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ScienceDirect

Energy Procedia 81 (2015) 1212 – 1221

Energy Procedia

69th Conference of the Italian Thermal Engineering Association, ATI2014

Experimental analysis of hydrocarbons as drop-in replacement in household heat pump tumble dryers

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Abstract

Mainly HFCs are nowadays used as refrigerants in commercial household heat pump tumble dryers; it is however crucial to look for long-term alternatives with low environmental impact. Hydrocarbons are considered a suitable alternative for this application. According to their characteristics and based on theoretical analysis, R290 and R441A were selected as substitutes of R407C in a commercial unit. While maintaining exactly the same components, the compressor was changed to comply with the required heating capacity and the expansion device was modulated to meet proper working conditions. Experimental results confirmed that R290 meets the reference energy performances; at the specific conditions, R441A underperforms R407C, due to increased compressor consumption. Technological focus on components is therefore important when refrigerant replacement takes place.

Keywords: Hydrocarbons; Dryer; Heat Pump; Compressor; R290; R441A

1. Introduction

Hydrofluorocarbons are currently used as refrigerants in commercial household heat pump tumble dryers; however regulations are moving towards controlling the use of HFCs. Consequently valid long-term alternatives with low environmental impact have to be searched; stakeholders are analysing different options for a new generation of fluids to be used. In some European countries, as Norway and Denmark, the use of halogenated refrigerants is already controlled, basically by taxing them [1]. The new European F-gas regulation came into force in May 2014 and will be applied from January 2015 [2], introducing new deadlines for HFCs phase out, as detailed in table 1 [2].

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Peer-review under responsibility of the Scientific Committee of ATI 2014
doi:10.1016/j.egypro.2015.12.153

Table 1. F-gas regulation

BANS	DEADLINES
Household refrigerators and freezers using HFCs with GWP>150	2015
Commercial refrigerators and freezers using HFCs with GWP>2500	2020
Commercial refrigerators and freezers using HFCs with GWP>150	2022
Movable air conditioners using HFCs with GWP>150	2020
Split air conditioners charged with less than 3 kg of HFCs with GWP>750	2025

The main novelty and driver for moving towards climate-friendly technologies is the introduction of a phase-down measure which, from 2015, will limit the total amount of HFCs to be used in new appliances; according to available documentation, the use of HFCs will be not only differently considered for each field of application, but also gradually decreased as shown in figure 1 [2,3].

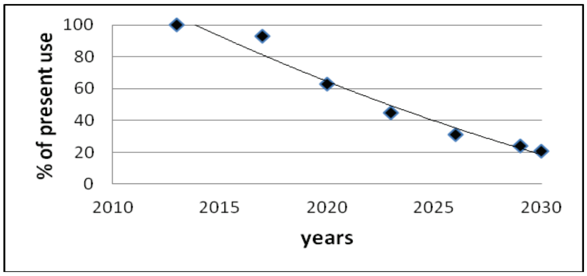

Figure 1: F-gas phase out

R134a and R407C are currently used as refrigerants in the HPTDs; however valid future alternatives need to be identified. In fact, even if small charge-size systems are not subjected to restrictions [2], HFCs availability could decrease and price of these fluids could increase. Two different scenarios studied by the Oko-Institut [4] are reported in figure 2, foreseeing important economic consequences for heat pump tumble dryers manufacturers.

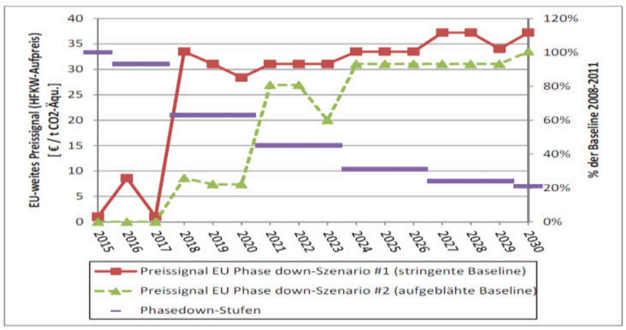

Figure 2: Two possible future scenarios of HFCs price increase [4]

Over the last years, different fluids were proposed as new refrigerants for HPTD; in particular, natural fluids as hydrocarbons and carbon dioxide were kept into account in order to move towards long-term alternatives.

