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Research Article

A novel method for measuring and improving the dehumidification process inside a direct contact condensation unit

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Abstract: This paper reveals the extracted results from the comprehensive experimental research on the direct contact dehumidification process in a washer-dryer machine. Pressurized spray water is injected into the moist air subject to dehumidification. The interplay between the saturated water droplet and process air leads to direct contact condensation on the droplet surface. As a result, latent heat of condensation is released, and saturated water temperature increases. This study investigates the detailed interaction between these two streams and evaluates the effects of temperature distribution on the elapsed time over the moisture removal rates. An effective drying simulation is put into practice through a numerical model. The ongoing drying process is handled without significant loss of dehumidification performance, resulting in a decreased in total energy and water requirements within an operational elapsed run time of 20 minutes. Results show that numerical results are generally in line with the experimental work, which proves the applicability of the Computational Fluid Dynamic (CFD) solutions to this tedious system modeling process.

Keywords: Drying technologies, dehumidification, direct contact condensation, CFD

1. Introduction

Removing water from a substance is frequently and widely used in industrial applications. This process, also called the drying and working principle of the drying systems, totally relies on the combined effect of the heating and condensation process. Between them, condensation has been extensively used in drying systems. Its utilization prevalently dominates the application areas of industrial dryers, cooling towers, and household appliances.

The demand for high-performance and more efficient washer and dryer goods has risen quickly within these two decades, similar to rapid advancements for other home appliances. In recent years, energy efficiency has been one of the most critical design criteria for engineering applications. It still dominates its undisputable prominence over a wide range of industrial appliances. Governments and international organizations strictly regulate the energy requirement of gadgets through imposed laws and predetermined standards. Household appliances have a large market and account for significant energy use; therefore, the set standards strictly regulate manufacturing and sales. Design and production processes receive increasing amounts of attention from manufacturers, as well as demanding end-users. As a result, it becomes of the utmost importance to create an effective drying system to satisfy the demands of different industrial fields.

Three different condensation systems have been commonly employed in the washer-dryer units of household appliances. In an air-cooled condenser drier with a heat exchanger, moist air is passed through the device to cool it and condense the water vapor into a drain pipe. Instead of radiating heat outside, the dryer emits the generated heat inside the immediate area, warming the surrounding environment. The coolant used by the heat exchanger is commonly ambient air. The second, less common washer-dryer set uses a regular heat pump cycle. Unlike air-cooled condensation dryers, which use a passive heat exchanger cooled by outdoor air, heat pump washer dryers use a heat pump unit. The hot and humid air from the tumble is transferred via a heat pump. The hot side reheats the air after the cold side condenses the water vapor into a drainpipe. In addition to avoiding the requirement for ductwork, this method allows the dryer to retain much of its heat inside the appliance rather than expel it outside. However, this system requires a compressor, heat exchangers for evaporation and condensation, and pipelines for cooling fluid. The water-cooled condenser is the third and most favored type of washer-dryer con-

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densing system, whose concurrent heat and mass transfer mechanism will be detailly investigated in this study. This system is mainly based on the direct contact water cooling dehumidification process. While hot and humid air passes through this condensing channel, relatively cold water is injected directly into the moist process air. The steam condenses on the surface of the water drops, extracting inherent moisture from the air.

Direct contact condensation of steam in moist air has been utilized in many prevalent practices because of its wide range of use in industrial applications. Researchers have widely investigated direct contact condensation of steam over sub-cooled water droplets. Variational influences of thermophysical conditions of air, water spray flow rate, and spray configuration on the heat transfer between the process air and subcooled water droplets were investigated by Gumruk & Aktas [1]. Plenty of experimental and numerical studies have been made on getting deep inside this complex heat and mass transfer phenomenon. However, the outcomes of these studies show that there has yet to be accomplished experimental research on this issue compared to the studies relying on numerical aspects, most of which include CFD and analytical calculations. For instance, Gulawani et al. [2] investigated and analyzed the merits of direct contact condensation in steam jet injectors and direct contact feedwater heaters by the CFD simulations. It was shown that numerical results reasonably agree with the experimental data. In addition, a comprehensive analysis was made of the literature's semi-empirical correlations. Montazeri et al. [3] conducted another numerical study based on CFD analysis to assess the performance of evaporative water spray systems designed for indoor and outdoor environment applications. Numerical outcomes revealed that evaporation simulated by the CFD approach could trace the experimental results' tendencies, besides some inherent deficiencies in the simulation modeling. Fletcher et al. [4] presented a numerical method for modeling fundamental flow characteristics in dryers performed over many years at Sydney University. After an exhaustive survey of completed studies, it was decided that it should be given utmost care in three-dimensional transient analyses. The precluded drying and wall interactions should be considered for modeling purposes. They asserted on concluding that these concepts should be added to numerical models by any means to procure clear insights into wall deposition and flow stability issues. Li et al. [5] numerically simulated the direct contact condensation of steam injected into the water pool through the Volume of Fluid (VOF) multiphase flow model and Large Eddy Simulation (LES) of the turbulent model developed for the FLUENT® finite element package. Apanasevich et al. [6] proposed an alternative CFD solution strategy for successful numerical modeling of stratified two-phase flows with the accompanying heat and mass transfer across a moving steam-water interface thoroughly induced by direct contact condensation.

There is also plenty of completed experimental study on this issue that has been carried out, most of which were supported by the numerical background to verify the obtained experimental data. Lekic and Ford [7] theoretically and experimentally examined the direct contact condensation of the moist air on the subcooled spray water. The influences of drop size distribution and characteristic motion of the drops on heat transfer efficiencies were thoroughly discussed. Besides that, a mathematical model based on the acquired experimental data was proposed to obtain transferred heat between mediums. Lee and Tankin [8] examined the characteristic behavior of a subcooled water spray in a water-steam environment. They found out that the length of the sheet region and pressure drop in the liquid sheet play an essential role in the contraction of spray angle in a non-condensable environment, contrary to the ruling concept, which prevails the idea that this is all concerned with the imposed drag on droplets. They developed an empirical-mathematical model to predict the break-up length of the sheet region, which is a function of dimensionless Weber and Jakob numbers. The numerical results obtained from the propounded mathematical model agree with the experimental outcomes. Celata et al. [9] performed comprehensive empirical research on direct contact condensation of saturated steam on subcooled liquid jets. Some of the literature correlations' deficiencies, such as the capability of predicting both the local conditions of the liquid spray and the total heat transfer, urged them to model a new correlation to conquer the inherent drawbacks of these literature mathematical models. Comparing the proposed model, former correlations and experimental data showed that results are within acceptable limits. However, the empirical study only covers the laminar flow regime. Therefore, the applicability of the developed mathematical model over the entire flow regime needs to be investigated. Mayinger and Chavez [10] measured the development of subcooled spray droplets in a saturated vapor using pulsed laser holography. Precise measurements disclosed a strong connection between the saturated vapor pressure and spray geometry.

