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Abstract: This paper reports on the experimental investigations in a non-steady state mode of a small capacity commercial diffusion-absorption refrigerator (DAR) and the development of a dynamic black-box model for the machine. For these investigations, the refrigerator was equipped with the appropriate metrology. Temperature time variations of the refrigerated room and of the ambient conditions were measured, monitored, and stored using a data acquisition unit connected to a computer. A standardized experimental procedure was used to determine the overall heat conductance of the refrigerated room: A value of (UA)_cab= 0.554 WK-1 was found. The time evolution of the cooling capacity for different driving heat inputs to the refrigerator was investigated. Based on the experimental data, a dynamic black-box model was developed using the Matlab identification package to correlate the power input to the generator and the cooling capacity of the refrigerator. A first order transfer function with a delay was found to describe quite accurately the time evolution of the cooling capacity for all considered heat rates supplied to the generator. In a further step, regressed analytical expressions of the transfer function parameters, as a function of the generator heat supply, were incorporated into the cooling capacity function. A generalized dynamic black-box model for the DAR system was thus obtained and was then validated using the Matlab Simulink® environment. The predictions made by the model were found to be well in agreement with the experimental data. In particular, the predictions for COP under steady state conditions agreed satisfactorily with the experimental data yielding a maximum relative deviation of about 8%.

Experimental investigations and modelling of a small capacity diffusion-absorption refrigerator in dynamic mode

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Abstract

This paper reports on the experimental investigations in a non-steady state mode of a small capacity commercial diffusion-absorption refrigerator (DAR) and the development of a dynamic black-box model for the machine. For these investigations, the refrigerator was equipped with the appropriate metrology. Temperature time variations of the refrigerated room and of the ambient conditions were measured, monitored, and stored using a data acquisition unit connected to a computer. A standardized experimental procedure was used to determine the overall heat conductance of the refrigerated room: A value of $(UA)_{cab} = 0.554$ WK^{-1} was found. The time evolution of the cooling capacity for different driving heat inputs to the refrigerator was investigated. Based on the experimental data, a dynamic black-box model was developed using the Matlab identification package to correlate the power input to the generator and the cooling capacity of the refrigerator. A first order transfer function with a delay was found to describe quite accurately the time evolution of the cooling capacity for all considered heat rates supplied to the generator. In a further step, regressed analytical expressions of the transfer

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Keywords: Absorption diffusion refrigeration, Dynamic modelling, Matlab Simulink®.

Highlights

- Dynamic operation of a small capacity commercial diffusion-absorption refrigerator is experimentally investigated.
- A first order transfer function is used to describe the relationship between the driving power and the cooling capacity.
- A generalized dynamic black-box model for the DAR refrigerator is developed using Matlab Simulink® environment.
- The predictions made by the model for steady-state COP are well in agreement with experimental data.

Nomenclature

COP: Coefficient of performance (-)

$F(t)$: Unit step function (-)

G_R: Transfer function (-)

K_P: Static gain (-)

$\dot{Q}_{gen}$: Generator heating power (W)

$\dot{Q}_{heat}$: Electric cable heating power (W)

$\dot{Q}_f$: Cooling capacity (W)

R: Resistance of the electric cable (Ohm)

s: Laplace variable (-)

T: Temperature (°C)

t: Time (s)

U, Y: Laplace transform of input and output (-)

(UA): Overall heat conductance (W K^{-1})

τ_d: Time delay (s)

θ_p: Time constant (s)

φ: Applied voltage (V)

Subscripts

amb: ambient

cab: Refrigerated room

elec: Electric

evap: Evaporator

int: Refrigerator interior

1. INTRODUCTION

Diffusion absorption refrigeration (DAR) systems are widely used in supermarkets, domestic freezers, and hotel rooms, etc. This technique for producing cold was patented in 1928 by the two Swedish engineers, van Platen and Munters [1]. The unique feature of the DAR cycle, compared to a conventional ammonia/water absorption cycle, is that it operates at a uniform pressure. The working fluid is a mixture of three components: ammonia as a refrigerant, water as an absorbent, and an auxiliary inert gas, frequently hydrogen or helium. The inert gas is necessary to reduce the partial pressure of the refrigerant in the evaporator, and allow for it to evaporate at low temperatures and for the production of useful cold. DAR systems are silent because they have no moving parts.

