conditions, the option "Auto-size Proposed HVAC Capacities using EnergyPlus" option was enabled.

To model the conventional system, the model "020012S-OffSml-CECStd.cibd19" was used. Upon observing that the simulated energy results for the "standard design site" did not match those of the VRF it was determined that the schedules (occupancy, lighting, loads, etc.) for the two models were not equivalent, so the conventional model was modified to align with the VRF model.

Further, fan energy values calculated by these models were unrealistically high, which was the result of two factors: 1) the calculation method and default values for fan energy were inconsistent with real system data and 2) a bug in CBECC-Com incorrectly wrote VRF fans to operate 24/7[†] with constant air volume. To rectify this, the following changes were made:

[†] Confirmed by June 23, 2020 meeting and review by CEC staff and resolved in subsequent versions of CBECC-Com

- The VRF fan power was changed from the default 0.687 W/cfm to 0.101 W/cfm based on an average of 9 indoor unit specifications provided by system manufacturer Daikin (examples from two manufacturers are included in Appendix A: Example VRF Fan Specs)
- The conventional system (RTU) fan energy calculation was changed to use the static pressure method with 2.5" W.C. pressure drop, consistent with DOE reference building models (Deru et al., 2011). Fan efficiencies, sizing, and schedules were retained from CBECC-Com's small office model.
- The VRF fan schedule was modified in the EnergyPlus file to follow the "OfficeHVACAvail" schedule rather than run 24/7
- The VRF fan was modified in EnergyPlus to operate as variable volume rather than constant volume, in line with actual equipment operation.

These model modifications were performed in order to make a fair comparison between the two building models and achieve more realistic performance predictions. The changes described in the last two bullets require running the VRF system simulations directly in EnergyPlus™ rather than through the CBECC-Com interface. Data was taken from public sources and actual product specifications to ensure reasonable results. Other modelers may achieve different results when examining equipment from different manufacturers or different building configurations.

3.2. Indirect Emissions from Energy Consumption

The consumption of energy by HVAC equipment indirectly results in the emission of CO_2 by either the direct combustion of natural gas or the generation of electricity. Natural gas emissions are straightforward to calculate: CBECC-Com reports CO_2 emissions in its user interface and dividing by the gas energy yields a value of **0.053 metric tons CO_2 per million Btu**. This is consistent with other public information (EIA, 2016).

CO_2 emissions from electricity generation are currently in flux as governments and utilities strive to decarbonize through the increased implementation of renewables. This is important for this analysis because emissions due to electricity consumption will likely decrease significantly over the lifetime of a piece of HVAC equipment, which may remain in service for 20 years. California is currently undergoing an aggressive decarbonization effort that seeks to achieve carbon neutrality and 100% clean energy by 2045 (Executive Department of the State of California, 2018). CBECC-Com has inbuilt features to compute the CO_2 emissions of buildings: dividing the calculated CO_2 emissions by electricity consumption for a CBECC-Com simulation results in an average emissions factor of 0.19 kg CO_2-eq./kWh, 30% lower than the present value of about 0.27 kg CO_2-eq./kWh (California Air Resources Board, 2020a).

Since CBECC-Com could not be used for modeling VRF systems in this study due to software errors in fan modeling described in Section 3.1, it was necessary to simulate these systems directly in EnergyPlus™ and multiply the electricity consumption by an appropriate emissions factor. A second approach to estimate the lifetime electricity emissions factor is to take the average of the projected emissions over the 20-year lifetime of the system. California Air Resources Board provided projections of emissions from 2018-2030 (California Air Resources Board, 2020a). After 2030, a linear trajectory down to 0 carbon emissions in 2045 (Figure 1) was assumed. The 20-year average of these projected values from 2020 through 2039 results in nearly the same value, **0.19 kg CO_2-eq./kWh,** as seen in CBECC-Com results. In subsequent calculations of indirect emissions, the electricity consumption of equipment is multiplied by this emissions factor representing average emissions over the equipment's life.

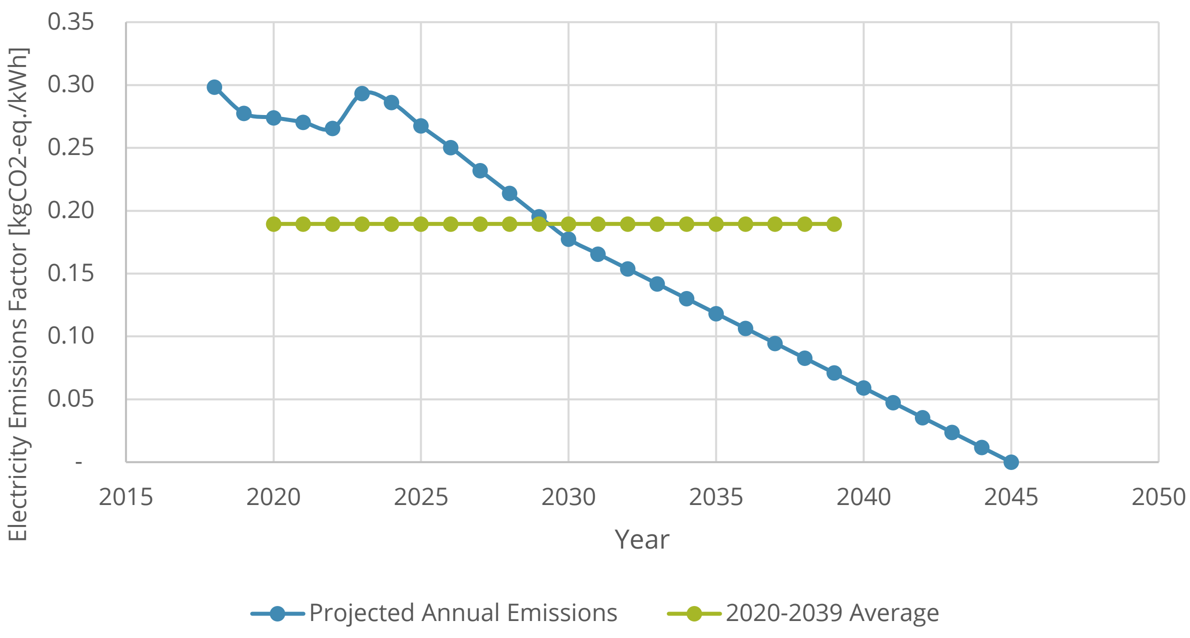


Figure 1: Electricity Emissions Projections for California

3.3. Refrigerant Charge and Leakage

In order to determine the impact of direct GHG refrigerant emissions into the atmosphere, it is necessary to determine the refrigerant quantity (charge) contained within typical systems and estimate the rates at which refrigerant is lost to the atmosphere. To do this, Daikin U.S. provided refrigerant charge quantity data for 79 RTUs from several manufacturers from the AHRI directory (AHRI, 2020). Figure 2 shows the reported charge quantity in existing R-410A systems; the selected products are compliant with the 2023 minimum efficiency performance standard (MEPS) of California's Title 24 building energy code. As expected, required refrigerant charge increases with system capacity and heat pumps (HP) generally require more charge than air conditioners (AC).