The direct-contact condensation (DCC) of subsonic steam injected into a subcooled water pool was simulated using the Volume of Fluid (VOF) multiphase flow model and Large Eddy Simulation (LES) turbulent flow model of FLUENT by Li et al. [11]. Similarly, the development and experimental verification of momentum, heat, and mass transfer models in spray drying were studied by Zbicinski [12]. Madejski et al. [13] presented a comprehensive literature survey on the experimental and numerical activities performed on direct contact condensers until now to investigate the ongoing scientific research and lingering past challenges concerning the direct contact condensation process. They comparatively analyzed the developed numerical models simulating the heat and mass transfer mechanism in condensing fluids. They reported some of the most influential experimental studies on the different types of direct contact condensers. Takahashi et al. [14] experimentally and theoretically investigated the condensation heat transfer of saturated steam in a hollow-cone spray of subcooled water. This study obtained a reliable temperature distribution within the spray droplet in both sheet and droplet regions using smaller thermocouples with a fast response. The measured temperature rise in water spray is much higher than predicted by the pure heat conduction models in spray sheets and droplets. This clarifies the heat transfer mechanism during the direct contact condensation of steam on the subcooled water spray.

The literature survey shows that direct contact condensation has been elaborately investigated up to now in both experimental and theoretical aspects by different researchers from different laboratories all around the world. This study aims to add another contribution to the existing literature by examining the difficult phase change process in a condensation unit of a washer & dryer machine in terms of numerical and experimental considerations. Furthermore, the authors aim to open a novel scientific dimension to this hot research field by numerically and experimentally investigating the coupled heat and mass transfer behaviour of the complex phase change phenomena that occurred in the condensation unit of an ordinary household washer & dryer machine. CFD codes developed for simulating flow characteristics of the condensing stream are taken as a benchmark to comprehend the merits of the experimental data, and hourly moisture removal of the condensing unit is assessed by the trial software algorithm developed for numerical simulations.

2. Condenser Type Washer & Dryer System

As mentioned before, drying made to happen by condensation gains frequent usage in daily life and is undeniably an energy-intensive operation. One of the standard applications of drying technology is removing the inherent moisture from porous fabrics such as clothes. Therefore, the dehumidification performance of the washer & dryer machine's condensing unit is analyzed in detail. Suppose it is to give the fundamental concepts of the direct contact condensation process. In that case, the following steps come into practice based on the basics of the thermodynamic principles. Its dry bulb temperature begins to decrease as soon as the hot, humid air interacts with the cooling channel that is kept at a temperature lower than its dew point temperature. Its temperature drops as it travels through the cooling medium, and with constant specific humidity, its relative humidity rises. The cooling process goes on until the air reaches the dew point temperature. At this time, the air's water vapor (moisture) begins to condense, lowering the air's moisture content and decreasing the humidity level. Hence, cooling and dehumidification of the air both occur when the air is cooled below its dew point temperature. Using a separate route, the water vapor that condenses out of the air during this operation is eliminated from the cooling area.

The system has a closed-loop drying cycle and a water-cooling condenser. The dehumidification occurs in the condenser channel, as shown in Figure 1. Components of the drying cycle are briefly expressed in Table 1. During dehumidification, the air is absorbed from the drum through the fan and sucked into the condenser. The moisture content of the air is maintained at a level that is within the limits of the designer's expectations. The dehumidified air is heated utilizing resistance heaters to supply the required heat and increase the air volume. Heated air is blown inside the drum where clothes are kept. Finally, extracted moisture is thrown away from the system by the drain.

3. Description of the Experimental Work

In the experimental studies, mathematical model follow-up tests have been made to simulate the dehumidification mechanism inside the condenser. Dehumidifying operation realized by the direct contact condensation inside the condenser unit has been examined by the two distinct humidity probes placed at the condenser's input zone and above the water spray nozzle at the outlet zone of the condenser, as depicted in Figure 2. Furthermore, the condenser unit where the concurrent heat and mass transfer occur is isolated with an isolation material to maintain reliable experimental conditions.

The area average velocity method calculates the average mass flow rate of the moist air passing through the condenser section where the flow regime condition is close to the laminar flow. The simulation-based CFD model calculates the total volume of system air. According to simulation results, the air volume of the washer group is 0.079 m^3. In comparison, the air volume of the dryer group is 0.0032 m^3, as schematically visualized in Figure 3.

To calculate the humidity absorption capacity of the hot air, the free air volume should be precisely determined when 6 kg of laundry load is placed inside the wash-

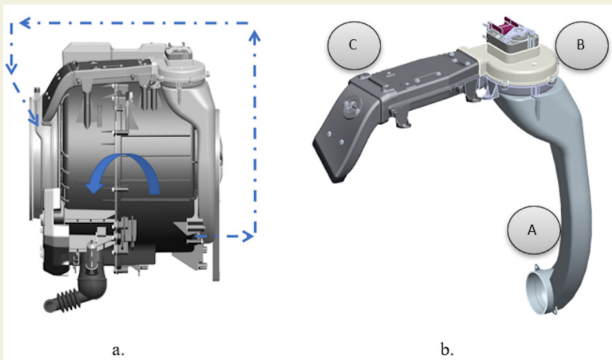

Figure 1. General graphical representation of the a) washer & dryer machine and b) dryer unit

Table 1. Components of the drying unit

Location in Figure 1	Associated section in the system
A	Condenser
B	Fan
C	Heater Unit

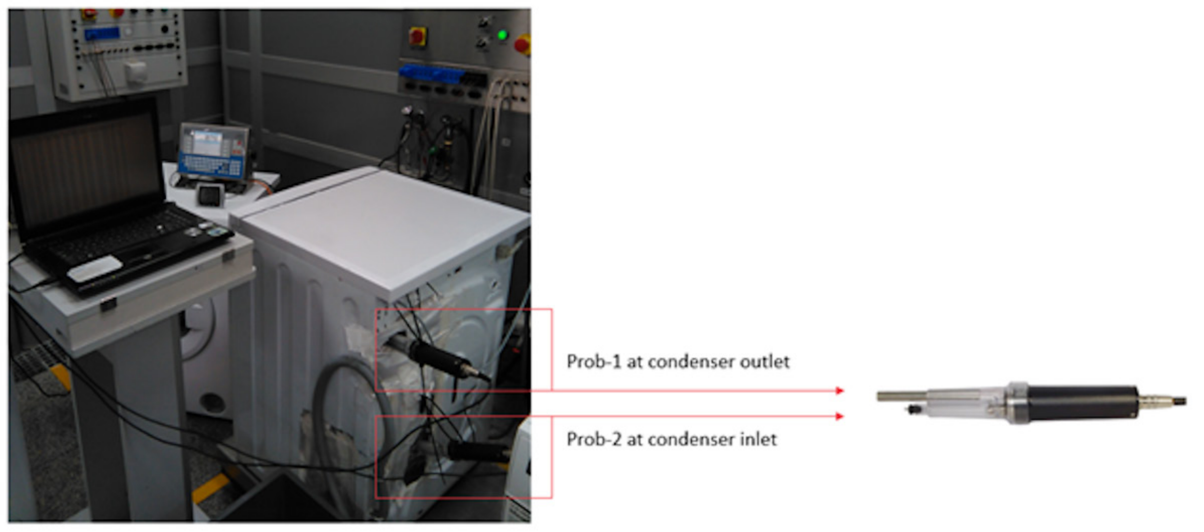

Figure 2. Placement of the humidity probes for simultaneous humidity measurement inside the condenser at the inlet and outlet zone