Over the years, investigations have been published on the performance of various configurations of DAR cycles using graphical, numerical and experimental approaches. Kouremenos *et al*. [2] investigated the use of helium as an alternative to hydrogen. The authors observed a similar behaviour of both inert gases. Zohar *et al.* [3] developed a thermodynamic model for an Electrolux DAR system and analysed its performance with both inert gases and found that the COP of the system operating with helium was 40% higher than that of the system operating with hydrogen. They further reported that higher COP values were obtained by using an ammonia mass fraction of 30% in the rich solution and of 10% in the weak solution. Zohar *et al.* [4] compared the configurations of a DAR cycle both with and without condensate sub-cooling prior to the evaporator entrance. They showed that in the DAR cycle without sub-cooling there was a 14-20% higher COP than in the DAR cycle with condensate sub-cooling. Zohar *et al.* [5] examined numerically the performance of a DAR system operating with the organic absorbent DMAC which was associated to five different refrigerants and with helium as an inert gas. They compared the performance of their systems with that of the system operating with ammonia-water and helium and found out that that the

latter had the highest COP values. Ben Ezzine *et al.* [6] investigated the feasibility of a DAR system operating with the working fluid system DMAC-*R*124-He and coupled to a solar collector. They reported that both the COP and the temperature of the cooling effect depended largely on the effectiveness of the absorber and the generator temperature. For solar cooling applications, this working fluid mixture could be an alternative to the conventional ammonia-water-hydrogen mixture. Starace and De Pascalis [7] developed an enhanced thermodynamic model to consider the more realistic operating conditions of the DAR cycle, such as the presence of some water (absorbent) in the refrigerant stream (ammonia). They used a magnetron activated thermal pump in order to reduce the start-up time of the refrigerator. The model's predictions were validated on a prototype built by coupling a domestic magnetron to a small purposely modified commercial DAR. A maximum deviation was noticed to be roughly 2% in the weak solution mass flow rate and lower than 5% in the COP between the predicted and measured data. Sayadi *et al.* [8] presented a HYSYS simulation model for a water-cooled DAR system using different binary mixtures of light hydrocarbons (C_3/*n*-C_6, C_3/cyclo-C_6, C_3/cyclo-C_5, propylene/cyclo-C_5, propylene/*i*-$C_4$and propylene/*i*-C_5) as working fluids and helium as an inert gas. The driving heat in the generator was supposed to be supplied by evacuated tube solar collectors. The most appropriate binary mixture was found to be C_3/*n*-C_6 with a generation temperature of 126°C. Long et *al.* [9] investigated numerically the possibility of using TFE-TEGDME in the DAR system together with two cooling mediums, namely water at 32 ºC and ambient air at 35 ºC. The authors performed a parametric analysis on the effects of the cooling medium, the generation and evaporation temperatures and the effectiveness of the absorber on the system performance. They compared the performance of the TFE-TEGDME and NH_3-H_2O DAR systems in terms of COP and circulation ratio. They concluded that the TFE-TEGDME mixture is a good working fluid for DAR systems and found that with an absorber effectiveness of 0.8, the