Schmidt et al. [5] compared drying heat pump processes where R134a and CO_2 are used as refrigerants. Simulations showed comparable energy performances if the same compression efficiency is realised; but better compression efficiency is expected with CO_2.

Honma et al. [6] tested a 4.5 kg HPTD charged with CO_2; they obtained an energy saving of 59.2% in comparison with a traditional tumble dryer equipped with HFCs.

Mancini et al. [7] compared a transcritical CO_2 cycle with a subcritical R134a process by theoretical and experimental analyses. They concluded that CO_2 can be a possible substitute of traditional refrigerants for HPTD; negligible decrease in electric power consumption was obtained, with a limited increase in the cycle time, in comparison with a traditional R134a cycle.

Valero et al. [8] investigated propane as a possible substitute for HFCs; a HPTD sized for R134a was analysed. The same machine was charged with propane, after proper compressor installation; tests showed that performances are very close to R134a and that an energy saving around 5% can be obtained.

Novak et al. [9] concluded that differences between R134a, R744 and R290 are not significant in terms of energy performances. They observed that R744 is the most environmental friendly refrigerant in terms of GWP; however, considering TEWI, propane is the most ecological fluid.

Bellomare et al. [10] considered several low-GWP refrigerants for heat pump tumble dryers as possible alternatives to HFCs; exergy analysis was proposed as a tool to evaluate refrigerant drop-in replacement in already existing systems where modifications want to be minimised. They concluded that, according to the developed analysis, propane confirms performance in line with R134a, while isobutane is penalised by pressure drops in heat exchangers.

This work aims at studying a specific HPTD when HCs are used as refrigerants instead of R407C; two different fluids were selected: R290 and R441A, a new hydrocarbons blend. No modifications were carried to the system layout; heat exchangers designed and built for R407C were used also for the other fluids. The compressor was changed in order to obtain the same heating capacity for all considered fluids; a proper compressor displacement was in fact selected for each fluid. Moreover, expansion device was modulated to meet proper working conditions.

Safety assessment related to the use of flammable fluids was out of the scope of this study, which only aimed at assessing performances.

Nomenclature

COP	coefficient of performance	Subscripts	
D	displacement (cm^3)	c	compression
h	enthalpy ($kJ \cdot kg^{-1}$)	comp	compressor
GWP	global warming potential (CO_2 ref)	cond	condenser
HCs	hydrocarbons	evap	evaporator
HFCs	hydroFluoroCarbons	H	heating
HPTDs	heat pump tumble dryers	ref	refrigerant
$\dot{m}$	mass flow rate ($kg \cdot s^{-1}$)	suc	suction
n	rotational speed (min^{-1})	v	volumetric
P	absorbed power (W)	Greek symbols	
Q	capacity (W)	Δ	Difference (-)
TEWI	total equivalent warming impact	η	efficiency (-)
		ρ	Density ($kg \cdot m^{-3}$)

2. Experimental analysis

2.1. Selected hydrocarbons

A Heat Pump Tumble Dryer working with R407C was selected as benchmark; this fluid is a HFCs blend, not flammable and not toxic and consequently classified by ASHRAE as A1 [12]. No charge limits are imposed by regulations in the use of this refrigerant, but it shows 1725 GWP (relative to CO_2, based on a 100-year time horizon) [11]. Moreover R407C is a zeotropic mixture with not-negligible temperature glide during condensation and evaporation process; this can be an advantage when heating processes with high temperature lift take place.