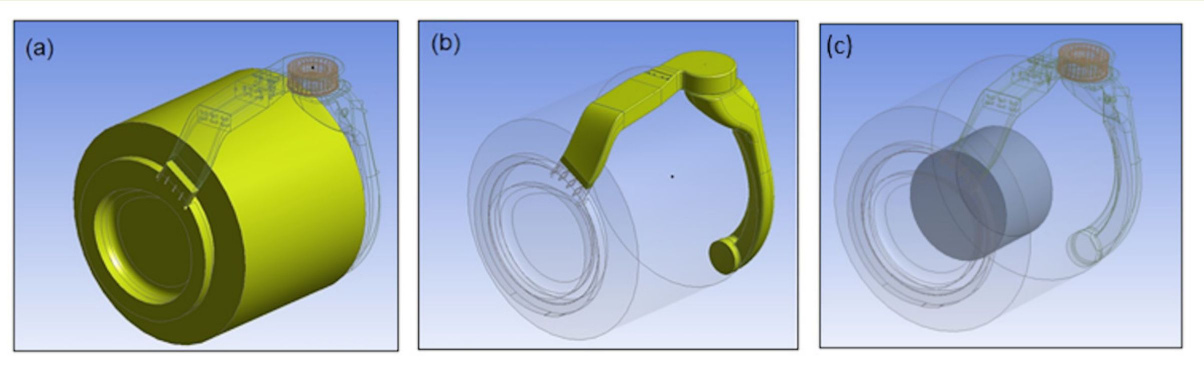

Figure 3. Graphical representation of the a) washer, b) dryer sections and c) laundry volume.

er-dryer machine. The numerical model revealed that the volume of the laundry for 6 kg load is measured as 0.0266 m^3. Moreover, a simple experimental verification based on the water volume replacement method has been performed to validate the numerically obtained laundry volume, as shown in Figure 4. It is observed that the numerical outcomes of the mathematical model are well-matched with the measured volume obtained by the water replacement method.

Figure 5 shows the simulation model of the fluid volume occupying the water-cooled condenser. To obtain accurate flow distribution estimation along the condenser section and calculate the system's moisture removal capacity, the mass flow rate of the air inside the washer-dryer machine, along with the prevalent working conditions, should be precisely defined. For calculating the accurate mass flow rate flowing through the condenser

Figure 4. The volume of laundry measurement by water replacement

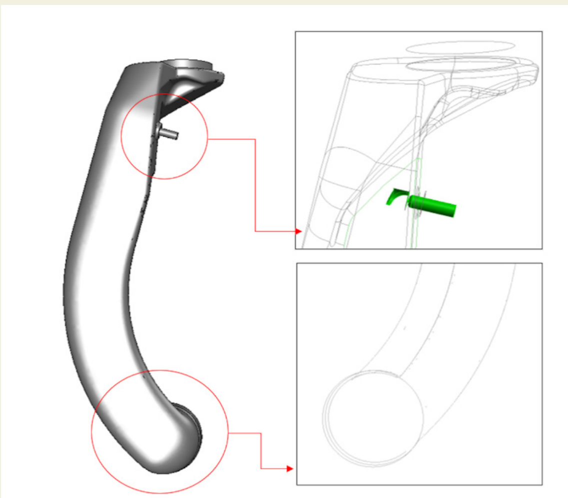

Figure 5. Fluid volume model of the condenser inlet zone and nozzle.

section, the area average velocity method is used by air velocity measurement from 20 different points at the condenser inlet zone through the thermo-anemometer probe shown in Figure 6. Technical specifications of the thermo-anemometer probe are also given in Table 2.

The velocity of the air is measured for 20 different points at the condenser inlet. Exhaustive air velocity measurements have been made for unloaded conditions. They will be repeated due to the upcoming resistances resulting from loaded conditions. Figure 7 and Figure 8 show the procedural steps of the velocity probe measurements at the condenser inlet.

The air stream's velocity does not vary evenly over the cross-section of the channel inlet in actual conditions like this condenser channel application. The forced air flow next to tube walls is retarded by the channel wall-induced friction, which accounts for the more excellent stream velocity rates along the duct's central axis. As shown in Figure 7, a series of velocity measures at sites of equal area should be taken to determine the average overall velocity in vents. The suggested measurement locations for circular ducts are shown in Figure 9. Traverse readings are what these are called.

Using a systematic design of sensing stations over the duct cross-section is advised, as shown in Figure 10. Respective process air velocity values for different measurement positions are tabulated in Table 3.

Table 2. Technical specifications of Thermo-anemometer probe [15].

Parameters	Valid ranges
Flow measuring range	0.2 – 20.0 m/s
Resolution	0.01 m/s
Response time	<1.5 sec.
Measurement accuracy	± (0.2 m/s +2% of meas. val.)
Nominal conditions	22 °C ±2 K - 45 % RH ±10 % RH - 1013 mbar
Temperature compensation	0 to +50 °C
Influence of temperature	±0.3% of measured value /°C at 0.3 to 20 m/s

The area average velocity is the average velocity of air flowing through a cross-sectional area, which is determined by dividing the airflow rate by the area of the duct or opening. The calculation is used in heating, ventilation, and air conditioning (HVAC) to determine the effectiveness of air distribution in space. The ASHRAE (American Society of Heating, Refrigerating, and Air-Conditioning Engineers) Standard 111-2008, "Measurement, Testing, Adjusting, and Balancing of Building HVAC Systems"