optimum generation temperature for the air-cooled TFE–TEGDME DAR system is around 170°C. A coefficient of performance (COP) up to 0.45 was obtained. However, the performance of the water-cooled system was better with a lower generation temperature of 130°C and a higher COP of 0.56. Rodriguez-Munoz and Belman-Flores [10] presented a review on diffusion-absorption refrigeration technology in which over 70 publications were analysed. The authors concluded that diffusion-absorption technology represents an interesting and feasible alternative for small capacity refrigeration applications. Rattner and Garimella [11] proposed a fully passive DAR system operating with the working fluid mixture NH_3-NaSCN-He. Detailed design models for the various components of the system were elaborated. They reported COPs in the range of 0.11-0.26 at an ambient air temperature of 24ºC, low heat source temperatures of 110-130°C and passive air cooling. These authors reported [12] on the development of a prototype of the theoretically investigated machine, activated by low temperature heat sources (110 - 130°C) and passively air-cooled. The cooling temperatures achieved were suitable for refrigeration (T_{evap}= 6 → 3°C, COP ~ 0.06) and air-conditioning (12 → 8°C, COP ~ 0.14; 18 → 14°C, COP ~ 0.17). Chen *et al*. [13] improved the coefficient of performance of the DAR system by 50%. They modified the design and construction of the generator by integrating a tube-in-tube solution heat exchanger into the generator. Vicatos [14] studied experimentally a modified domestic DAR in order to reduce the response time of the system. Koyfman *et al.* [15] presented an experimental investigation on the bubble pump performance in a DAR system. They used a solution of an organic solvent and HCFC as refrigerant. Their results showed that the performance of the bubble pump depends mostly on the motive head and the heat input to the bubble pump. Jacob *et al.* [16] conducted a theoretical and experimental study on a solar driven ammonia-water diffusion absorption cooling machine (DACM). They designed four prototypes for air-conditioning applications: water chillers with an evaporator temperature in the range of 6-

12°C and ceiling cooling with an evaporator temperature of 15-18°C. The COP values achieved ranged from 0.10 to 0.45.Yıldız and Ersöz [17] designed a DAR system driven by electricity. They investigated numerically and experimentally the energy and exergy losses for each component of the system and compared the theoretical and experimental values. They concluded that the highest exergy losses took place in the solution heat exchanger. The experimental and predicted COP was around 0.19 and the exergy efficiency between 0.03 and 0.04. Mazouz *et al*. [18] studied experimentally a commercial DAR refrigerator in order to determine its performance under various operating conditions and developed a theoretical simulation model of different DAR configurations.

Ben Jemaa *et al*. [19] examined and discussed the relationship between the driving heat and the cooling capacity of a DAR refrigerator in start-up mode. They reported that a transfer function of a first order with a delay is an adequate model to predict the cooling capacity of the refrigerator in steady state mode with a maximum deviation of 8% for different values of the driving heat in the generator.

The majority of published works on diffusion absorption refrigeration systems available in the literature deals with steady-state modelling associated or not to experimental tests, but few investigations have been performed on the dynamic behaviour of this kind of refrigeration system. In fact, most of the studies published on the dynamic behaviour of cooling systems [20-28] do not deal with DARs. However, it is of great importance to understand the dynamic operation of these systems. In the present work, a black box dynamic model was developed to predict the refrigerator performance in terms of cooling capacity and COP. The model was based on experimental measurements taken from a small capacity commercial diffusion-absorption refrigerator. The specific objectives are as follows:

i. Experimental characterization of a small capacity commercial diffusion-absorption refrigerator in dynamic mode.

ii. Development and validation of a detailed black-box model for this refrigerator, based on the experimental data and using Matlab Simulink® environment.

2. Working principle

A 3-D scheme of the investigated refrigerator is presented in Fig. 1. It is a small capacity refrigerator of 44 litres and internal dimensions of 490 mm x 360 mm x 250 mm (height x width x depth), designed for hotel rooms, activated by an electric heater, and operating with the ternary fluid mixture ammonia-water-hydrogen. The machine has of a generator which consists of a combined bubble pump and a boiler, a rectifier, a condenser, an evaporator, a gas heat exchanger (GHX), an absorber, a solution heat exchanger (SHX) and a solution tank.

