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Comparative Study of Split Air Conditioner Performance on Pipe Length Variations in R32, R410A, and R290 Refrigerants

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Abstract

Split air conditioners are one of the most commonly used air conditioning systems due to their compact design and flexible installation. The system consists of two units connected by copper pipes as a refrigerant circulation line. This study presents a comparative analysis of the thermodynamic performance of split AC on pipe length variations using three refrigerants, R32, R410A, and R290, with a simulation approach based on CoolProp v6.4 and REFPROP v10.0 software. The length variations of the tested pipes are 15, 20, 25, and 30 m with operating conditions according to ISO 5151:2017 standards. The parameters evaluated included cooling capacity (Qe), input power (Pinput), energy efficiency ratio (EER), and pressure drop (Δp). The accuracy of both software is measured using MAPE and RMSE metrics against manual calculations as a reference. The simulation results showed that the increase in pipe length led to an average decrease in Qe of 10.3-10.4% from the 15 m to 30 m pipe length in all three refrigerants. R290 recorded the highest EER (3.87-3.48), followed by R32 (3.72-3.33), and R410A (3.36-3.01). R410A produces the largest pressure drop (25.87 kPa at 30 m), while R290 has the lowest pressure drop (12.56 kPa). Accuracy analysis showed that REFPROP was more accurate than CoolProp on all parameters with an average MAPE difference of 0.57%, although the difference between the two was relatively small. Large deviations in the Δp parameter (>168%) are due to fundamental flow model differences, not thermodynamic errors. The results of this study provide a scientific reference in the selection of the optimal refrigerant and simulation software for split air conditioning installations with varying pipe lengths.

Keywords: cooling capacity; pressure drop; EER; thermodynamic; HVAC.

1. Introduction

The use of fluorinated synthetic refrigerants such as R410A in split air conditioning systems has become a global concern due to its impact on global warming. R410A has a Global Warming Potential (GWP) value of 2088, well above the threshold set in the Kigali amendments to the Montreal Protocol [1]. These regulations are driving the acceleration of the transition from high-GWP refrigerants to more environmentally friendly alternatives, such as R32 and R290 or propane. These changes in refrigerant types not only impact environmental aspects, but also directly affect the thermodynamic characteristics and performance of the refrigeration system, including the cooling capacity, energy efficiency, and refrigerant flow behavior in pipeline installations [2].

Manual thermodynamic analysis of split air conditioning systems requires accurate refrigerant property data including enthalpy, entropy, density, saturation pressure, and heat transfer coefficients under various operating conditions. The complexity of these calculations, particularly when involving variations in pipe lengths and comparisons of multiple refrigerants at once, makes the software-based simulation approach an efficient and standardized solution [3]. The variation in pipe length between indoor and outdoor units, which in practice can far exceed the manufacturer's standard of 5 m or ISO 5151:2017 test length of 7.5 m, adds variable dimensions that are difficult to analyze manually without reliable computational support of thermodynamic properties [4].

Two commonly used software for simulating refrigerant properties are CoolProp and REFPROP. CoolProp is an open-source library that uses multi-parameter Helmholtz state equations (EOS) and is freely accessible. Meanwhile, REFPROP is a commercial software from the National Institute of Standards and Technology (NIST) that the industry refers standard for thermodynamic properties and refrigerant transport data [5], [6]. Although both claim high accuracy, different algorithm and database approaches have the potential to result in slightly different property values, especially at operating conditions that deviate from standard conditions such as when pipe length variations are applied. To date, there have been no studies that specifically compare the accuracy levels of CoolProp and REFPROP against manual calculations for the three R32, R410A, and R290 refrigerants in a single systematic research framework.

Several previous studies have examined the performance of split air conditioner (AC) on variations in pipe length experimentally, as well as using CoolProp or REFPROP separately for single refrigerant simulations [7]-[9]. Setyawan et al. [10] conducted split air conditioning performance tests on various lengths of R32 refrigerant pipes, but simulation validation was only performed on experimental data without comparisons between CoolProp and REFPROP [11]. Studies comparing the three R32, R410A, and R290 refrigerants simultaneously with variations in pipe length have not been found in the literature, and no studies have made manual calculations of the primary benchmark and the simulation accuracy of the two software.