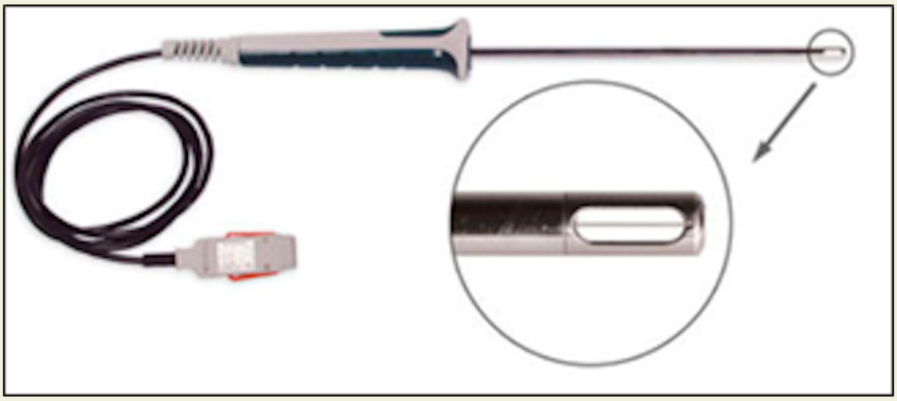

Figure 6. Almemo FVAD 35- TH5 type thermo-anemometer probe

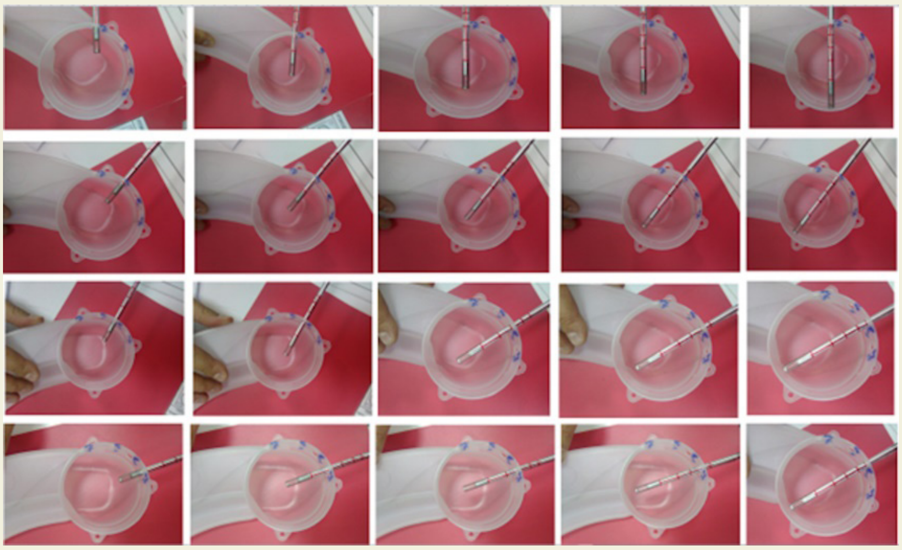

Figure 7. Thermo-anemometer probe measurement placements at the condenser inlet

Figure 8. Employed procedure for thermo-anemometer probe measurement

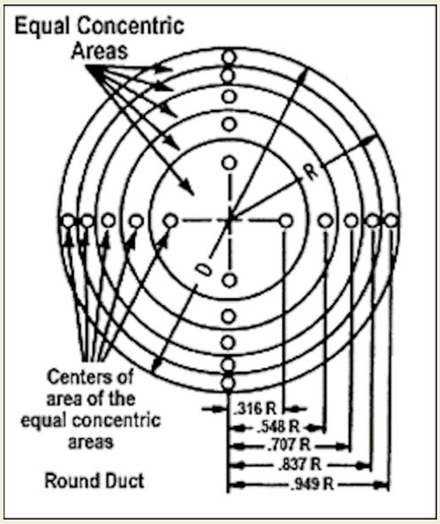

Figure 9. Traverse on round duct area [16]

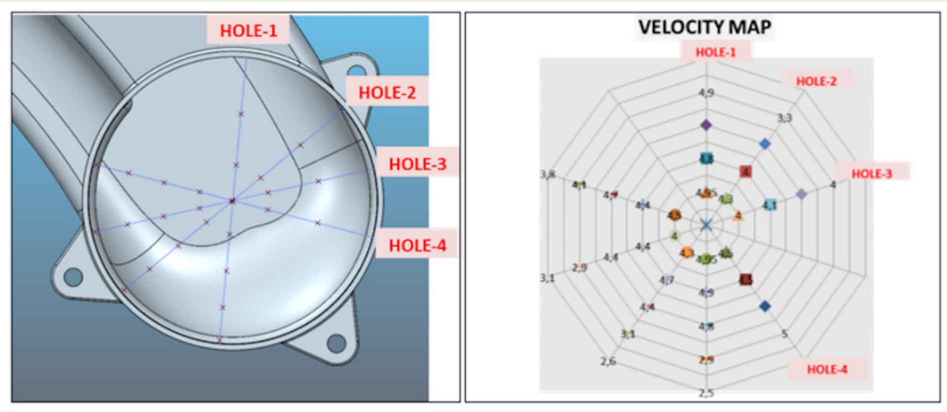

Figure 10. Velocity measurement points at the condenser inlet

Table 3. Measurement points and velocity data for the loaded machine

For D=78.6 mm	Position - 1	Position-2	Position-3	Position-4	Position-5	Position-6	Position-7
Standard * (%) mm	96.8	86.5	67.9	32.1	-	13.5	3.2
Standard-D (%) mm	76.08	67.98	53.36	25.23	-	10.61	2.51
Measurement point **(mm)	77.5	68.5	58.5	48.5	39.5	29.5	15.5
Velocity @							
Hole - 1	2.5	2.9	4.8	4.9	4.95	4.8	4.9
Hole - 2	2.6	3.1	4.4	4.7	4.3	4.0	3.3
Hole - 3	3.1	2.9	4.4	4.4	4.0	4.1	4.0
Hole - 4	3.8	4.1	4.7	4.4	4.5	4.5	5.0
Avg. Velocity (m/s)	3.00	3.25	4.58	4.60	4.44	4.35	4.30

* Traverse reading on round duct area according to the ASHRAE Standard
**Distance between hole inlet and measurement point of the probe (mm)

[15], provides guidelines for measuring and calculating the average area velocity. According to the defined prevailing standard, the average area velocity is calculated using the following equation

$$V = Q / A \tag{1}$$

V is the average area velocity in m/s, Q is the airflow rate in m3, and A is the area of the duct or opening in m2. The average speed at the condenser inlet area was calculated as 4.05 m/s, obtained without a laundry load inside the machine. When velocity measurements in the condenser section were carried out for 6 kg with %50 wetted laun-

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dry to investigate further the influence of laundry load over the mean velocity at the condenser entrance, the average speed at the condenser inlet area was calculated as 4.44 m/s.

4. Numerical Simulations

In the complete design of the washer-dryer machine, one critical component that influences the drying performance is the centrifugal fans that route the moist air to the condenser and then through the heater channel before blowing the drying air to the drum. The experimentally obtained fan speed was measured at 1876 rpm, defined as a ruling boundary condition for the fan work definition. To reduce the computational load of the solver process, solid domains such as the shell of the blower housing and the fan impellers have been defined as adiabatic walls assuming that total heat transfer occurred through these regions has a negligible effect on the numerical model. However, their imposed influences over the flow conditions are considerable. The proper interface models between the fluid and solid domains, including the rotating and stationary domains, have also been specified in the CFD analysis. The fan region is a rotating frame to drive the centrifugal fan flow. To simulate the characteristics of the turbulent flow inside the condenser channel, the standard k–ε turbulence model is utilized because of its comprehensive and dexterous capability.

When the velocity distribution measurements were compared with those obtained from ANSYS Fluent CFD results, which are explicitly visualized in Figure 11, it is observed that the level of the air velocity obtained through numerical simulation is in line with those extracted in the experimental tests.

Table 4. Comparison of theoretical and CFD calculations of mass flow rates

	Theoretical calculation	CFD Results	Units
$V_{air,in}$	3.37	3	m/s
$A_{air,in}$	0.0048	0.0048	m^2
ρ_{air}	1.225	1.225	kg / m^3
Ʊ$_{air,in}$	58.23	51.84	m^3/h
$\dot{m}_{air,in}$	0.020	0.018	kg /s

As shown in Table 4, mass flow average velocity is calculated as 3 m/s with 0.018 kg/s mass flow rate through CFD analysis, as shown in Figure 11.