The generator contains initially a rich ammonia-water solution. When sufficient heat is supplied, ammonia vapour is generated and moves up through the vertical bubble pump tube carrying the liquid solution with it. At the top of the bubble pump the ammonia solution and ammonia vapour are separated. The liquid solution falls in the boiler and continues further on to the absorber by force of gravity. The vapour flows to the condenser, where it condenses. The ammonia liquid flows further to the evaporator where it meets hydrogen coming from the absorber. Its partial pressure is so decreased that the liquid begins to evaporate by taking heat from the refrigerated space. After leaving the evaporator, the gaseous mixture is conducted through the annulus of the tube-in-tube gas heat exchanger (GHX) to ensure that it precools the liquid refrigerant and the hydrogen-rich gas flowing counter-currently on their way up to the evaporator. The mixture of refrigerant and hydrogen flows further down to the absorber inlet. In the absorber, the ammonia vapour is absorbed by the ammonia-weak solution returning from the generator, transforming it to an ammonia-rich solution, which is collected in the solution tank. From there it starts its way to the bubble pump via the solution heat exchanger (SHX) and then back to the generator.

3. Experimental characterization of the dynamic operation of the refrigerator

To experimentally investigate the functioning of the DAR system in dynamic mode, the refrigerator was equipped with 11 K-thermocouples placed at the entrance and exit of each of its components and connected via a data acquisition unit (34970A AGILENT) to a computer (Fig. 2) where the experimental data were collected and saved. The thermocouples were calibrated beforehand using an ice-water bath. A fluctuation in the temperature of ±0.5°C was noticed. The electric heating system was connected to a power controller (Fig. 2). The experimental tests were carried out by first adjusting the heating power to the generator, $\dot{Q}_{gen}$, then measuring the temperatures and storing the data all the time, from the start-up until the steady-state regime was reached. The thermostat was disconnected so that the refrigerator was operated in a continuous mode and an air-conditioned environment at a set point temperature of 24°C. It is worthy of note that the experimental procedure employed in the present work doesn't follow any protocol of the standards developed for laboratory testing of household refrigerators [29-34]. The experiments were performed with different heat supplies $\dot{Q}_{gen}$ to the generator, namely 35, 39, 44, 46, 48, 51, 53, 56, 58, 61 and 67 W. Fig. 3 shows the time evolution of the liquid stream temperature at the generator exit for the two lowest values of the energy supply to the machine. For $\dot{Q}_{gen}$= 35W, the temperature exhibits an oscillatory behaviour: it fluctuates between 205°C and 155°C with a time period of 100 minutes. This heat supply ensures the functioning of the refrigerator, but not its stability. With a little increase in the heat supply, $\dot{Q}_{gen}$= 39W, some fluctuations are initially observed. They then cease after 150 minutes, leading to a steady-state operation with the temperature of the solution leaving the generator stabilizing at 158°C. In fact, when the heating power supplied to the generator was not sufficient, the bubble pump did not reach steady operation point. The large vapour bubbles responsible for the pumping of the liquid were not continuously formed. Although ammonia vapour was generated, the flow regime was almost bubbly and the liquid

was only occasionally conveyed upwards. With increasing heat supply, a stable plug flow regime was reached and the steady pumping of the liquid was ensured. Moreover, increasing the heat power supply to the generator $\dot{Q}_{gen}$ from 44W to 67W the initial fluctuations of the solution temperature at the generator exit were attenuated (Fig. 4(a)) and the time required by the refrigerator to reach a steady-state regime reduced (Fig. 4(b)) at a given temperature of the refrigerated room. Fig. 5 shows how the temperatures at the evaporator inlet and outlet and in the refrigerated room respond to four different values of the energy supply $\dot{Q}_{gen}$ in the range of 39-67W. It is worthy of note that in the case of $\dot{Q}_{gen} = 39\text{W}$, the temperatures at the inlet and outlet of the evaporator are very far apart (-22°C *vs.* -5°C) and that the temperature at the inlet fluctuates largely between +10 and -22°C.