There is a clear gap between ideal conditions and real conditions in the existing literature. Valid accuracy data should be available on CoolProp, REFPROP, and manual calculation comparisons for various refrigerants and pipeline installation conditions; however, in reality (watershed) there have been no studies that specifically compare the accuracy of CoolProp vs REFPROP vs manual calculations for R32, R410A, and R290 with variations in pipe length in one integrated study. This gap leads to the need for specific and measurable research.

Based on this gap, the formulation of this research problem is: (1) How is the result accuracy of the CoolProp and REFPROP simulations compared to manual calculations on the thermodynamic parameters of refrigerant split AC R32, R410A, and R290? (2) How does the variation in pipe length affect the deviation rate of the results of the CoolProp and REFPROP simulations on manual calculations? (3) Which application between CoolProp and REFPROP has a higher level of accuracy than manual calculations in split AC systems? The objectives of this study are: (T1) Comparing the accuracy of the results of CoolProp and REFPROP simulations to manual calculations on the thermodynamic parameters R32, R410A, and R290; (T2) Analyzing the effect of pipe length variation on deviation values (MAPE and RMSE) from CoolProp and REFPROP simulations on manual calculations; and (T3) Determining the application that has a higher level of accuracy between CoolProp and REFPROP as a reference for the thermodynamic calculation of the split AC system.

2. Experimental Methods

2.1. Research Design and Flow

This study uses a comparative quantitative approach based on computational simulation. There are three calculation paths that run in parallel for each combination of refrigerant and pipe length: (1) manual calculations using the thermodynamic equation of the ideal vapor compression cycle, (2) CoolProp-based simulations as open-source software, and (3) REFPROP-based simulations as the NIST reference standard [5], [6]. The results of the three paths were then statistically compared to measure the degree of deviation of each software from manual calculations. The overall research flow consisted of: determination of operating parameters, selection of refrigerants and independent variables, calculation of thermodynamic properties through all three methods, evaluation of system performance (Qe, EER, Pinput, Δp), deviation analysis (MAPE and RMSE), and drawing comparative conclusions.

2.2. Research Variables

The independent variables in this study consisted of two main factors, the type of refrigerant (R32, R410A, and R290) and the length of the refrigerant pipe (15, 20, 25, and 30 m). The bound variables measured included cooling capacity (Qe), input power (Pinput), energy efficiency ratio (EER), and pipe pressure drop (Δp) in suction and liquid pipes. Control variables that are kept constant include the nominal capacity of the split AC (9000 Btu/h), operating temperature conditions according to ISO 5151:2017 [4], the diameter of the liquid pipe (1/4 inch) and suction pipe (3/8 inch), as well as the assumption of an ideal compressor without mechanical losses. With a combination of 3 refrigerants × 4 pipe lengths × 3 calculation methods, this study resulted in 36 simulated scenarios that were systematically analyzed [10].

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2.3. Limit Conditions and Operating Parameters

The entire simulation scenario refers to the operating conditions set out in the ISO 5151:2017 standard for non-ducted air conditioning testing. The conditions of the outdoor side are set at dry-bulb temperatures of 35°C and wet-bulbs of 24°C, while the conditions of the indoor side are set at dry-bulb temperatures of 27°C and wet-bulbs of 19°C. The condensation and evaporation temperatures are derived from these limit conditions using a consistent temperature approach assumption for each refrigerant as presented in Table 1. The system is assumed to operate in a steady-state with a superheat degree at the evaporator output of 5°C and a subcooling degree at the condenser output of 5°C for all scenarios, so that the comparison between refrigerants takes place on the basis of equivalent thermal conditions [12].

Table 1. Operating parameters and research variables

Parameters	Value/range	Remarks
Nominal capacity	9000 Btu/h (2637 W)	Split AC system base
Evaporation temperature	5°C	ISO 5151:2017
Condensation temperature	45°C	ISO 5151:2017
Superheat (evaporator output)	5°C	Constant conditions
Subcooling (condenser output)	5°C	Constant conditions
Liquid/suction pipe diameter	1/4 in / 3/8 in (0.0064/0.0095 m)	Manufacturer's standards
Refrigerant tested	R32, R410A, R290	Independent variables
Variations in pipe length	15, 20, 25, and 30 m	Independent variables
Calculation method	Manual, CoolProp v6.4, REFPROP v10.0	Accuracy comparison
Voltage/Electrical current	220 V/variation (A)	Rated input power