First of all, as already indicated by previous works [8,10], R290 was selected as suitable alternative to R407C; it is a pure fluid, classified by ANSI/ASHRAE Standard 34-2004 as A3 (highly flammable) [12]; a charge limit of 150 g is imposed by regulations [14] when this gas is used in small household appliances as HPTDs. However it shows a very low GWP [13] and doesn't have temperature glide during condensation and evaporation processes.

R441A was also selected as alternative to R407C; it is a Hydrocarbons blend, flammable and classified by ASHRAE as A3 [12]; a charge limit of 150 g is also required by regulations [14] when this gas is used in small household appliances. However it shows a very low GWP and has also a high temperature glide during condensation and evaporation processes. Choosing R441A, potentialities of high glide blends are focused, to investigate benefits as highlighted by Bellomare et al. [10].

The main characteristics of R407C and selected alternative fluids are summarised in Table 2.

Table 2. Selected fluids for experimental activity

FEATURES	R407C	R290	R441A
GWP [13]	1725	8	5
Composition [13]	HFCs blend R32/R125/R134a (0.23/0.25/0.52)	Propane	HCs blend (R170/R290/R600a/R600) (0.031/0.548/0.06/0.361)
ASHRAE Classification [12]	A1	A3	A3
Charge Limits [14]	No limits	<150g	<150g
Temperature Glide [15]	4.0 °C @2.7 MPa	No	14.4°C @1.5 MPa

2.2. Experimental setup

A block scheme of household heat pump tumble dryer is represented in Figure 3 [10].

A heat pump assisted dryer involves a closed-loop air circulation, where the air enters the drum through a fan and is forced through the evaporator, where moisture removal takes place; then the air stream is driven through the condenser, where it is heated up, before re-entering into the drum again. The experimental setup is shown in figure 4.

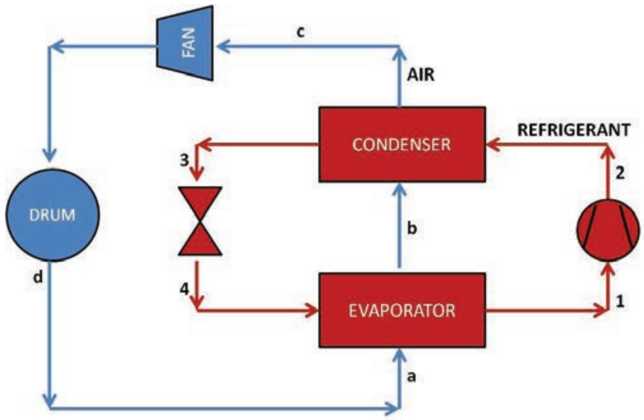

Figure 3: Heat Pump Tumble Dryer block scheme

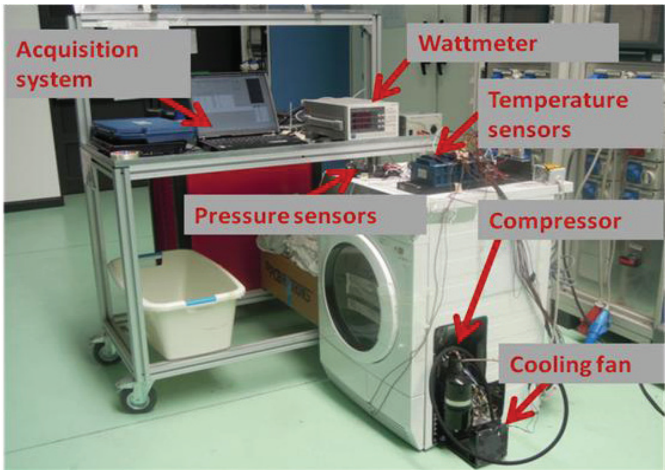

Figure 4: Experimental setup

Refrigerant and air temperatures were measured in all significant points of the system by means of T type thermocouples, as well as ambient temperature; the accuracy of the temperature chain is ±0.3˚C. Refrigerant pressures were acquired at compressor suction and discharge by means of piezoresistive transmitters; with a total accuracy equal to 0.25% of span.