Velocity distribution along the condenser obtained from CFD and experimental results are compared with the practical work's results. Comparative results between the experimental and theoretical works are mostly well-matched.

5. Condensation Performance Tests

To fully comprehend the influences of the drying program parameters on the overall condensation performance, different test procedures have been performed for varying experimental conditions, tabulated in detail in Table 5. The variations in relative humidities at the inlet and outlet of a condenser are the imminent result of the combination of temperature difference, moisture content, and fluid flow interactions. These values vary depending on the specific operating conditions of the condenser. This is why different operational test combinations have been extensively performed; to observe the changes in the decisive model parameters. Enriching

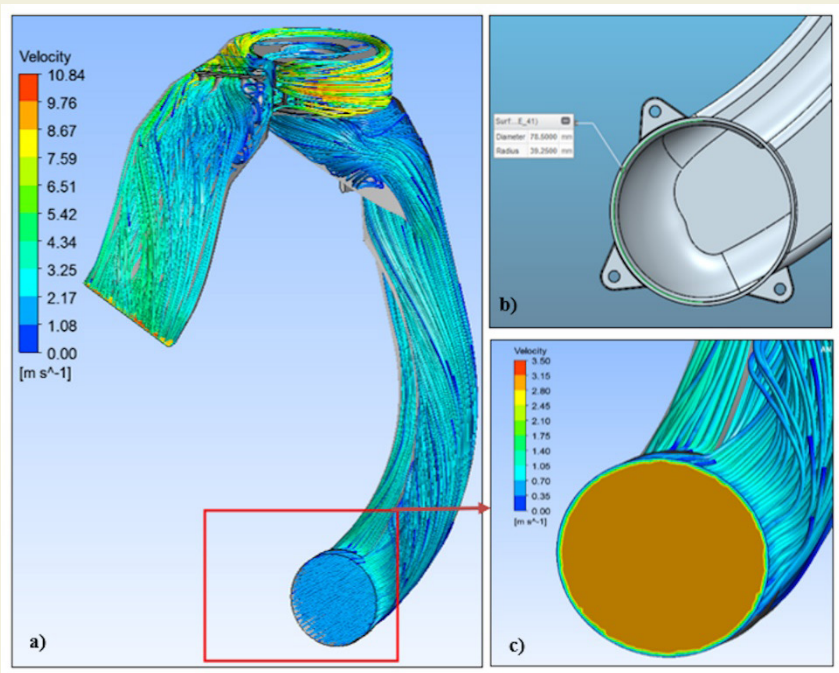

Figure 11. a) Velocity distribution at the condenser channel according to the CFD analysis b) 3D model of the condenser channel inlet, c) Detailed view of the velocity distribution at the inlet.

the combinations of these commanding experimental parameters for various case studies is also possible. Test duration is considered and limited to 1 hour in all operational test conditions. A trial algorithm was developed that only considers the effects of the heater, fan, and valve activations. The primary purpose of developing the trail algorithm is to simplify the evaluation of test samples obtained in the drying software.

Water valve activations at the condenser channel were settled in three conditions; continuously On, continuously Off, and software controlled On & Off. The two-staged heaters activations were defined and controlled by Negative Temperature Coefficient (NTC) sensor, which is necessary to obtain different temperature ranges for different drying levels. In all test conditions, active fan work is considered, and the same fixed flow rates are employed in the system.

To obtain the experimental inputs, absolute humidity, temperature, and pressure change tracking tests were performed during the defined drying program cycle. Thermo-anemometer probes, humidity probes, test panels, and data loaders were used to measure the model parameters. The inlet and outlet water amounts were also tediously measured to verify the employed numerical model. A test machine was placed onto a weighing device to avoid any incorrect measurement, as shown in Figure 2. Two different humidity probes are simultaneously utilized to measure the dry bulb temperature, wet bulb temperature, relative humidity, and total pressure at the condenser dryer inlet and outlet in the one-hour drying program. During the one-hour drying cycle completed in the washer dryer, absolute humidity changes are also considered depending on the temperature and water spray activations.

5.1. Test-1: Spray Water is Continuously ON

In the first test condition, the spray water valve and fan are continuously activated during the drying tests. One of the two heaters is constantly active. At the same time, the other one is controlled with the NTC, a negative temperature sensor responsible for measuring the air temperature. Utilizing NTC and software control, heater activation can be reliably administrated. In this specified test condition, influences spray water activation over the dehumidification process regarding the changes in relative humidity, absolute humidity, and process air temperature values have been experimentally investigated, and measurement plots obtained during the one-hour trial drying program are shown in Figure 12 and Figure 13. The process air in the condenser inlet arrives at its dew point in the early phases of the experiments. In contrast, condenser outlet relative humidity decreases to the lowest point of %55 relative humidity. Then it shows a rapid increase due to the water spray activation intensity. At the end of the drying problem simulation, the outlet's relative humidity rate is around %93. Process air at the condenser inlet shows a rapid increase. Then it becomes a stagnation phase due to the influences of active heaters. In contrast, the operating temperature of process air at the condenser outlet shows a negligible increase during the drying experiments, as shown in Figure 12. The absolute humidity difference between the condenser outlet and inlet increases for the first forty minutes and then falls into a decreasing trend, as shown in Figure 13.

5.2. Test-2: Spray water is continuously OFF

In the second test condition, contrary to the experimental conditions of the first test, spray water is deactivated. At the same time, fans and heaters are still active. Operational requirements considered for this case study aimed to investigate the time-varying inclinations of design variables when spray water is deactivated. This second test condition also analyzed the effects of not using water during dehumidification. Relative humidity, absolute humidity, and temperature measurements are shown in Figure 14 and Figure 15 during the one-hour trial drying program. Process air reaches to dew point within the first two minutes of the drying tests while relative humidity of the process air decreases for the first fifteen minutes; then humidity rates increase and become saturated after forty-five minutes of drying time. Variations of these two operational parameters significantly influence the changes in absolute humidity rates between the condenser inlet and outlet. Sharp decreases follow a considerable increase in the fundamental humidity differences between the condenser outlet and inlet during the drying experiments.

5.3. Test-3: Spray Water Valve Activation is Alternately ON & OFF

The third test condition covers the operational parameters in which the spray water is activated for the first 15 seconds. Then it shifts into deactivation phases for 65

Table 5. Varying experimental conditions for the comparative study

	Test - 1	Test - 2	Test - 3	Test - 4	Test - 5
Duration	1 Hour	1 Hour	1 Hour	1 Hour	1 Hour
Spray Water Activation	Continuously ON	Continuously OFF	Valve activation with the software control	Continuously OFF	Continuously OFF
Heater-1 Activation	Continuously ON	Continuously ON	Continuously ON	Continuously ON	Continuously OFF
Heater-2 Activation	Activated with the control of NTC	Activated with the control of NTC	Activated with the control of NTC	Continuously OFF	Continuously OFF
Fan Activation	Continuously ON	Continuously ON	Continuously ON	Continuously ON	Continuously ON

seconds, which is deliberately considered to rely on the original and know-how-based algorithm. This periodic switch between the water spray activation and deactivation occurs until the experiments end for the assumed test conditions. Valve and pump capacities are the reasons for selecting the correct time intervals. Similar operational conditions have been considered for the remaining design parameters as practiced in the first and second cases. As it is shown in Figure 16 and Figure 17, similar relative humidity and working temperature measurements are made with those obtained for the operational conditions of Test-1 and Test-2.