To evaluate the cooling capacity of the refrigerator, $\dot{Q}_f(\text{t})$, the overall heat conductance $(UA)_{cab}$ of the refrigerated room is needed. For the present study, it was determined by applying a standardized procedure in a separate test set. To this purpose, the refrigerated room of the refrigerator was heated by an electric resistance cable (R = 135Ω) placed inside. This heater was connected to a power controller in order to vary the electric power supplied. Seven tests were performed by varying the voltage φ applied to the resistor, namely φ: 17, 20, 22, 26, 28, 30 and 32 V. For each test, the indoor and outdoor temperatures were measured. When steady-state regime is reached, the indoor temperature becomes constant, i.e. the heat generated in the electric resistance cable $\dot{Q}_{heat} = \left[{\varphi^2}/{R}\right]$ is equal to the total heat losses $\dot{Q}_{loss}$ from the refrigerated room (Eq. 1):

$$\dot{Q}_{loss} = (UA)_{cab}(T_{int} - T_{amb}) \qquad (1)$$

It follows then:

$$(UA)_{cab} = \frac{\dot{Q}_{heat}}{(T_{int} - T_{amb})} = \frac{\left[{\varphi^2}/{R}\right]}{(T_{int} - T_{amb})} \qquad (2)$$

The experiment was repeated for seven different values of $\dot{Q}_{heat}$. The slope of the regression line representing $\left[\varphi^2/_R\right]$ *vs.* $(T_{int} - T_{amb})$, shown in Fig. 6, leads to an average value of $(UA)_{cab} = 0.55$ WK^{-1}, with an absolute uncertainty of ±0.01 WK^{-1}.

The material properties of the refrigerated room walls and the thermo-physical properties of ambient air outside the refrigerator were constant during the tests and the mean properties of air inside the refrigerator can be assumed roughly unchanged, so the obtained value of $(UA)_{cab}$ was used to determine the instantaneous cooling capacity $\dot{Q}_f(t)$ from the start-up of the refrigerator till it reaches a steady-state regime (Eq. 3), the relative uncertainty being around 4%.

$$\dot{Q}_f(\mathrm{t}) = (UA)_{cab}(T_{int}(t) - T_{amb}) \qquad (3)$$

4. Dynamic black-box modelling of the refrigerator

Very often process models are developed from knowledge gained of the underlying process mechanisms and they describe the physicochemical processes taking place in the system. These mechanistic models known as "white boxes" are developed for design and optimization applications.

Contrarily, empirical models are solely the result of experiments and observation and do usually not rely on the knowledge of basic principles and mechanisms. Equation fitting is then employed to determine the model parameters, which have little or no physical meaning. These so-called "black box" models are of purely descriptive nature and with limited ranges of application however, they do constitute helpful tools for process control and automation purposes.

In this kind of modelling, the pertinent information for model identification is obtained through exciting or stimulating the system. Structure and parameter estimation methods

developed in the field of process identification are then used to establish the model. These two steps are however, not really independent, as the criterion of the choice of a model from a set of candidates with unknown parameters is the best fit for the measured data.

In the choice of a black-box model for a DAR system, one should keep in mind requirements for good model structure, namely:

- Simplicity—in its mathematical form,
- Generality—in its ability to describe the refrigerator behaviour for all power inputs.

4.1 Mathematical model

The model developed is based on some simplifying assumptions.

- First, the ambient temperature is supposed constant. This takes in account the fact that the tests are performed in a temperature-controlled room, with a temperature varying between 24 and 26°C.
- Second, the steady-state values of the overall heat conductance $(UA)_{cab}$ are supposed to be the same in transient mode. This means that the latent heat contribution during the start-up is neglected, as the steady state $(UA)_{cab}$-value accounts only for sensible load.
- Further, at constant ambient temperature, the heating power supplied to the generator is the only input that affects the operation of the refrigeration unit. Hence, the input signal to the system is $\dot{Q}_{gen}(t)$:

$$\dot{Q}_{gen}(t) = \dot{Q}_{gen}\, F(t) \qquad (4)$$

where $F(t)$ is the unit step function.