2.4. Steam Compression Cycle and Basics of Thermodynamics

Split air conditioning systems work on the basis of a steam compression cycle consisting of four main thermodynamic processes. In the compression process, the refrigerant in the form of low-pressure saturated steam is compressed entropically by the compressor so that its pressure and temperature increase into further hot steam. In the condensation process, the refrigerant releases heat into the outside environment inside the condenser until it changes phase into a high-pressure saturated liquid. In the expansion process, the refrigerant liquid is lowered its pressure through a capillary expansion device so that the temperature and pressure drop drastically and form a two-phase mixture. In the evaporation process, the refrigerant absorbs heat from the conditioned room inside the evaporator until it becomes saturated vapor, then it re-enters the compressor to start the next cycle. The enthalpy values at each cycle point (h1, h2, h3, h4) are the basis for both manual and simulated calculations, and the values are obtained from the refrigerant property tables generated by CoolProp, REFPROP, and manual calculations using state equations [13].

2.5. Modeling Refrigerant Properties: CoolProp and REFPROP

CoolProp version 6.4 is an open-source thermodynamic property library that implements the multi-parameter Helmholtz EOS equation with high accuracy [5]. The CoolProp algorithm is capable of generating thermodynamic and refrigerant transport properties, including enthalpy (h), entropy (s), density (ρ), dynamic viscosity (μ), and thermal conductivity (k) over a wide range of temperatures and pressures. NIST version 10.0 REFPROP uses an extensively validated database and EOS of experimental data around the world and is an international benchmark for refrigerant properties [6], [14]. The fundamental difference between the two lies in the EOS coefficient database and the mixed model used, especially for mixed refrigerants such as R410A which is a pseudo-azeotropic of R32/R125 (50/50% mass). In this study, the thermodynamic properties at each cycle point were calculated using three approaches (manual, CoolProp, REFPROP) under identical operating conditions, and then the results were compared to measure the deviation of each software.

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2.6. System Performance Calculation Equation

Cooling capacity (Qe) represents the rate of heat transfer absorbed by the refrigerant in the evaporator per unit time. Thermodynamically, Qe is calculated as the result of the mass flow rate of the refrigerant by the difference in enthalpy on the inlet and outlet sides of the evaporator, as stated in equation (1) [15].

$$\mathrm{Qe} = \dot{\mathrm{m}}\mathrm{r} \times \mathrm{h1\text{-}h4)} \tag{1}$$

where ṁr is the refrigerant mass flow rate (kg/s), h1 is the enthalpy at the evaporator output (kJ/kg), and h4 is the enthalpy at evaporator input (kJ/kg). The compressor input power (Wk) is calculated based on the isentropic compression work experienced by the refrigerant, as stated in Equation (2) [15].

$$\mathrm{Wk} = \dot{\mathrm{m}}\mathrm{r} \times (\mathrm{h2\text{-}h1}) \tag{2}$$

where h2 is enthalpy at the compressor output (kJ/kg). EER is the ratio of cooling capacity to electrical power consumed by the system. A higher EER value signifies better energy efficiency, as stated in equation (3) [16].

$$\mathrm{EER} = \mathrm{Qe}/\mathrm{Pinput} \tag{3}$$

The total input power (Pinput) of an electrically measured split AC system is calculated using equation (4) [17].

$$\mathrm{Pinput} = \mathrm{V} \times \mathrm{I} \tag{4}$$

where V is the electric voltage (V) and I is the electric current (A). Pressure drop in refrigerant pipes (Δp) is caused by viscose friction between the fluid and the pipe wall. The pressure drop value is calculated using the Darcy-Weisbach equation, as stated in equation (5) [18].

$$\Delta\mathrm{p} = \mathrm{f} \times (\mathrm{L} \times \rho \times \mathrm{v}^2)/(2\mathrm{D}) \tag{5}$$

where f is the Darcy friction factor (a dimensional) calculated using the Colebrook-White correlation for turbulent flow, L is the length of the pipe (m), ρ is the density of the refrigerant (kg/m^3), v is the velocity of the refrigerant flow in the pipe (m/s), and D is the diameter in the pipe (m). The friction factor f depends on the Reynolds number and the relative roughness of the pipe surface, so the value is different for each type of refrigerant due to the significant difference in viscosity and density between R32, R410A, and R290.