Power input and energy consumption were measured by means of a Yokogawa wattmeter WT230 (0.1%accuracy); compressor consumption was measured independently from other components.

The experimental data were acquired through a LabVIEW 12.0 dedicated software; refrigerant and air properties were calculated and elaborated by means of Refprop 9.0 [15].

A specific commercial heat pump tumble dryer working with R407C was selected as benchmark; tests were firstly carried out in the standard configuration. Afterwards, while maintaining the system layout and heat exchangers, the compressor was substituted in order to achieve the same heating capacity with all considered refrigerants; different displacements were in fact selected for each fluid, as shown in table 3, where displacement values for R290 and R441A are reported with reference to R407C.

Table 3. Compressor displacements selected for each fluid with reference to R407C

FLUID	COMPRESSOR DISPLACEMENT [%]
R290	+28
R441A	+67

Rotary compressors with a nominal speed of 2900 rpm were employed in all tests for each fluid; an external cooling fan was also used in order to balance the system when a specific compressor discharge temperature was achieved.

Rotary compressors for R441A are not currently available in the market; consequently a compressor designed for R290 with proper displacement was selected to perform tests.

Tests were carried out with 9 kg cotton loads composed by sheets, dishcloths and pillowcases [16].

3. Results and discussion

Experimental results are shown in table 4; total energy consumption, drying time, heat pump energy consumption and refrigerant charge are reported with reference to standard machine equipped with R407C.

Table 4. Experimental tests results

Refrigerant	R290	R441A
Total Energy Consumption	+ 6 %	+ 15 %
Drying Time	+ 8 %	+ 7 %
Compressor Energy Consumption	+ 9 %	+ 20 %
Refrigerant Charge	- 50 %	- 50 %

An increment of total energy consumption (heat pump and dryer) was obtained in both cases: +6% with propane and +15% with R441A.

Figure 5 shows measured compressor power input throughout the cycle for the considered refrigerants. R441A compressor was found to require a significantly higher power input. At the same time, a lower air temperature lift when flowing through condenser is found for both R441A and R290. Air temperature lift through condenser (Figure 6) is representative of condenser heating capacity Q_H, as we can assume that air flow rate is maintained when changing from one refrigerant to another in the same heat pump unit.