5.4. Test-4: Spray Water and One Heater are Simultaneously OFF

Experimental results of the first three test conditions reveal that water utilization is essential in dehumidification. However, there should be a trade-off between the amount of water consumption as there would be drastic consequences between using too much water and not using water in the dehumidification process. The fourth test condition considers the activation of only one heater and constant water spray deactivation to observe their variational influences over working temperatures and relative humidity rates. Relative humidity, absolute humidity, and temperature measurements are shown in Figure 18 and Figure 19. The relative humidity rates at the condenser outlet decrease with the elapsed time. In contrast, the operation temperature of the process air increases at the inlet and outlet of the condenser increases, which increases the change of absolute humidities between the condenser inlet and outlet.

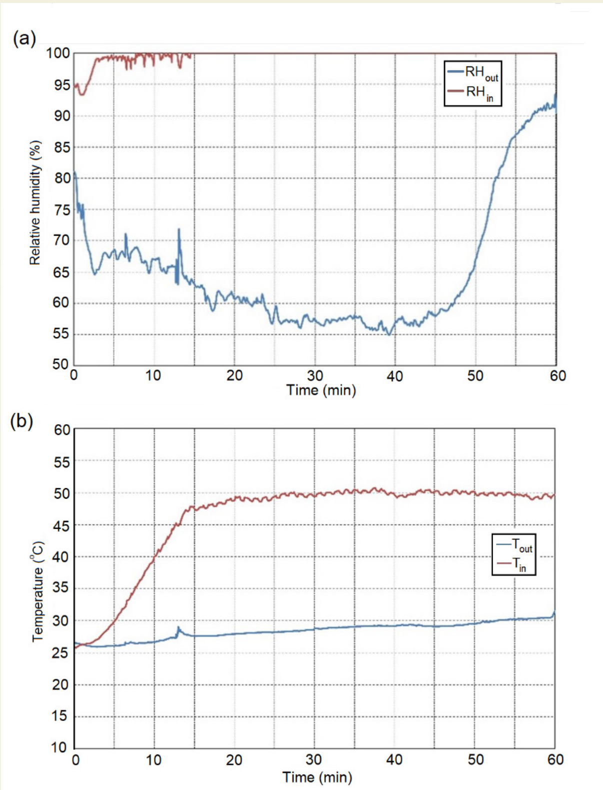

Figure 12. a) Relative Humidity and b) Temperature measurements for Test-1

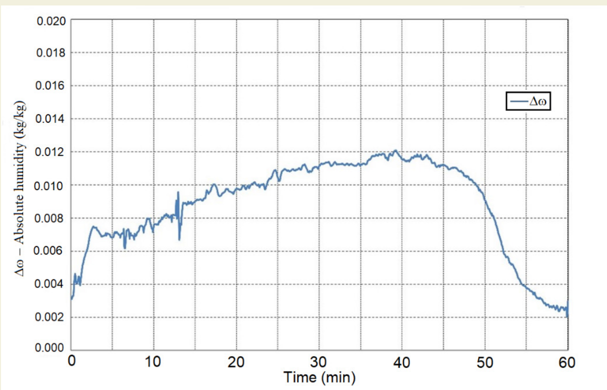

Figure 13. Absolute humidity measurements for Test-1

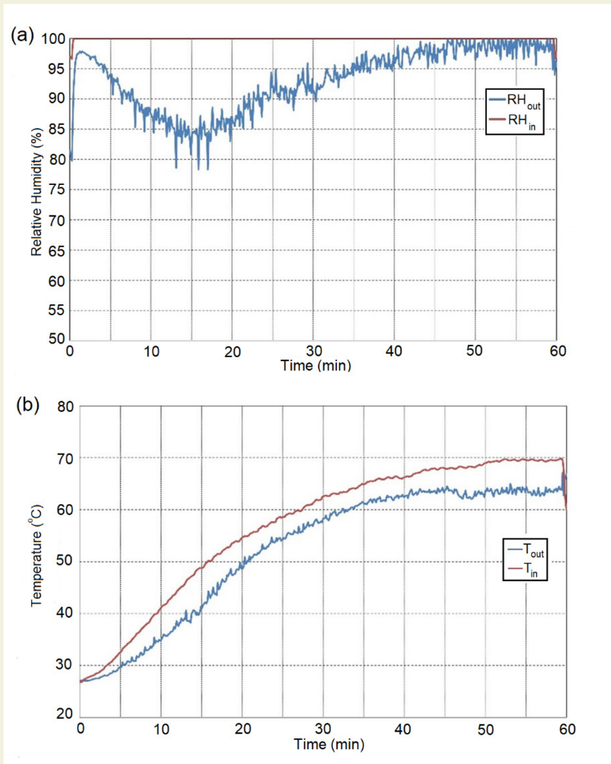

Figure 14. a) Relative Humidity and b) Temperature measurements for Test-2

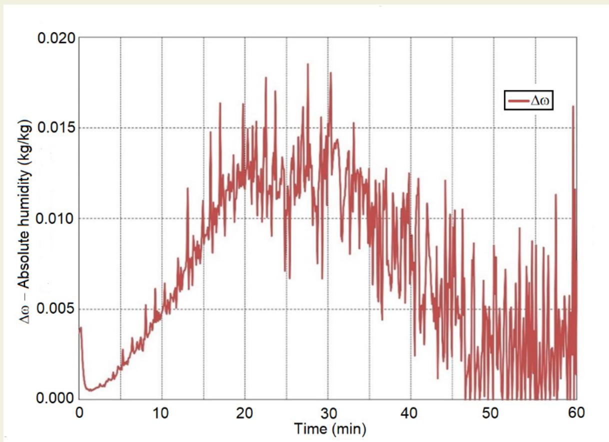

Figure 15. Absolute humidity measurements for Test-2

5.5. Test-5: Spray Water and Two Heaters are Simultaneously OFF

The last test condition aims to observe the fan flow's effect on dehumidification. So, in the fifth test, only the fan is activated while other operational parameters stay deactivated. As shown in Figure 20, relative humidity rates at the condenser inlet slightly increase throughout the elapsed experiment time. The relative humidity of the outlet process air turn into an increasing trend and nearly reaches its dew point with unpredictable fluctuations. The process air temperature decreases at the condenser inlet and outlet in the early phases, followed by stepwise increases during the experimental tests. The absolute humidity at the condenser outlet is higher than that of the condenser inlet for the first 30 minutes of experiment time, as observed in Figure 21.