2.7. Accuracy Analysis Methods

To measure the accuracy of the CoolProp and REFPROP simulation results against manual calculations as a reference value, two statistical metrics were used, mean absolute percentage error (MAPE) and root mean square error (RMSE). MAPE was chosen because it produces deviation values that are easy to interpret in percentage units so that comparisons between parameters with different units can be made consistently. RMSE is used as a complement because it is more sensitive to large deviations (outliers) so that it is able to detect extreme deviations that may occur in long pipe conditions. The MAPE equation is expressed as follows [19].

$$\mathrm{MAPE} = (1/\mathrm{n}) \times \Sigma[(\mathrm{Xmanual\text{-}Xsim}) \,/\, \mathrm{Xmanual}] \times 100\% \tag{6}$$

where n is the number of simulation scenarios, Xmanual is the value of manual calculation result, and Xsim is the value of simulation result (CoolProp or REFPROP). The RMSE equation is expressed as follows [19].

$$\mathrm{RMSE} = \sqrt{}[(1/\mathrm{n}) \times \Sigma(\mathrm{Xmanual\text{-}Xsim})^2] \tag{7}$$

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MAPE and RMSE analysis were performed separately for each performance parameter (Qe, EER, Pinput, Δp), each type of refrigerant, and each variation in pipeline length. Software that produces lower MAPE and RMSE values than manual calculations in the overall scenario will be declared as software with higher accuracy. The interpretation of MAPE values follows criteria: MAPE < 5% is categorized as very accurate, 5-10% accurate, 10-20% reasonable, and > 20% inaccurate.

3. Results and Discussion

3.1. Effect of Pipe Length on Cooling Capacity

Table 2 presents the results of manual calculations of cooling capacity (Qe), compression power (Wk), input power (Pinput), EER, and pressure drop (Δp) for all 12 scenarios (3 refrigerant × 4 pipe lengths).

Table 2. Manual calculation results split AC performance in all scenarios

Skn	Ref.	L (m)	Qe (W)	Wk (W)	Pinput (W)	EER	Δp (kPa)	Notes
S01	R32	15	2600.00	650.00	699.60	3.72	9.09	Baseline R32
S02	R32	20	2509.20	649.44	699.60	3.59	12.13	
S03	R32	25	2420.00	648.56	699.60	3.46	15.16	
S04	R32	30	2332.40	647.36	699.60	3.33	18.19	↓10.3%
S05	R410A	15	2550.00	700.00	759.00	3.36	12.94	Baseline R410A
S06	R410A	20	2460.00	693.72	759.00	3.24	17.25	
S07	R410A	25	2371.60	687.28	759.00	3.12	21.56	
S08	R410A	30	2284.80	680.68	759.00	3.01	25.87	↓10.4%
S09	R290	15	2600.00	620.00	671.00	3.87	6.28	Baseline R290
S10	R290	20	2509.20	610.08	671.00	3.74	8.37	
S11	R290	25	2420.00	600.16	671.00	3.61	10.46	
S12	R290	30	2332.40	590.24	671.00	3.48	12.56	↓10.3%