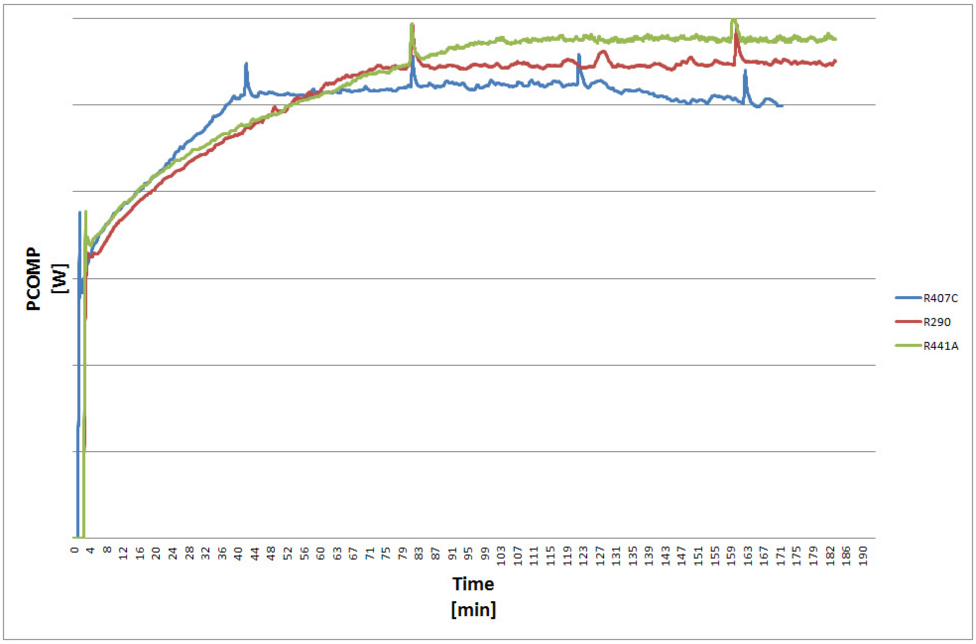

Figure 5: Compressor power input

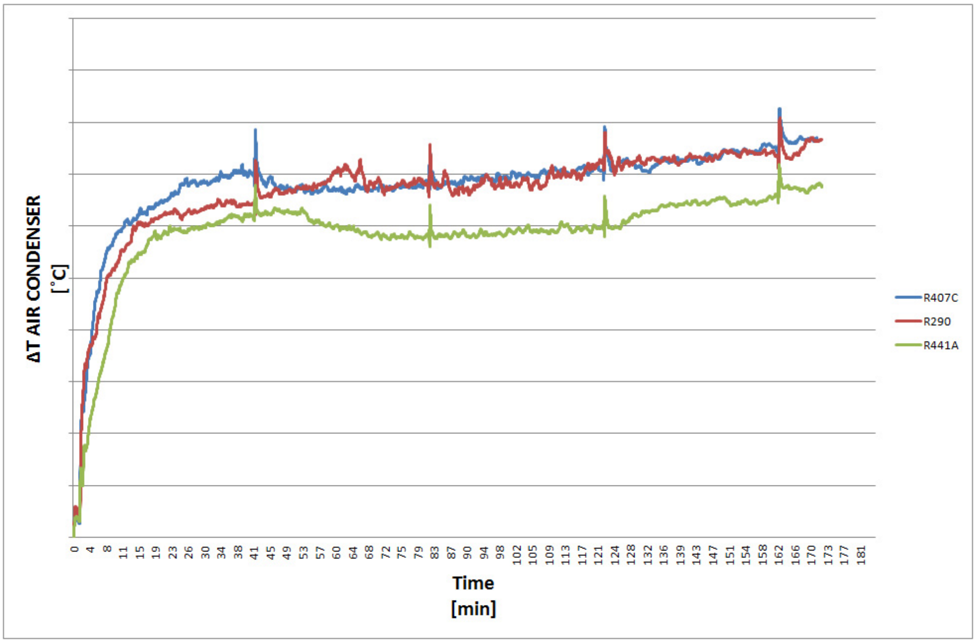

Figure 6: Air temperature lift through the condenser

A previous theoretical analysis was performed by Bellomare et al. [10], aiming at comparing heat pump performances in a tumble dryer when using different refrigerants in the same unit; results are shown in Table 5, considering R407C performance as the reference for the reader's convenience, while in the quoted paper the reference was R134a. COP_H is defined by eq. (1). In the theoretical analysis, performed at typical working conditions for heat pump dryers, when near-steady state conditions are achieved, the same compression efficiency was maintained for all the refrigerants, to evaluate losses sources. R441A was added with respect to [10], still maintaining the same evaluation criteria.

$$COP_H = \frac{Q_H}{P_{comp}} \tag{1}$$

Table 5. Obtained results from simulations [10]

REFRIGERANT	COP_H %	P_{comp} %
R290	-7.2	8.7
R441A	4.0	-3.3

While R290 theoretical performance is in line with experimental results (assuming to maintain COP_H proportion between fluids throughout working conditions range), R441A theoretical prevision is completely mismatching experimental behavior. For this reason, in-depth analysis of experimental data was undertaken. The possible source of inefficiency was identified in the compressor, as theoretical analysis did not account for real compression efficiency.

While R407C compression and volumetric efficiency are available from compressor datasheet, no data were provided for R290 and neither for R441A, whose employment was not even foreseen by manufacturer.