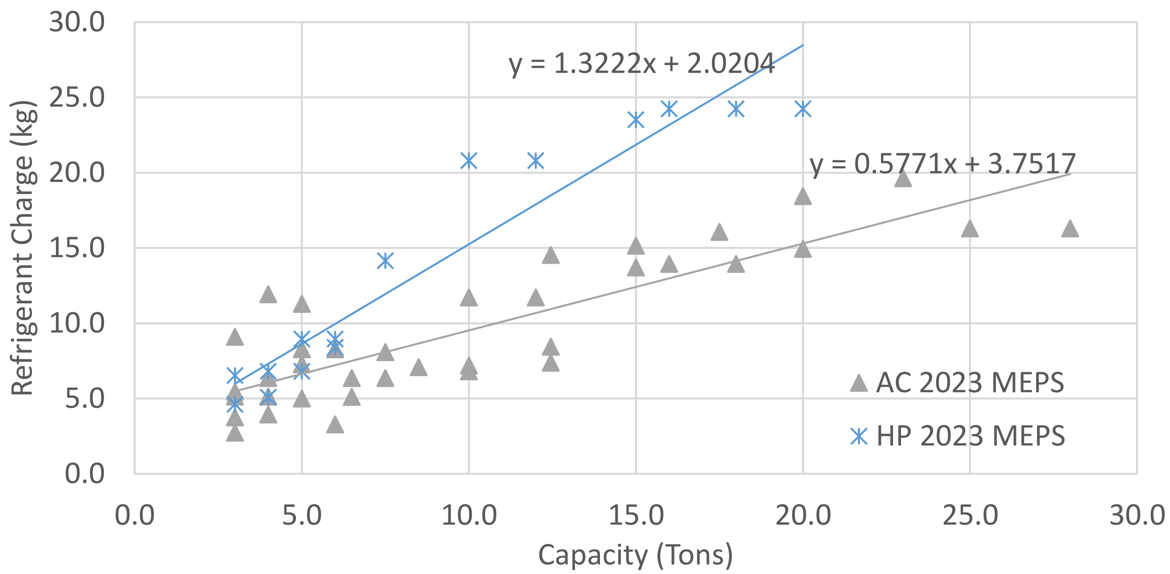


Figure 2: Industry Refrigerant Charge Quantities (R-410A)

The small office model used for this study has 5 separate conditioned zones. While it may be possible to condition this space with a single large RTU ducted to each zone, this is not the intent of the reference building model. The model contains individual system parameters for each zone. Referring back to the support documents used to develop DOE's reference buildings, the authors point to the ASHRAE handbook and state "A general design practice is to use multiple units to condition the building, with less duct work and the flexibility to maintain comfort in the event of partial equipment failure" (Jarnagin et al., 2006). As such, it is necessary to determine the size and refrigerant quantity for each of the 5 RTUs that would be installed in the building. CBECC-Com was used to perform simulations and size the HVAC equipment for the small office building when located in Los Angeles and Fresno. Equipment for each zone was rounded up to the nearest half ton of capacity and the regressions from

Figure 2 were used to calculate the R-410A charge of each AC and HP RTU used in later building simulations.

Finally, it must be considered that the different fluid properties of R-410A and R-32 will result in different charge quantities. This results from several factors: 1) liquid densities of R-32 are nearly 10% lower than R-410A at relevant conditions and 2) the higher heat capacity and higher efficiency of R-32 allows for systems to operate with less charge and to be constructed with smaller heat exchangers. This difference in charge has been well-documented by multiple authors who typically find R-32 system charge to be about 20% lower than R-410A (Kamioka, 2014; Pham and Monnier, 2016; Schultz et al., 2015). These charge reductions were observed in drop-in replacement or soft-optimization scenarios where systems were originally designed for R-410A; in reality, the higher-efficiency of R-32 allows for the use of smaller heat exchangers and a further reduction of charge. Based on these references and further data provided by Daikin, a conservative estimate was made in assuming R-32 charge to be 75% of equivalent R-410A system charge. Table 1 and Table 2 summarize the charge quantity estimates of the individual RTUs considered in this analysis.

Table 1: RTU Charge Quantity Calculations: LA

LA	Rated capacity [tons]	Rounded [tons]	R410A AC Charge [kg]	R410A HP Charge [kg]	R32 AC Charge [kg]	R32 HP Charge [kg]
RTU 1	4.06	4.5	6.35	7.97	4.76	5.98
RTU 2	2.24	2.5	5.19	5.33	3.90	3.99
RTU 3	1.21	1.5	4.62	4.00	3.46	3.00
RTU 4	1.87	2	4.91	4.66	3.68	3.50
RTU 5	1.26	1.5	4.62	4.00	3.46	3.00
				Building Total:	19.26	19.48

Table 2: RTU Charge Quantity Calculations: Fresno

Fresno	Rated capacity [tons]	Rounded [tons]	R410A AC Charge [kg]	R410A HP Charge [kg]	R32 AC Charge [kg]	R32 HP Charge [kg]
RTU 1	4.2	4.5	6.35	7.97	4.76	5.98
RTU 2	2.37	2.5	5.19	5.33	3.90	3.99
RTU 3	1.56	2	4.91	4.66	3.68	3.50
RTU 4	2.69	3	5.48	5.99	4.11	4.49
RTU 5	1.9	2	4.91	4.66	3.68	3.50
				Building Total:	20.13	21.46

VRF systems were sized at 10-tons for LA and 12-tons for Fresno based on building simulations. Daikin U.S. shared refrigerant charge data for existing heat pump and heat recovery VRV units of these sizes using R-410A (Table 3).

Table 3: VRF System Charge Quantities

Tons	R410A VRF HP (kg)	R410A VRF HR (kg)
10	16.74	23.45
12	18.60	23.13

Assumptions about leakage rates and service and recovery procedures have great influence when considering the impacts of direct refrigerant emissions into the atmosphere. Such values are inherently difficult to quantify because data is not routinely collected and outcomes can vary widely from case to case. Several organizations have provided guidelines for estimated refrigerant leakage rates based on available research and data. The International Institute of Refrigeration (IIR) published a guideline for LCCP analysis in 2016; the assumed annual leakage rates for commercial packaged and split units are 5% and end-of-life (EOL) leakage is assumed to be 15% (IIR LCCP Working Group, 2016). These authors cite a report by the UN Environmental Programme (UNEP, 2002). The Intergovernmental Panel on Climate Change (IPCC) published a 2003 guideline on national GHG inventories which was updated in 2019 (IPCC, 2019). This document lists 1-10% as the typical range of annual leakage rates and states 0-80% of initial charge remains at EOL and can be recovered with an efficiency of 0-80%. In a peer-review of EPA's emissions vintaging model (VM) citing reviewers comments, IPCC, and UNEP, the assumed annual leakage rates for small and large unitary ACs were updated to 4.7% and 4.3%, respectively (ICF, 2018). California's Air Resources Board also estimates leakage rates in their F-gas inventory (California Air Resources Board, 2020b); these values are summarized in Table 4. For this analysis, a **7% annual leakage rate and 20% loss at EOL** was chosen for all systems. By selecting a leakage rate higher than other sources, this analysis suggests considerably higher direct refrigerant emissions than would be estimated by lower assumptions.