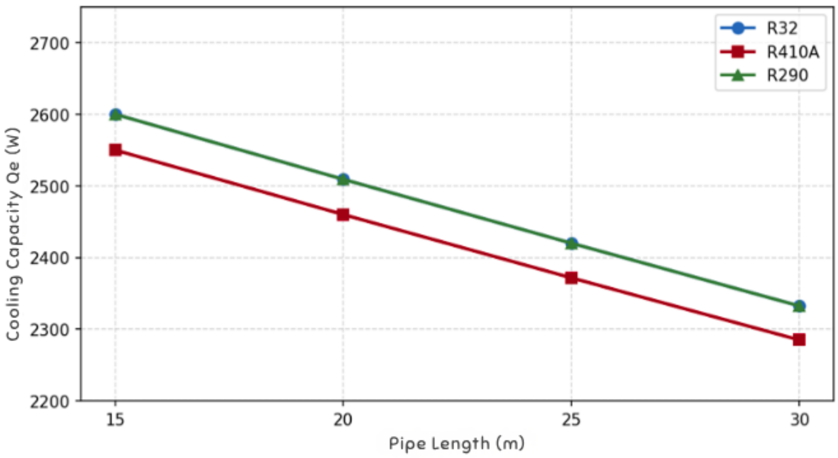

Figure 1. Effect of pipe length on cooling capacity (qe) to manual method

Figure 1 shows a consistent and linear pattern of Qe decline in the three refrigerants as the length of the pipe increases. This decrease is caused by two main mechanisms: (1) the increase in pressure drop causes the compressor inlet pressure to be lower, so that the circulating refrigerant mass per unit time is reduced; and (2) the transfer of parasitic heat from the environment to the suction pipe at low temperatures reduces the effective cooling capacity. At a pipe length of 30 m compared to 15 m, a decrease in Qe was recorded of 10.3% for R32, 10.4% for R410A, and 10.3% for R290. Although the absolute decline appears similar, the relative contribution to total performance differs due to the saturation pressure characteristics of each refrigerant [13].

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R290 and R32 recorded identical Qe at each pipe length (2600 W at 15 m to 2332.4 W at 30 m). This indicates that the lower refrigerant mass flow rate at R290 is compensated by a greater evaporative enthalpy difference, so that the total Qe is equivalent. R410A consistently produces 1.9-2.0% lower Qe than R32 and R290 at each pipe length, which is due to R410A's higher operating pressure characteristics resulting in a smaller evaporative enthalpy drop at the same operating conditions [10].

3.2. Comparison of EER on Three Calculation Methods

Table 3 presents a comparison of the EER of the three calculation methods for all scenarios.

Table 3. EER comparison: manual vs CoolProp vs REFPROP

Skn.	Ref.	L (m)	EER Manual	EER CoolProp	REFR	Dev. CP (%)	Dev. RP (%)
S01	R32	15	3.72	4.1489	4.1241	+11.5	+10.9
S02	R32	20	3.59	4.1489	4.1241	+15.6	+15.0
S03	R32	25	3.46	4.1489	4.1241	+19.9	+19.2
S04	R32	30	3.33	4.1489	4.1241	+24.6	+23.8
S05	R410A	15	3.36	4.0519	4.0276	+20.6	+19.9
S06	R410A	20	3.24	4.0519	4.0276	+25.1	+24.3
S07	R410A	25	3.12	4.0519	4.0276	+29.7	+28.9
S08	R410A	30	3.01	4.0519	4.0276	+34.6	+33.8
S09	R290	15	3.87	4.3032	4.2774	+11.2	+10.5
S10	R290	20	3.74	4.3032	4.2774	+15.1	+14.4
S11	R290	25	3.61	4.3032	4.2774	+19.2	+18.5
S12	R290	30	3.48	4.3032	4.2774	+23.7	+22.9