In the case of R407C, refrigerant mass flow rate $\dot{m}_{refr}$,volumetric efficiency η_v and overall compression efficiency η_c were derived, according to eq.s (2), (3) and (4) from catalogue data.

$$\dot{m}_{ref} = \frac{Q_{evap}}{\Delta h_{evap}} \tag{2}$$

$$\eta_v = \frac{\dot{m}_{ref}}{\rho_{suc} D n} \tag{3}$$

$$\eta_c = \frac{\dot{m}_{ref} \Delta h_c}{P_{comp}} \tag{4}$$

where Q_{evap} is the cooling capacity obtained at specific temperature conditions provided by compressor supplier, Δh_{evap} is the difference of refrigerant enthalpy between the evaporator inlet and outlet, ρ_{suc} is the refrigerant density at compressor suction line, D is the compressor displacement and n the compressor rotational speed, Δh_c is the isentropic difference of refrigerant enthalpy through the compressor and P_{comp} is the power absorbed by the compressor.

Consequently, being volumetric efficiency available as a function of compression ratio, condenser heating capacity Q_H was calculated according to equation (5), when experimental data were obtained in order to calculate the difference of enthalpy between condenser inlet and outlet Δh_{cond}.

$$Q_H = \dot{m}_{refr} \Delta h_{cond} \tag{5}$$

In the case of both R290 and R441A, refrigerant mass flow rate was estimated from measured heating capacity, which was evaluated according to eq.s (6) starting from R407C heating capacity.

$$Q_{H\,R290} = Q_{H\,R407C} \frac{\Delta t_{air\,R290}}{\Delta t_{air\,R407C}} ; Q_{H\,R441A} = Q_{H\,R407C} \frac{\Delta t_{air\,R441A}}{\Delta t_{air\,R407C}} \tag{6}$$

Δt_{air} is the temperature difference between air after and before condenser. Being this difference quite high (around 30°C in the calculation near steady-state interval), it was considered satisfactorily representative of heating capacity within the scope of this analysis. Once that Q_H was available for both R290 and R441A, mass flow rates were derived as well as volumetric and compression efficiency.

The evaluation interval was chosen between 60 and 80 minutes after the drying cycle start up, where working conditions were relatively stable. Results are listed in Table 6.

Table 6. Calculated compression efficiency from experimental data

	R407C	**R290**	**R441A**
η_c	0.60	0.57	0.46
η_{vol}	0.81	0.75	0.60

Compression efficiency was found to be 23% lower for R441A than for R407C, while no relevant differences were found for R290, thus justifying the coherence between experimental and numerical results. The reason for R441A bad performance are then assumed to be caused by the compressor, while the cycle itself should provide better performances than R407C when suitable compressor, properly designed for R441A, is available.

Going back to global experimental results (Table 4), an increment of drying time was obtained in both cases; however difference with standard machine is not relevant, in a marketing perspective.

On the other hand, the refrigerant charge is halved by using hydrocarbons, which has positive economic consequences.

4. Conclusions

A commercially available HPTD based on R407C was selected and tested as benchmark. Compressor was then substituted and tests were performed with two HCs: R290 and R441A.

An increment of total energy consumption (heat pump + dryer) was obtained with both new refrigerants; the increment of energy consumption was basically due to the Heat Pump. While the higher energy consumption, though not relevant, was theoretically foreseen with R290, a better result was expected in the case of R441A.

Analysis of experimental data shown that R441A had very low compression efficiency (-23% at near steady-state conditions with respect to R407C compressor), thus affecting the performance of the heat pump and destroying the potential benefit deriving from the use of R441A.

It means that, while it is important moving towards new refrigerant with low environmental impact, it is mandatory having technology support in terms of properly designed components, in order to not deteriorate system performances when a refrigerant drop-in replacement takes place. It is possible to conclude that a rough refrigerant drop-in replacement might lead to higher energy consumption.

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