Table 4: Summary of Leak Rate Assumptions from Literature

Source / Type	Annual Leakage Rate [%]	EOL Leakage Rate [%]
IIR LCCP Guideline "Commercial Packaged and Split Units"	5.0	15.0
EPA VM "Small Commercial Unitary" AC"	4.7	20.0
EPA VM "Large Commercial Unitary AC"	4.3	17.5
CA F-gas Inventory "Non-residential AC 65-135 kBtu/hr"	10.0	56.0
CA F-gas Inventory "Non-residential AC >135 kBtu/hr"	**7.0***	**20.0***

***Values used in this study**

3.4. Manufacturing and EOL Emissions

The manufacture of new refrigerant results indirectly in GHG emissions. For this analysis, values compiled by the International Institute of Refrigeration's (IIR) LCCP working group are used: **for each kg of R-410A manufactured, 10.7 kg CO_2-eq. are emitted and for each kg of R-32, 7.2 kg CO_2-eq.** are emitted (IIR LCCP Working Group, 2016).

Additionally, the manufacturing of all materials comprising the system also indirectly contribute GHG emissions. The IIR's LCCP handbook compiles emissions values for material manufacturing and EOL recycling of the most common materials: steel, copper, aluminum, and plastics (IIR LCCP Working Group, 2016). These material quantities are not widely available and are variable for different brands (e.g. some manufacturers may use more steel in their enclosures where others use more plastics, some use all-aluminum heat exchangers where other use copper-tube aluminum-fin heat exchangers). Preliminary analyses of 10-ton equipment weighing 300~500kg found material manufacturing emissions to be less than 2,000 kg CO_2-eq., which will be seen later to be less than 2% of lifetime emissions of these systems. **Since this data is lacking and these indirect emissions are trivial when considering a single building, material manufacturing and EOL emissions are excluded from this analysis.**

3.5. Calculation Summary

The calculation of emissions through the lifetime of these systems is described below. The model details and assumptions mentioned above determine the indirect emissions from energy consumption, manufacturing & EOL, and the direct emissions of refrigerant to represent the total GHG emissions the system contributes during its lifetime on a CO_2-equivalent basis. As described in Section 3.4, materials manufacturing and EOL are omitted from this analysis but refrigerant manufacturing is included. Some details such as energy to recover refrigerant and transport and install products, are not included in this analysis because of their relative insignificance for a single system.

$$Lifetime\ Emissions = Energy\ Emissions + Refrigerant\ Emissions + Mfg.\&\ EOL\ Emissions$$
$$Indirect\ Energy\ Emissions = L * \left(AEC * EM_{elec} + AGC * EM_{gas}\right)$$
$$Direct\ Refrigerant\ Emissions = C * GWP * (L * ALR + EOL)$$
$$Indirect\ Mfg.\&\ EOL\ Emissions = \cancel{\sum m_{matl.,i} * EM_{matl.,i}} + C * RFM * (L * ALR + EOL)$$

$Where$:

$L = lifetime, 20\ [years]$

$AEC = Annual\ Electricity\ Consumption\ [MWh]$[‡]

$EM_{elec} = Electricity\ Emissions\ Factor, 189.5\ [\frac{kgCO_2-eq.}{MWh}]$ [‡]

$AGC = Annual\ Gas\ Consumption\ [MBtu]$[‡]

$EM_{gas} = Gas\ Emissions\ Factor, 0.05\ \left[\frac{kg\ CO_2-eq.}{MBtu*}\right]$[‡]

$C = Refrigerant\ Charge\ [kg]$[§]

$GWP = Global\ Warming\ Potential\ \left[\frac{kg\ CO_2-eq.}{kg\ Refrigerant}\right]$[**]

$ALR = Annual\ Leakage\ Rate\ [\%\ of\ nominal\ initial\ charge]$[§]

$EOL = End\ of\ Life\ Leakage\ Rate\ [\%\ of\ nominal\ initial\ charge]$[§]

$m_{matl.i} = Mass\ of\ Material\ i\ [kg]$[††] - *excluded from this analysis*

$EM_{matl.,i} = Manufactring\ Emissions\ of\ Material\ i\ \left[\frac{kg\ CO_2-eq.}{kg\ material}\right]$[††]$-excluded\ from\ this\ analysis$

$RFM = Refrigerant\ Manufacturing\ and\ EOL\ Emissions\ Factor\ \left[\frac{kg\ CO_2-eq.}{kg\ Refrigerant}\right]$ [††]

[‡] Determined from CBECC-Com and Energy Plus simulations
[§] Section 3.3
[**] (Forster et al., 2007), Section 2.2
[††] Section 3.4. Emissions due to material manufacturing and EOL are neglected in this analysis
*Here "MBtu" is used consistent with CBECC-Com's labeling to mean 1,000,000 BTU – this is often otherwise denoted "MMBtu"

3.6. Scenario Simulation Parameters

Several scenarios are presented here to provide meaningful comparisons between VRF equipment using R-410A and conventional RTU systems using R-32. Two locations are examined: Los Angeles (LA) and Fresno. The building in LA nominally requires 10-tons capacity while the building in Fresno requires 12-tons. The same analysis could be expanded statewide without great effort. Two types of conventional RTU systems are considered consisting of: single zone air conditioning (SZAC) with gas heating and single zone heat pumps (SZHP) meeting 2023 minimum efficiency requirements. Four types of VRF systems are included: two are heat pumps and two have heat recovery; within those groups, one is minimum efficiency under Title 24 and another is an efficiency of a typical product (because VRF systems are generally premium products where minimum efficiency versions are not available). Figure 3 demonstrates this by showing the minimum efficiency levels from ASHRAE 90.1, which would be effective in 2023, alongside the IEER values of all VRF systems in the AHRI directory: the majority of systems have much higher efficiencies than the minimum. The details of the modeled systems are summarized in Table 5. VRF charge quantities are based on typical equipment shared by Daikin; since minimum efficiency equipment does not exist, the same charge is used, however if a lower-efficiency VRF were to be manufactured it would likely have smaller heat exchangers, which would reduce charge. For VRF equipment, only EER values are listed because this is the only input used by CBECC-Com for VRF efficiency. The values vary by case because minimum efficiency requirements are different by system size and for heat pump vs heat recovery; the typical efficiency values come from actual product specifications.