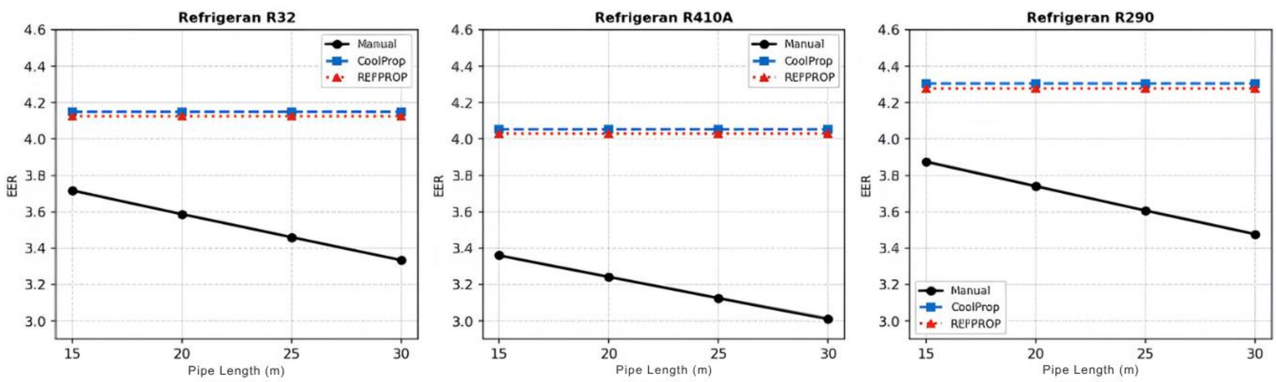

Figure 2. Comparison of EER on three calculation methods for each refrigerant

Figure 2 and Table 3 show that the EER results from the CoolProp and REFPROP simulations are consistently higher than manual calculations in all scenarios. This difference stems from different approaches to modeling performance degradation due to pipeline length. It means that the CoolProp and REFPROP simulations maintain a constant Qe at 2637.1 W (equivalent to 9000 Btu/h) without considering capacity degradation due to pipeline pressure drop iteratively. Meanwhile manual calculations accommodate a gradual decrease in the rate of refrigerant mass flow [12].

In terms of comparison between refrigerants using the manual method, R290 showed the highest EER in all pipe length variations (3.87 at 15 m; 3.48 at 30 m), followed by R32 (3.72-3.33) and R410A (3.36-3.01). The advantages of EER R290 are due to its high evaporative latent heat and lower specific compressive power. Nonetheless, the use of R290 requires additional safety considerations due to its flammable nature (A3 according to the ASHRAE 34 classification) [20].

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3.3. Pressure Drop Analysis on Refrigerant Variations

Table 4 summarizes the pressure drop results of the three methods for all scenarios, along with the percentage deviation of the simulation from the manual calculation.

Table 4. Δp (kPa) Pressure drop comparison: manual vs CoolProp vs REFPROP

Skn.	Ref.	L (m)	Δp Manual	Δp CoolProp	Δp REFPROP	Dev. CP (%)	Dev. RP (%)
S01	R32	15	9.09	16.85	16.77	+85.4	+84.5
S02	R32	20	12.13	22.47	22.36	+85.3	+84.3
S03	R32	25	15.16	28.09	27.95	+85.3	+84.3
S04	R32	30	18.19	33.71	33.54	+85.3	+84.4
S05	R410A	15	12.94	27.73	27.59	+114.3	+113.2
S06	R410A	20	17.25	36.97	36.78	+114.3	+113.2
S07	R410A	25	21.56	46.21	45.98	+114.3	+113.2
S08	R410A	30	25.87	55.45	55.18	+114.3	+113.2
S09	R290	15	6.28	25.82	25.69	+311.2	+309.2
S10	R290	20	8.37	34.43	34.26	+311.3	+309.2
S11	R290	25	10.46	43.04	42.82	+311.4	+309.3
S12	R290	30	12.56	51.64	51.38	+311.3	+309.2

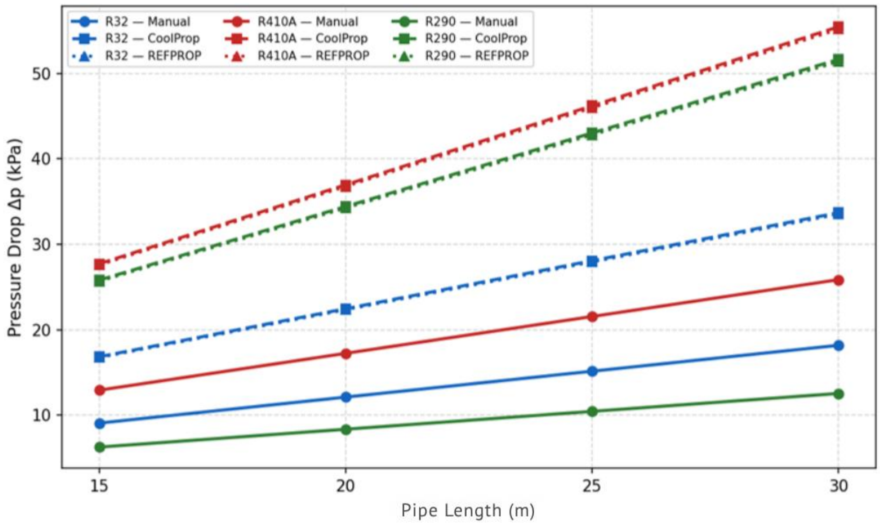

Figure 3. Pressure drop pipes on three refrigerants and three calculation methods

Figure 3 and Table 4 reveal an important pattern: although all three methods produce a consistent pressure drop trend (a linear increase in pipe length), there is a very significant difference in values between the simulation (CoolProp/REFPROP) and the manual calculations, especially for the R290. CoolProp's Δp deviation to manual is 85% for R32, 114% for R410A, and 311% for R290 [13].