The system response to this input is the cooling effect $\dot{Q}_f(\mathrm{t})$ achieved in the refrigerated room. In terms of system theory, the used DAR unit is then a single-input, single-output (SISO) system. To establish a dynamic black-box model of the refrigerator, nine (repeated) tests were carried out using different inputs, i.e. the heat supply to the generator of the

refrigerator $\dot{Q}_{gen}(t)$, and the response of the system, i.e. the time evolution of the cooling effect $\dot{Q}_f$(t), noticed. As mentioned before, the next two steps in the elaboration of a model are the model structure and the estimation of model parameters. In the present work, these steps were performed by analysing the experimental input and output data using the software MATLAB and its system identification package. It was found that the most adequate dynamic model for the description of the time evolution of $\dot{Q}_f$(t) for every chosen energy supply $\dot{Q}_{gen}$ corresponds to the response of a first order system with a delay. Fig. 7 illustrates this finding in the case of the four values of $\dot{Q}_{gen}$= 44W, $\dot{Q}_{gen}$= 48W, $\dot{Q}_{gen}$= 51W and $\dot{Q}_{gen}$= 53W. This means that the transfer function $G_R(s)$ of the DAR refrigerator is in the form:

$$G_R(s) = \frac{K_P}{1+\theta_p s} e^{-\tau_d s} \quad (5)$$

where K_P is the static gain, θ_p the time constant and τ_d the time delay. When applied to our machine data this simple three-parameter model describes quite accurately the time evolution of the cooling capacity for all values of the heat power supply to the generator as shown in Fig. 7. The fitting quality, an indicator of the quality of the regression results, ranges from 90% to 98%.

The transfer function that relates the rate of heat supply $\dot{Q}_{gen}$ as an input signal and the cooling capacity $\dot{Q}_f$(t) as an output signal is:

$$G_R(s) = \frac{Y(s)}{U(s)} = \frac{K_P}{1+\theta_p s} e^{-\tau_d s} \quad (6)$$

with $U(s)$is the Laplace transform of $\dot{Q}_{gen}(t)$,

$$U(s) = \mathcal{L}\left[\dot{Q}_{gen} F(t)\right] \quad (7)$$

and $Y(s)$ the Laplace transform of the cooling capacity $\dot{Q}_f(t)$,

$$Y(s) = \mathcal{L}\left[\dot{Q}_f(t)\right] \quad (8)$$

The time domain of the cooling capacity $\dot{Q}_f(\mathrm{t})$ can be deduced from equation (6),

$$\dot{Q}_f(\mathrm{t}) = \dot{Q}_{gen} K_P \left(1 - e^{\frac{-t-\tau_d}{\theta_p}}\right) F(t-\tau_d) \qquad (9)$$

Fig. 7 shows the calculated time evolution of $\dot{Q}_f(\mathrm{t})$ in comparison with the experimental curves for the indicated values of the heating power supply to the generator. It can be appreciated that the black-box model reproduces quite accurately the unsteady-state performance of the DAR refrigerator.

Table 1 summarizes the model parameters and the fitting quality indicator for all investigated values of $\dot{Q}_{gen}$.

4.2 Generalized black-box dynamic model & validation

The analysis of the results in Table 1 reveals that the three model parameters K_P, $\theta_p$and τ_d depend on the heat input $\dot{Q}_{gen}$. This dependency is graphically depicted in Figs. 8, 9 and 10 with dots representing the values of the parameters and continuous lines their least-square fitted curves. If the corresponding regression relations $K_P(\dot{Q}_{gen})$,$\theta_p(\dot{Q}_{gen})$ and $\tau_d(\dot{Q}_{gen})$ are incorporated into the expression of $\dot{Q}_f(\mathrm{t})$ (Eq. 9), one obtains a generalized model of the DAR refrigerator, i.e. a function $\dot{Q}_f(t, \dot{Q}_{gen})$ which predicts the instantaneous evolution of the cooling capacity for every heating power supply to the generator within the range investigated.