Table 5: System Configuration Summary

System	Location	Refrigerant	EER (SEER[‡‡])	COP / AFUE	Charge [kg]
SZAC	LA	R-32	11.2 (14.6)	80% AFUE	19.3
SZAC	Fresno	R-32	11.2 (14.6)	80% AFUE	20.1
SZHP	LA	R-32	11.2 (14.6)	3.4	19.5
SZHP	Fresno	R-32	11.2 (14.6)	3.4	21.5
VRF-HP-min. eff.	LA	R-410A	11.0	3.3	16.7
VRF-HP-min. eff.	Fresno	R-410A	10.6	3.2	18.6
VRF-HP-typ. eff.	LA	R-410A	12.0	3.4	16.7
VRF-HP-typ. eff.	Fresno	R-410A	12.1	3.6	18.6
VRF-HR-min. eff.	LA	R-410A	10.8	3.3	23.1
VRF-HR-min. eff.	Fresno	R-410A	10.4	3.2	23.5
VRF-HR-typ. eff.	LA	R-410A	13.2	3.8	23.1
VRF-HR-typ. eff.	Fresno	R-410A	11.9	3.8	23.5

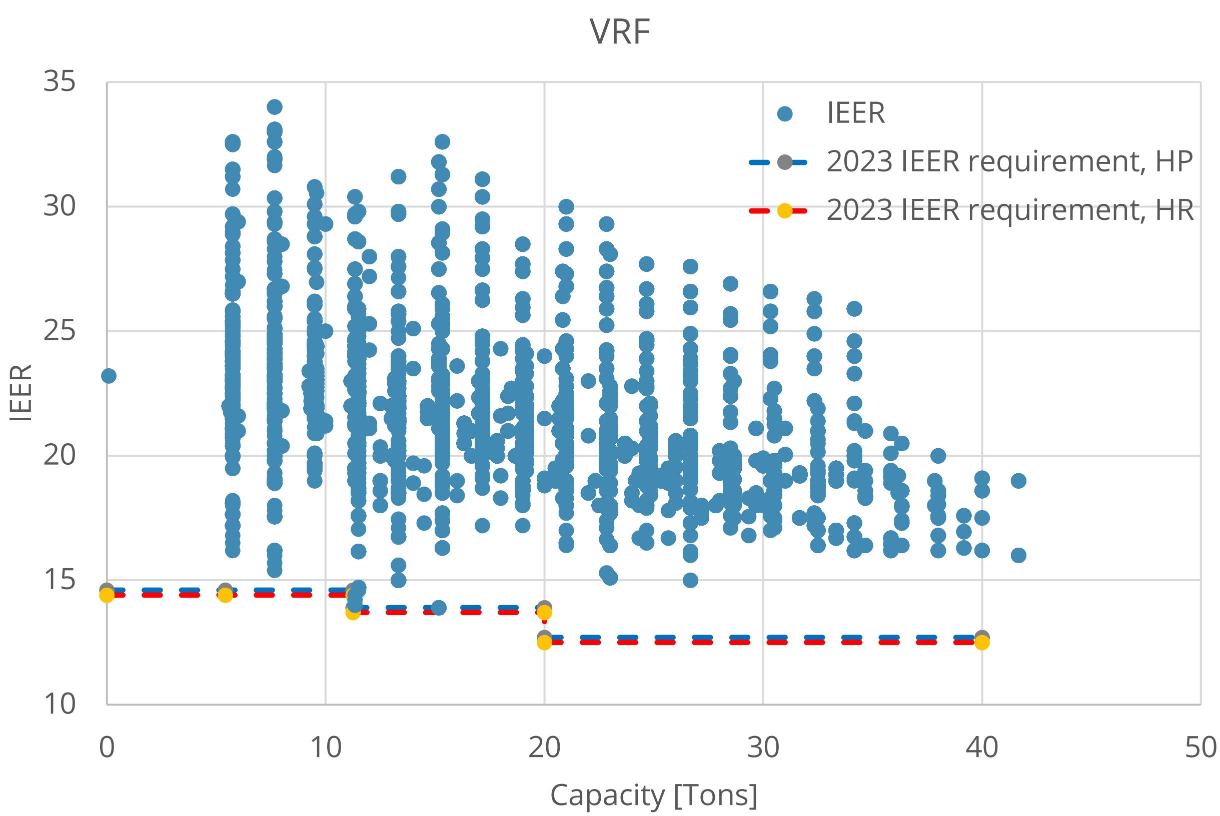


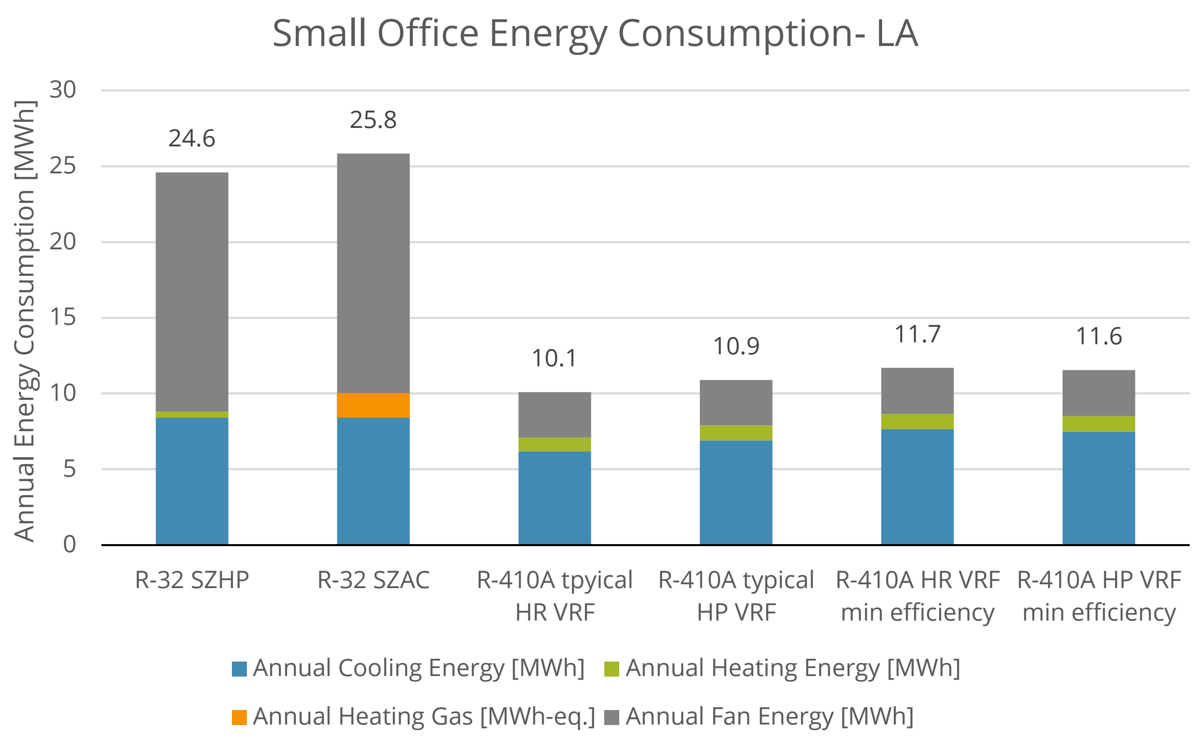
Figure 3: IEER of Typical VRF Systems in AHRI Directory

[‡‡] Minimum efficiency values are reported as IEER, but CBECC-Com requires SEER as a user input

4. Results and Conclusions

The twelve scenarios outlined in Table 5 were simulated in CBECC-Com (and EnergyPlus™ in the VRF cases which required modification to the .idf file). Figure 4 shows the energy consumption from cooling, heating, and fans of the conventional and VRF systems in LA and Fresno. Under these model assumptions, the minimum-efficiency VRF systems consume 43-55% less energy than the conventional systems, and the typical VRF systems consume about 6-14% further less energy than the minimum efficiency ones.