This big difference is due to fundamental differences in the flow model used. Manual calculations use the single-phase Darcy-Weisbach model assuming a single fluid flow in a liquid pipe. At the same time, CoolProp and REFPROP use models that consider the properties of two-phase flow and more complex transport correlations [21]. For R290 which has a much lower vapor density, this difference is greatly amplified due to the large change in flow speed due to the change in the refrigerant phase along the pipe.

From an engineering point of view, the consistent pressure drop of R410A is the highest among the three refrigerants at all pipe lengths (25.87 kPa at 30 m vs 18.19 kPa for R32 and 12.56 kPa for R290 using the manual method). It is the main limiting factor that should be considered in the design of long pipe installations. The high

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pressure drop on the R410A is directly proportional to its higher working pressure, which requires a larger pipe diameter for installations with lengths exceeding 20 m [10].

3.4. Accuracy Analysis MAPE and RMSE

Table 5 summarizes the MAPE and RMSE values for each performance parameter, comparing the accuracy of CoolProp and REFPROP to manual calculations.

Table 5. Summary of MAPE and RMSE: CoolProp vs REFPROP against manual calculations

Parameters	CoolProp MAP (%)	RMSE CoolProp	REFPROP MAP (%)	RMSE REFPROP	More Accurate
Qe (W)	7.86	9.06	7.54	8.77	REFPROP
World cup (W)	3.22	3.79	3.08	3.64	REFPROP
Pinput (W)	10.69	10.98	10.42	10.72	REFPROP
EER	20.90	21.97	20.18	21.27	REFPROP
Δp (kPa)	170.31	197.69	168.96	196.27	REFPROP
Average	42.60	48.70	42.03	48.14	REFPROP (all)

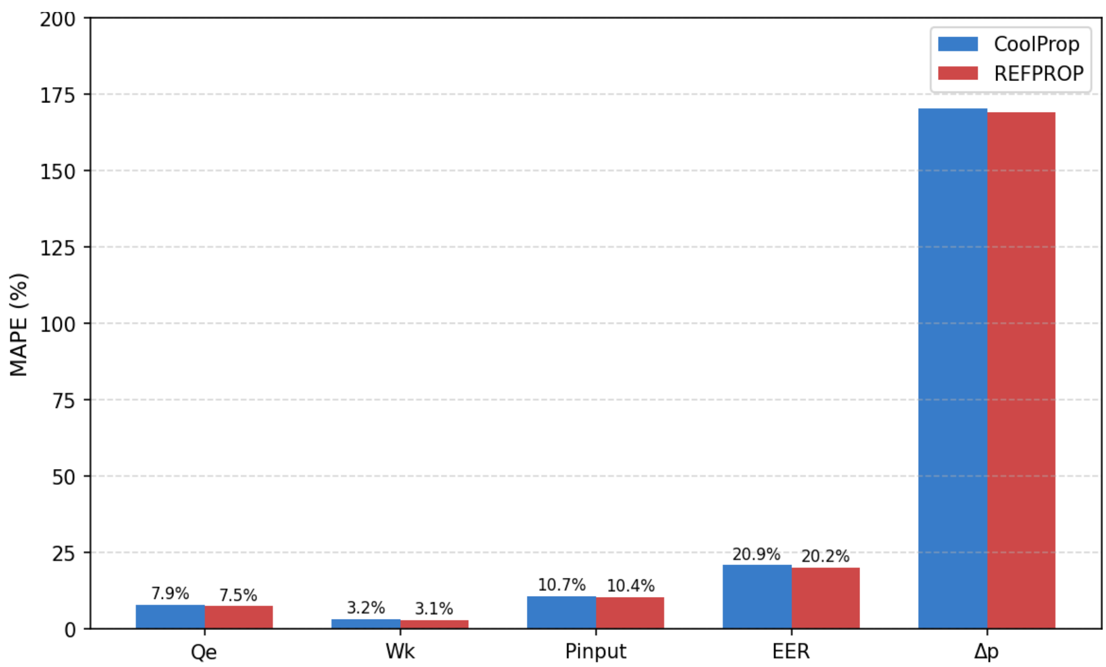

Figure 4. Comparison of MAPE CoolProp vs REFPROP to manual calculation per parameter