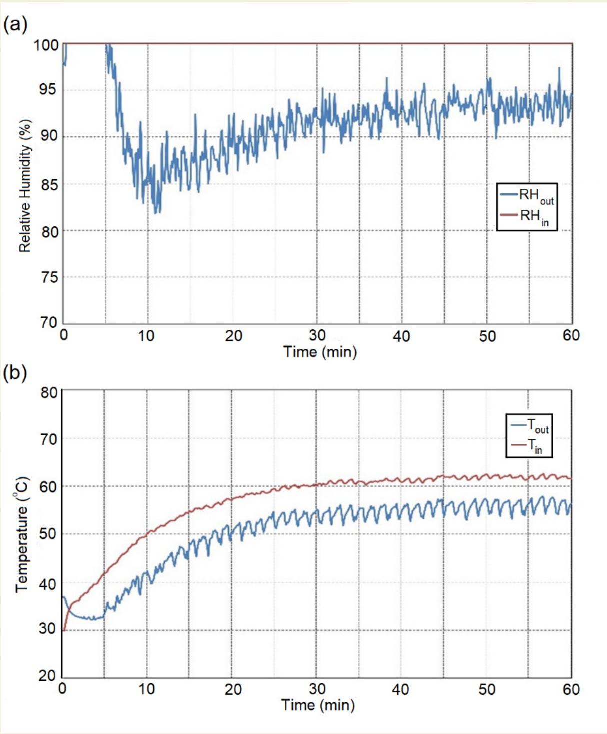

Figure 16. a) Relative Humidity and b) Temperature measurements for Test-3

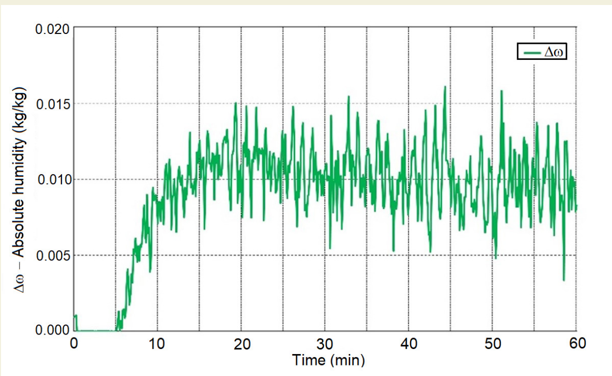

Figure 17. Absolute humidity measurements for Test-3

6. Discussion on the Condensation Test Results

All tests have taken relative humidity measurements just above the spray nozzle. Theoretically, this psychometric process involves removing water from the air as the air temperature falls below the dew point temperature and reaches %100 relative humidity. Extracted test data is obtained through heat transfer and mass transfer mechanisms to take the clothes' moisture during the drying circuit. During the one-hour drying cycle, absolute humidity level changes are experimentally extracted according to the shifted activations of the two responsible heaters and the water spray. The schematic of the test setup and measurement points are summarized and sche-

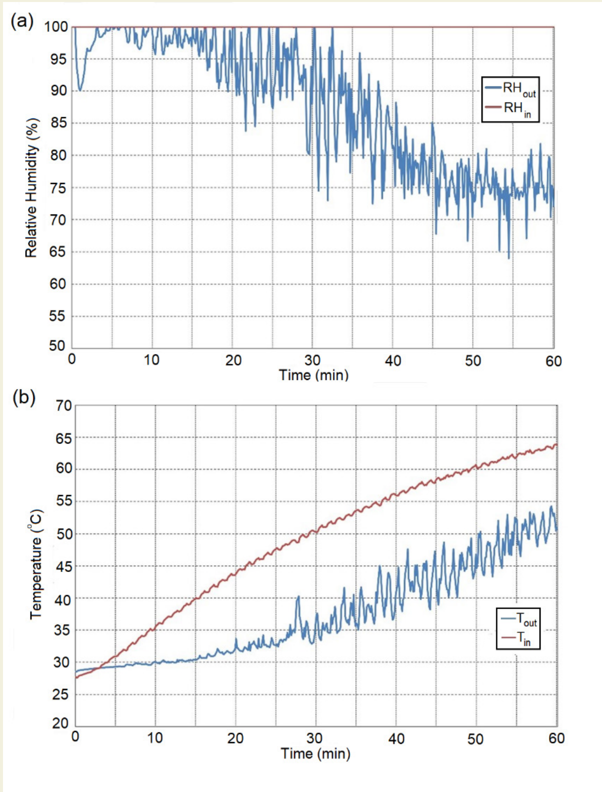

Figure 18. a) Relative Humidity and b) Temperature measurements for Test-4

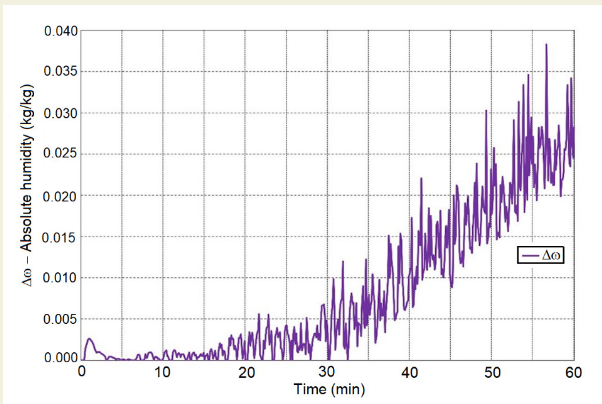

Figure 19. Absolute humidity measurements for Test-4

matized in Figure 22.

The relative humidity is the amount of moisture in the air at a given temperature compared to the maximum amount of water the air can hold, expressed as a percentage. On the other hand, absolute humidity is the amount of moisture in the air, usually measured in grams of water per cubic meter of air. When air is drawn into a condenser, its relative humidity and absolute humidity rates are influenced by the test environment in which it is circulated. Through the retrieved data obtained for five different experimental test conditions, time-dependent variation of the relative humidity is thoroughly evaluated. It is observed that effective drying can only occur in the last minutes of the drying cycle, which theoretically reached to 100% relative humidity range on the psychometric chart. Depending on the activation timespan of the spray nozzle, at the exit of the condenser, moist air reaches 100% relative humidity under the influence of the water spray. One can conclude from the observed time-varying inclinations of the experimental data set that a considerable amount of energy and water consumption could be avoided within a profitable time range if an efficient dehumidification regime is maintained in which the process air reaches the level of 100 % relative humidity.