While the VRF systems do exhibit significantly lower cooling energy than the single zone conventional systems, the biggest difference in energy consumption derives from the reduced fan energy. A VRF system should be expected to consume less fan energy because, unlike the conventional system, it is ductless and does not have to overcome nearly as much flow resistance to condition the space. Fan performance values for the VRF systems are based on actual manufacturer specifications, while the RTU fan energy is based on a calculation method from the DOE reference buildings document. Although the RTU fan energy simulated in these models (using the 2.5" W.C. static pressure calculation method) is lower than the default CBECC-Com models, it is possible that other RTU model configurations with more efficient fans or less-restrictive ducts could consume considerably less fan energy. Table 6 summarizes the simulated fan energy for several scenarios to evaluate the validity of this model result. In comparable scenarios, the original RTU reference models configured by CEC and packaged with CBECC-Com predict 24.6 MWh fan energy, while the models used for this study predict 15.8 MWh (36% less). Referring to the original reference building models from DOE (which predate the CBECC-Com models), the simulated fan energy in EnergyPlus™ is just 8.7 MWh. This is because the original models have lower fan runtime than the CBECC-Com models, with the latter running all 7 days per week. The choice to modify the CBECC-Com model to use the 2.5" static pressure calculation was intended to preserve the CEC model intent of longer fan runtime while also aiming to achieve more realistic fan energy values. These values of fan energy are lower than the original CBECC-Com model, though not as low as the DOE reference building model.



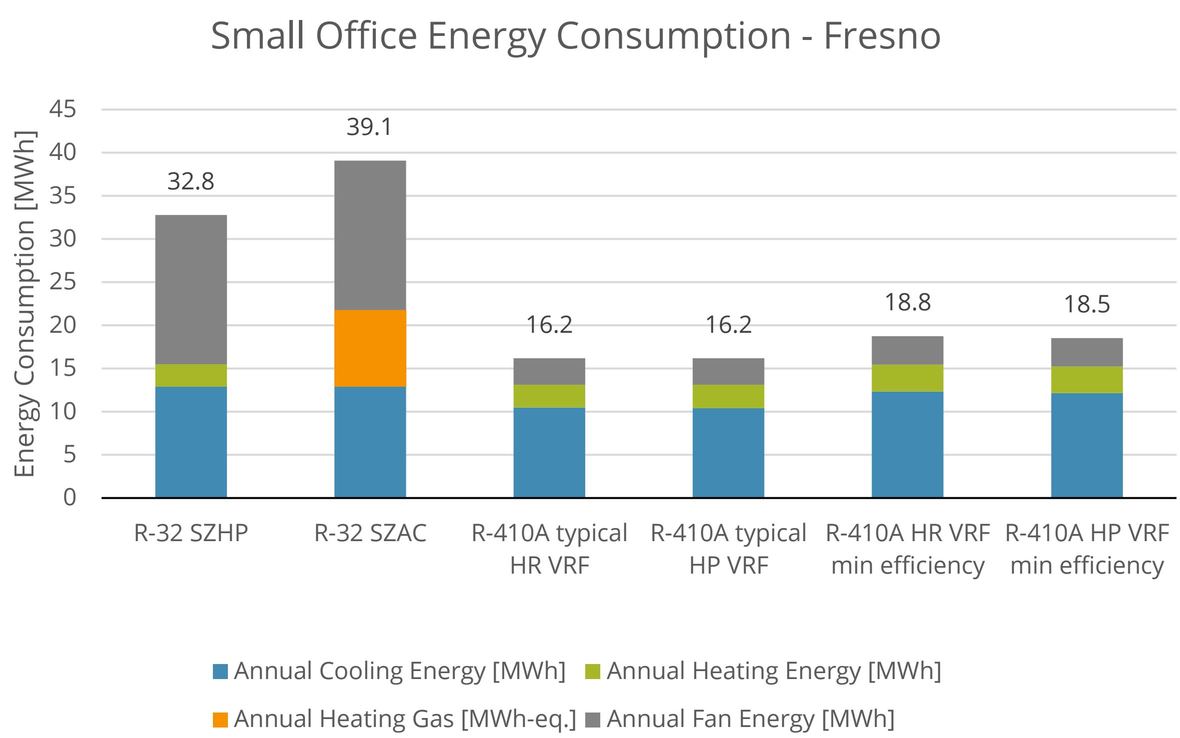


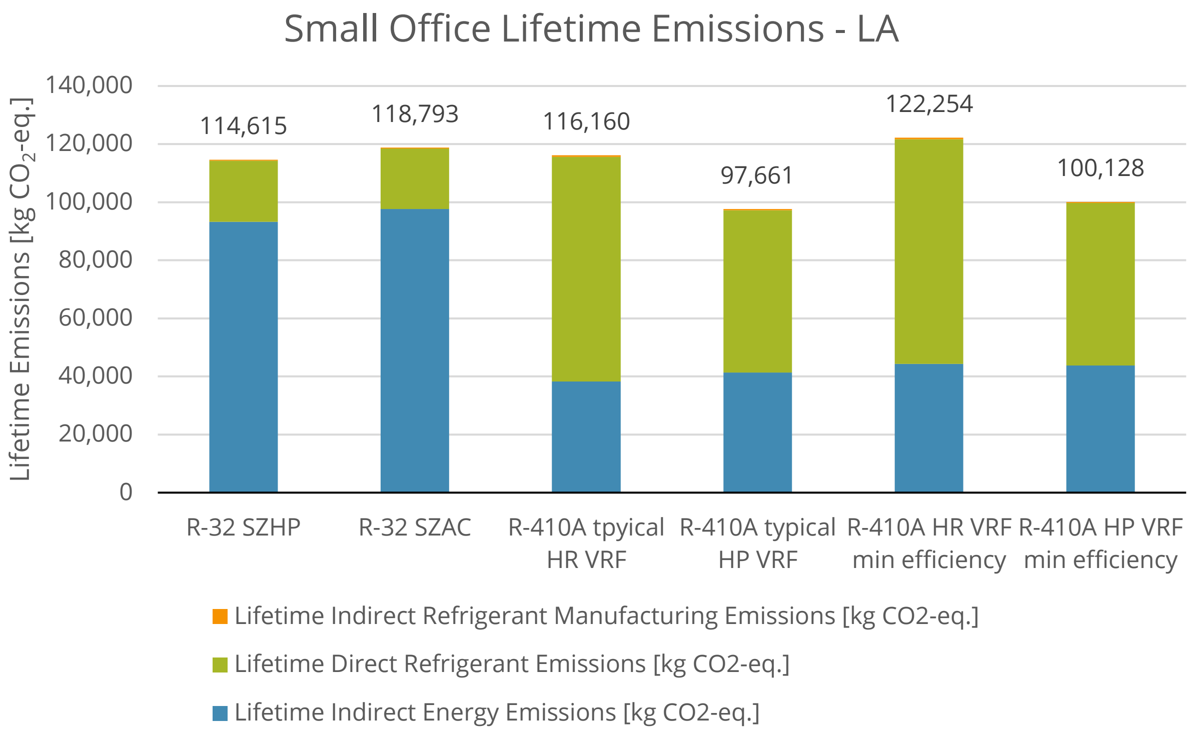
Figure 4: Energy Consumption of all systems in LA (above) and Fresno (below)