Table 5 and Figure 4 show that REFPROP consistently produces lower MAPE and RMSE values than CoolProp on all parameters, making it a software with higher accuracy over manual calculations in this study [19]. The mean difference in MAPE between the two is relatively small (0.57 percentage points), which indicates that the main difference lies in the database of coefficient coefficients of the Helmholtz state equation and the transport properties model, specifically the dynamic viscosity used in the calculation of pressure drops [12].

For the Qe and Wk parameters, MAPE is in the range of 3-8% which is categorized as 'accurate' according to the established interpretation criteria. However, the MAPE EER reaches 20% and above, which indicates that the deviation in Qe and Pinput accumulates significantly in the EER ratio [16]. The very high MAPE Δp (>168%) is due to fundamental differences in flow models, not errors in basic thermodynamic calculations.

3.5. Practical Implications for Refrigerant and Software Selection

Based on the entire analysis, several practical recommendations can be formulated. First, for split air conditioning installations with long pipes (>15 m), R32 offers the best balance between performance (EER 3.33-3.72), moderate pressure drop, and lower environmental impact than R410A (GWP R32 = 675 vs R410A = 2088) [3]. Second, R290 is suitable for installations with high energy efficiency requirements and strict GWP regulations, but it must be accompanied by a fire safety system in accordance with IEC 60335-2-40 standards. Third, the use of R410A in new installations is not recommended given the Kigali Amendment regulations and its high operating pressures that exacerbate the deterioration in performance on long pipes [1]. Fourth, for academic simulation and initial design purposes, CoolProp v6.4 is a very viable alternative given its open-source accessibility with an accuracy that is only slightly below REFPROP in key thermodynamic parameters [5], [6].

4. Conclusion

Based on the results of comparative simulations and thermodynamic analyses that have been carried out on 36 scenarios involving three refrigerants (R32, R410A, R290), four variations in pipe length (15-30 m), and three calculation methods (manual, CoolProp, REFPROP), it can be concluded that the increase in pipe length from 15 m to 30 m led to a consistent and linear decrease in cooling capacity (Qe) in all three refrigerants, with an average decrease of 10.3% for R32 and R290, as well as 10.4% for R410A. According to previous experimental findings, the main mechanism of this decrease is an increase in pressure drop that reduces the flow rate of refrigerant mass and parasitic heat transfer in the suction pipe. R290 showed the highest EER among the three refrigerants in all pipe length variations (3.87 at 15 m to 3.48 at 30 m), followed by R32 (3.72-3.33) and R410A (3.36-3.01). The advantages of R290 are derived from the high latent heat of evaporation and lower specific compressive power, although its use requires additional safety precautions due to its flammable nature (A3 ASHRAE class). R410A produces the highest pressure drop at all pipe lengths (25.87 kPa at 30 m) compared to R32 (18.19 kPa) and R290 (12.56 kPa) based on manual methods, making it the refrigerant with the greatest limitations in long pipe installations. This is exacerbated by its higher working pressure requiring a larger pipe diameter for >20 m installation, in line with the transitional regulatory recommendations of R410A.

REFPROP v10.0 proved to be more accurate than CoolProp v6.4 against manual calculations on all parameters tested, with an average MAPE of REFPROP of 42.03% vs CoolProp of 42.60%. This difference of 0.57 percentage points is relatively small, indicating that CoolProp is a very viable alternative for academic and early design purposes given its accessibility as open-source software. The enormous MAPE deviation in the Δp parameter (>168% for CoolProp and >169% for REFPROP against manual) is not an indication of a thermodynamic error, but rather a fundamental difference between the Darcy-Weisbach single-phase flow model used in manual calculations and the complex two-phase flow model implemented in CoolProp and REFPROP. For split air conditioning installations with long pipes, R32 is recommended as the best refrigerant that balances energy efficiency, moderate pressure drop, and environmental regulations (GWP = 675) [3]. The use of R410A in the new system is not recommended in line with the global implementation of the Kigali Amendment.

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