Absolute humidity data extracted from the inlet and outlet of the condenser during the dehumidification process are considered for performance evaluation of the dehumidification system occupied in the water-cooled condenser. Collected data is utilized to gain insightful ideas on the drying system's water and energy consumption rates. They can also be evaluated as a reliable indicator of when the drying process can be terminated. Experi-

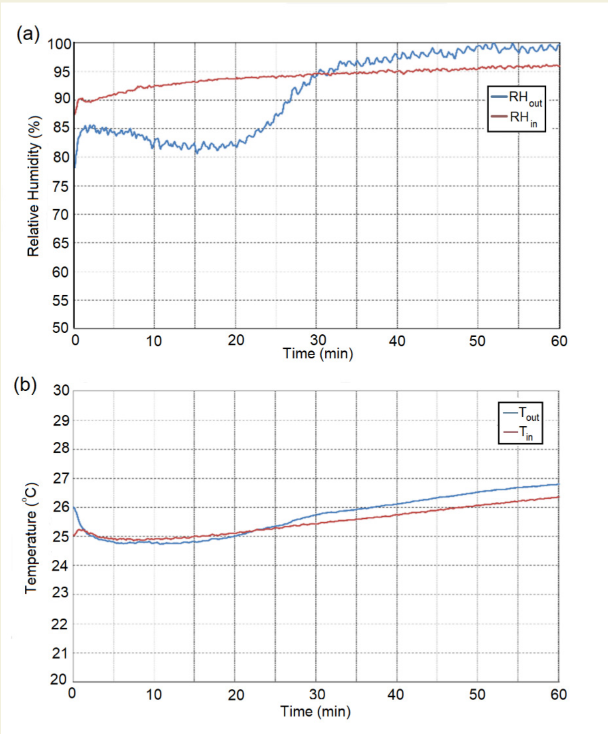

Figure 20. a) Relative Humidity and b) Temperature measurements for Test-5

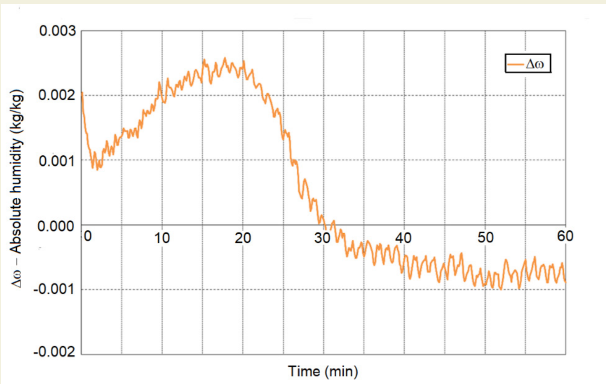

Figure 21. Absolute humidity measurements for Test-5

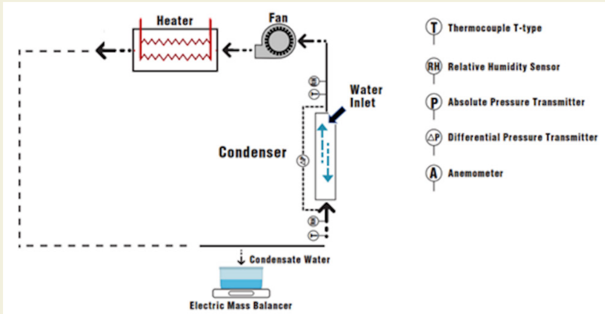

Figure 22. Schematic view of the test setup

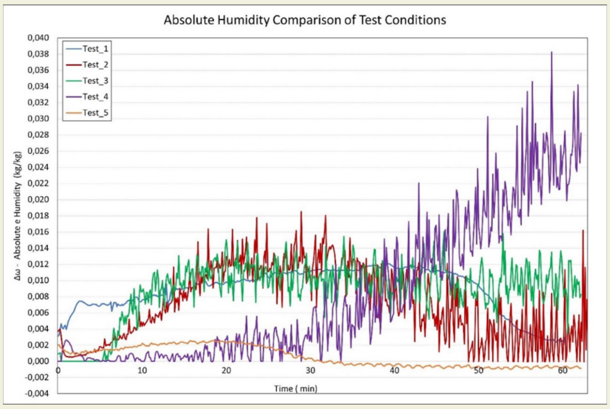

Figure 23. Comparison of the absolute humidity for five different test conditions

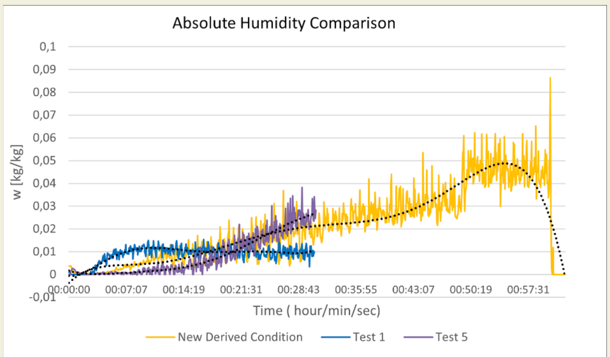

Figure 24. Comparison of the absolute humidity of new derived condition with Test 1 & Test 4

ments realized in Test-3 are conceptualized on the actual algorithm; however, according to the test results, it is seen that overall system efficiency in terms of dehumidification performance could be more satisfactory and consistent than the operational conditions employed for Test 4. As seen from the comparison chart in Figure 23, if a new software algorithm is developed considering the operating conditions used for Test-3 and Test-4, more efficient and prolific dehumidification performance can be obtained.

Heater and valve activations are settled in the new software like Test-4 conditions 20 minutes before the cooling phase, as given in Figure 24. Only one heater is activated during these 20 minutes, while water spray intake is not allowed into the system. This way, while the drying process continues without any loss at dehumidification performance, energy, and water requirements decrease for 20 minutes. The total test duration is also not changed. So, total water consumption was reduced by % 21.

7. Conclusive Remarks

This study investigates the coupled heat and mass transfer interactions between process air and operational environment in the water-cooled condenser of a washer& dryer machine. As mentioned in literature studies, mathematical modeling of the spray drying process requires tedious and time-consuming iterative calculations, which is very hard to accomplish using conventional solution methods. To overcome this computational drawback, this study aims to establish a relationship between drying control parameters obtained in the inlet and outlet of the condenser dryer. Experimental data of dry bulb temperature (*Tdb*), wet bulb temperature (*Twb*), relative (ϕ), and absolute humidity (w) of the process air measured in the condenser dryer inlet and outlet are correlated.

The procedural steps of the conducted experiments have been orderly summarized below, and the following conclusions have been drawn from the carried out experimental test as well as the outcomes of the numerical simulations;

- Various test conditions have been applied to comprehend the effects of drying parameters on condenser performance.
- These tests include different combinations of valve and heater activations to investigate the influences of the system components on the spray drying process.
- Relative and absolute humidity rates of the process air are the variables that monitor the drying performance of the condenser dryer. Therefore, utmost care should be taken during their measurements in the experiments.
- A trial algorithm is developed that only considers the effects of the heater, fan, and valve activations. The primary purpose of developing the trail algorithm is to simplify the evaluation of test samples obtained in the drying software.
- Two different humidity probes are simultaneously utilized to measure the dry bulb temperature, wet bulb temperature, relative humidity, and total pressure at the condenser dryer inlet and outlet in the one-hour drying program.
- During the one-hour drying cycle completed in the washer dryer, absolute humidity changes are also considered depending on the temperature and water spray activations.
- Energy and water requirements decrease for 20 minutes if the correct configuration of drying components is orderly activated.
- Furthermore, total water consumption is reduced by % 21.

This study proposes an effective drying model for textile drying applications. The numerical results show the model's reliability in reducing the experimental drying data density for textile drying.

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