Table 6: Simulated RTU Fan Energy Comparison

Model	Simulated RTU Fan Energy [MWh]
Original CEC file (Sacramento)	23.0
Original file updated to match VRF schedules (LA)	24.6
2.5" W.C. static pressure calculation (**Model used throughout this report**) (LA)	**15.8**
Original DOE Reference refbldg_smalloffice_new2004_v1-4_7-2 (LA)	8.7

Figure 5 shows the resultant CO_2-equivalent emissions for these same scenarios. Here it is evident that the lower energy consumption of the VRF equipment results in much lower indirect energy emissions compared to the conventional systems. Contrastingly, the higher GWP of R-410A results in considerably higher direct refrigerant emissions for the VRF than the RTU using R-32. Several important observations are listed below:

- The significantly reduced energy consumption of the R-410A VRF systems leads to comparable, and even fewer lifetime emissions than an RTU system using lower-GWP refrigerant.
    - Specifically, both the typical and minimum-efficiency R-410A VRF HP systems produce less lifetime emissions than both the R-32 RTU systems (heat pump and AC+gas furnace) located in LA and Fresno
- The simulated heat recovery (HR) VRF systems have higher lifetime emissions than their heat pump (HP) counterparts. This occurs because the higher R-410A refrigerant charge of these higher-efficiency systems outweighs the simulated energy savings in these environments.
- In climates requiring more heating/cooling energy (Fresno), the energy savings from the more-efficient VRF system result in a greater reduction in lifetime emissions than in climates where less heating/cooling is required (LA). In other words, in mild climates where HVAC systems are underutilized, their direct refrigerant emissions are more impactful than in environments where HVAC systems are used heavily and their indirect emissions from energy consumption are more significant.
- The direct emissions due to R-410A leakage are considerable and contribute more than half of the lifetime emissions of VRF systems given the assumptions in this study.



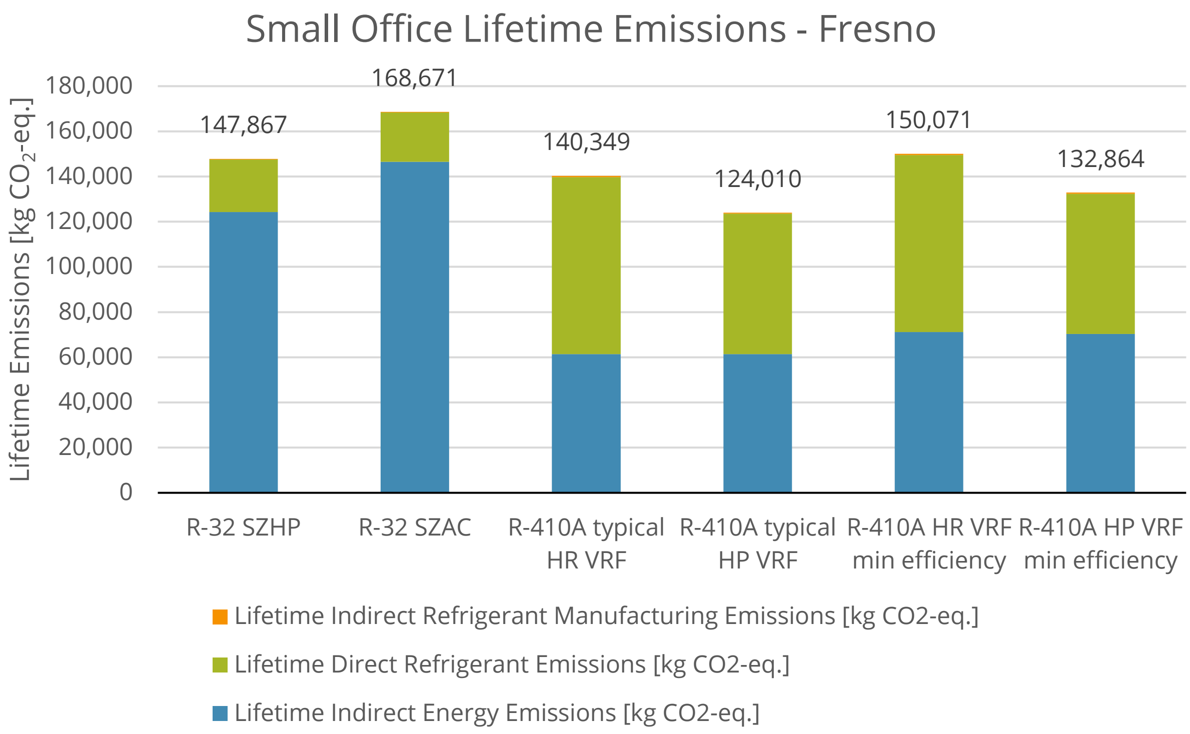


Figure 5: Lifetime CO_2-eq. Emissions in LA (above) and Fresno (below)

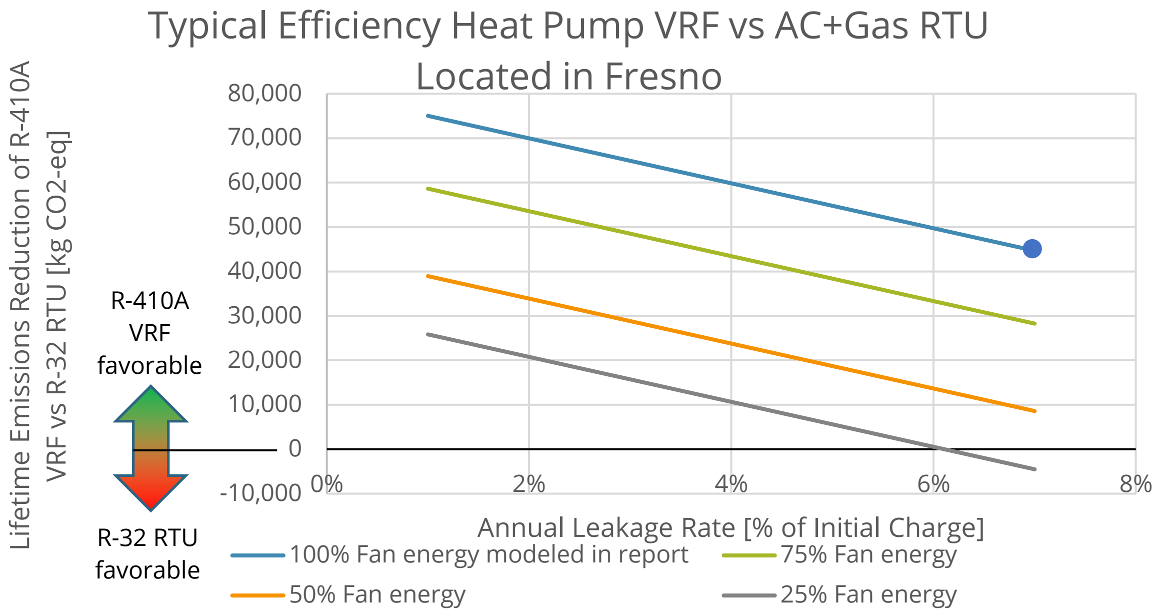
5. Discussion

The work in this report represents one plausible set of assumptions and modeling choices used to estimate the lifetime CO_2-equivalent emissions of both conventional RTU systems using R-32 and VRF systems using R-410A in one building in two California locations. Fundamentally, the results show that the lifetime emissions of these two choices can be similar and even lower for the VRF system using a higher GWP R-410A refrigerant due to its greater energy efficiency. But the results also illustrate the strong influence of modeling assumptions on these conclusions, most notably:

- The assumption of 7% annual refrigerant leakage leads to a determination of a lifetime emissions equivalent to 56-78 metric tons of CO_2 in VRF systems, comprising more than half of their total lifetime emissions. Combined with the 20% EOL loss this assumption means that every VRF system is assumed to lose 160% of its initial charge over its 20-year lifetime. Assuming even a slightly lower value for annual leakage has a considerable impact on the conclusions of such an analysis.
- The high fan energy consumption of the buildings simulated with RTU systems is a major factor that leads to cases where R-32 RTUs have higher lifetime emissions than R-410A VRF systems. This is a consequence of modeling choices discussed in Section 4.

To demonstrate the sensitivity of these results to these assumptions, Figure 6 shows the difference in lifetime emissions for two comparisons when annual leakage rate varies from 1-7% and when RTU fan energy is reduced by 25%, 50%, and 75% from what the modeling in this report predicted. The plot on the top compares a typical efficiency heat pump VRF system located in Fresno against an AC+gas RTU. It illustrates that the R-410A VRF system produces less emissions than the R-32 RTU system in almost all scenarios unless an RTU system could be configured with 75% fan energy savings. If the lower 4.3% annual leakage rate from EPA is assumed, the VRF produces 58 metric tons less CO_2 than the RTU (the 7% leakage rate leads to a smaller 45 ton advantage for the VRF). Even if the RTU fan energy is reduced by 75%, the VRF still emits 9 tons less CO_2 over its lifetime with a 4.3% leak rate.

The lower plot shows the worst-case comparison for R-410A VRF: a heat recovery unit with higher refrigerant charge compared against an R-32 RTU heat pump in LA, a location where energy consumption is moderate and contributes proportionally less to lifetime emissions. The top line shows that under the modeling assumptions in this report, the VRF system emits about 7.6 metric tons of CO_2 more than the RTU during its lifetime. In scenarios where an RTU can be configured to consume considerably less fan energy, the RTU becomes even more favorable to the VRF. If a 4.3% annual leakage rate is assumed, the VRF emits 11 tons less CO_2 than the RTU. But if the RTU power were cut in half, the VRF would emit 22 tons more CO_2.



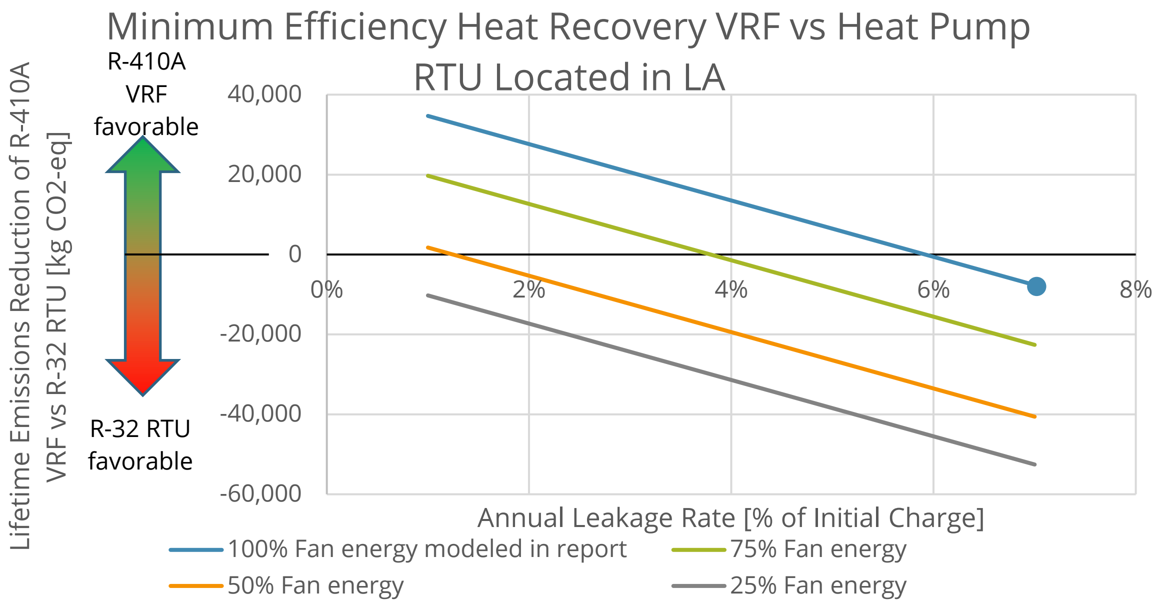


Figure 6: Lifetime Emissions Comparison Subject to Changes in Modeling Assumptions (blue dot denotes results under main assumptions of this study)

The way in which systems are modeled also greatly influence the results of such studies. For example, CBECC-Com uses an EnergyPlus™ model of VRF systems based on a publication from the Florida Solar Energy Center (Raustad et al., 2013). This model includes default polynomial coefficients for performance based on manufacturer specifications of one VRF system. This modeling was intended to be a framework used to fit new coefficients for each piece of equipment, however CBECC-Com retains these coefficients for all VRF systems. Other authors have proposed curve fits based on larger datasets representative of multiple manufacturers; these models tend to suggest lower energy consumption for VRF systems than the model currently used in CBECC-Com (NORESCO, 2016). Discussion of the merits of these models is outside the scope of this report, but it is worthy of note that the selection of models used to represent VRF systems can have significant impact on the results.

Numerous other factors can influence these results. Different building types in other locations will have different performance characteristics. Changing building controls, occupancy, loads, and ventilation will all impact energy consumption. Fortunately, all of these considerations can be addressed through building energy modeling to compare the impacts of HVAC system choice. With careful consideration of modeling choices and assumptions, this type of approach can be expanded to wider state-level or national analyses to make informed decisions about policies, refrigerant selection, HVAC system design, and equipment selection. The most important takeaway from this work is the observation that both direct refrigerant emissions and indirect emissions from energy consumption comprise significant portions of the environmental impacts of HVAC systems and a holistic approach that considers both is necessary to make logical and effective decisions.

6. Conclusions

Just as numerous past publications on LCCP have demonstrated, the consideration of both indirect emissions from energy consumption and direct emissions from refrigerant leakage are critical in characterizing the environmental impacts of HVAC equipment. To minimize these impacts, HVAC equipment must evolve to use lower-GWP fluids, minimize leaks, and also maximize energy efficiency. This report set out to answer one main question: if VRF systems cannot operate with the lower-GWP A2L refrigerant R-32 in the near term due to building regulations, what would the environmental impacts of continued R-410A use be? The building modeling presented here using CBECC-Com demonstrated considerable energy savings for VRF systems over RTUs for this scenario. Combined with assumptions about refrigerant leakage, these results showed that in many of these cases VRF equipment's reduced energy consumption outweighed the negative impact of direct refrigerant emissions and resulted in lower lifetime CO_2-equivalent emissions than RTU systems. The findings are also highly dependent on model details and assumptions. The benefit of VRF is more clear-cut when comparing against an RTU with gas furnace located in Fresno than against a heat pump that is utilized less in LA, for example. Results will certainly vary in different locations, buildings, and when modeling different equipment, but these existing tools provide the ability to evaluate these impacts. When building codes and equipment adapt in the near future to enable the use of lower-GWP A2L refrigerants with VRF equipment, the total lifetime emissions of such systems will be substantially lower than both the R-410A VRF and R-32 RTU systems considered in this report.

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8. Appendix A: Example VRF Fan Specs

Daikin 2-ton cassette: FXFQ24TVJU

https://www.daikinac.com/content/assets/DOC/SubmittalDataSheets/VRV-IU/FXFQ24TVJU.pdf

80W indoor unit power / 777 cfm = 0.103 W/cfm

Submittal Data Sheet

2.0-Ton Round Flow Sensing Cassette

FXFQ24TVJU

PERFORMANCE			
Indoor Unit Model No.	FXFQ24TVJU	Indoor Unit Name:	2.0-Ton Round Flow Sensing Cassette
Type:	Cassette	Rated Cooling Conditions:	Indoor (°F DB/WB): 80 / 67 Ambient (°F DB/WB): 95 / 75
Rated Cooling Capacity (Btu/hr):	23,000	Rated Heating Conditions:	Indoor (°F DB/WB): 70 / 60 Ambient (°F DB/WB): 47 / 43
Sensible Capacity (Btu/hr):	20,000	Rated Piping Length(ft):	
Cooling Input Power (kW):	0.080	Rated Height Separation (ft):	
Rated Heating Capacity (Btu/hr):	27,000		
Heating Input Power (kW):	0.08		

INDOOR UNIT DETAILS			
Power Supply (V/Hz/Ph):	208-230 / 60 / 1	Airflow Rate (HH/H/L) (CFM):	777/618/477
Power Supply Connections:	L1, L2, Ground	Moisture Removal (Gal/hr):	
Min. Circuit Amps MCA (A):	0.7	Gas Pipe Connection (inch):	5/8
Max Overcurrent Protection (MOP) (A):	15	Liquid Pipe Connection (inch):	3/8
Dimensions (HxWxD) (in):	9-11/16 x 33-1/16 x 33-1/16	Condensate Connection (inch):	1-1/4
Net Weight (lb):	51	Sound Pressure (H/L) (dBA):	32/28
Ext. Static Pressure (Rated/Max) (inWg):	/	Sound Power Level (dBA):	

Mitsubishi 2-ton cassette:
http://www.mitsubishitechinfo.ca/sites/default/files/SB_PLFY-EP48NEMU-E_201901.pdf

120W indoor unit power / 1,236 cfm = 0.097 W/cfm

CITY MULTI Model: **PLFY-EP48NEMU-E**

Job Name:		
Schedule Reference:		Site:

PLFY-EP48NEMU-E

SPECIFICATIONS:

Capacity*			Power Input (kW)	Current input (A)
Cooling	Btu/h	48,000	0.11	1.01
Heating	Btu/h	54,000	0.11	0.96

* Cooling / Heating capacity indicated at the maximum value at operation under the following conditions:
Cooling | Indoor : 80° F (27° C) DB / 67° F (19° C) WB
Cooling | Outdoor : 95° F (35° C) DB
Heating | Indoor : 70° F (21° C) DB
Heating | Outdoor : 47° F (8° C) DB / 43° F (6°C) WB

Electrical * * All electrical work shall comply with National (CEC) and local codes and regulations.		
Electrical Power Requirements	1-phase, 208 / 230V, 60Hz	
Minimum Circuit Ampacity (MCA)	A	1.27 / 1.27
Maximum Fuse Size	A	15

External Dimensions		Unit	Grille
Height	In.(mm)	11-3/4 (298)	1-9/16 (40)
Width	In.(mm)	33-1/8 (840)	37-3/8 (950)
Depth	In.(mm)	33-1/8 (840)	37-3/8 (950)
Net Weight	Lbs.(kg)	55 (25)	11 (5)

External Finish Unit (Grille)	Galvanized steel sheet (MUNSELL (6.4Y 8.9/0.4))
Coil Type	Cross Fin (Aluminum Plate Fin and Copper Tube)

Fan		
Type x Quanity (Drive)	Tubro fan x 1 (Direct drive)	
Airflow rate (Low - Mid1 - Mid2 - High)	CFM	777 - 954 - 1,095 - 1,236
External Static Pressure	In. WG (Pa)	0.00 (0)
Motor Type (Output kW)	DC motor (0.120)	

Air Filter	Polypropylene Honeycomb (Long life filter, anti-bacterial type)

Refrigerant Piping Diameter		
Liquid (High Pressure)	In.(mm)	3/8 (9.52) Flared
Gas (Low Pressure)	In.(mm)	5/8 (15.88) Flared
Field Drain Pipe Size (O.D.)	In.(mm)	1-1/4 (32)

Sound Data (Low - Mid - High) (measured in anechoic room)		
Sound Pressure Level	dB(A)	36 - 39 - 42 - 45

GENERAL FEATURES

- Dual set point functionality (*1)
- New 3D i-see sensor built-in (*2)
- New stylish and square design - easier to intall
- New Draft Save Mode
- Lightweight, low-profile compact design
- Cutomizable wide airflow pattern with adjustable vane control through unit controller
- Auto wave airflow in heating mode - independent cycling of horizontal and vertical vane positions for even heat distribution
- Bult-in condensate lift mechanism; lifts to 33-1/2 in.
- Ventilation air intake supported
- Four-speed
- Auto fan
- IT Terminal plug

Note: Mitsubishi Electric (MESCA) supports the use of only MESCA supplied and approved accessories for proper functioning of the unit(s). Use of non-MESCA supported accessories will affect warranty coverage.

ACCESSORIES:

- ☐ Grille.................................. (Required ordered separately) PLP-40EAEU
- ☐ External Heating Adapter PAC-YU25HT
- ☐ Multi-funcation casement PAC-SJ41TM-E
- ☐ Fresh-air intake Duct Flange PAC-SH65OF-E
- ☐ High effciency filterelement (MERV 10) PAC-SH59KF-E
- ☐ 4-way Cassette Signal Receiver PAR-SR3LA-E
- ☐ Simple Remote Controller PAC-YT53CRAU
- ☐ Smart ME Controller PAR-U01MEDU
- ☐ Wireless Remote Controller PAR-FL32MA
- ☐ Wired MA Remote Controller (*2) PAR-32MAA

Notes:
- *1 All components of the system must be compatible. For more details on system compatibility, please refer to Technical Bulletin 100-151 avaiable on our website.
- *2 Unit comes standard with built-in 3D i-see sensor. 3D i-see sensor functions only available with PAR-32MAA wired remote controller.
- Ventilation air to be introduced independent of or in series with VRF indoor units. Please refer to local codes for the required ventilation rates specific to the application.