The test for this generalized black-box dynamic model of the DAR refrigerator was run using the software package MATLAB SIMULINK® (Fig. 11). The time evolution of the cooling capacity calculated by both the specific and generalized models in comparison with experimental data describes reasonably well the dynamic behaviour of the DAR refrigerator and predicts fairly well the unsteady-state performance of the refrigerator. The fitting quality of the generalized model for the different inputs $\dot{Q}_{gen}$ ranges between 84% and 94%.

Fig. 12 shows a further test on the black-box models. It represents a comparison between the experimental cooling capacity and its calculated values for different values of $\dot{Q}_{gen}$ and for both the heat-rate-specific and generalized dynamic models. As can be seen, both models satisfactorily predict the cooling capacity of the used DAR refrigerator when steady-state regime is reached. Table 2 summarizes the relative deviations in predicting the cooling capacity at the different heat rates supplied to the generator. The maximum relative error is 8.2% and 7.5% for the specific and generalized models, respectively.

The experimental steady-state coefficient of performance (COP) values of the DAR (Eq. 9), predicted for different heat supplies to the generator by both models are well in agreement with the experimental data. The maximum relative deviation in COP prediction is about 8%.

$$COP = \frac{\dot{Q}_{evap}}{\dot{Q}_{gen}} \quad (9)$$

Therefore, it can be concluded that both specific and generalized black-box models predict the cooling capacity and the coefficient of performance of the DAR refrigerator well when a steady-state regime is reached.

5. Conclusion

In the present work, experimental investigations were carried out on the operation of a small capacity commercial diffusion-absorption refrigerator (DAR) at an unsteady-state regime and a dynamic black-box model was developed for the refrigerator. The results of these investigations can be summarized as follows:

- A heat supply to the generator of over 35 W is required to ensure the functioning of the commercial DAR refrigerator and its stability.

- The standardized experimental procedure used to determine the overall heat conductance of the refrigerated room leads to an average value of $(UA)_{cab} = 0.554$ WK^{-1}.
- A first order transfer function with a delay correctly describes the relationship between the power input to the generator and the cooling capacity as a response of the machine. A maximum relative deviation of about 8.2% between the predicted values by the model and the experimental data was noted.
- A generalized black-box model was then built by integrating analytical expressions of the three parameters of the model as function of the heat power supplied to the generator in the original specific model.
- Simulations with the obtained generalized model using Matlab Simulink® environment show that the predictions made by the model deviate by a maximum of 8% from the experimental data.
- The steady-state COP is also well predicted by the generalized black-box model.

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Table captions

Table 1. Model parameters and fitting quality for all investigated values of $\dot{Q}_{gen}$.

Table 2. Relative deviation in predicting the cooling capacity for both heat-rate-specific and generalized models.

Table 1. Model parameters and fitting quality for all investigated values of $\dot{Q}_{gen}$.

$\dot{Q}_{gen}$ (W)	K_p	τ_d	θ_p	Fit (%)
44	0.1395	78.927	66.829	93.91
46	0.1450	71.546	66.618	91.27
48	0.1354	72.092	57.055	94.16
51	0.1253	63.109	57.24	93.08
53	0.1120	60.232	52.523	98.36
56	0.1156	69.676	55.609	93.44
58	0.1096	67.327	54.474	96.90
61	0.0965	56.378	41.623	89.86
67	0.0947	71.510	45.930	95.41

Table 2. Relative deviation in predicting the cooling capacity for both heat-rate-specific and generalized models.

$\dot{Q}_{gen}$ (W)	Cooling capacity (W) *Experimental*	Cooling capacity (W) *Specific model*	Relative deviation (%)	Cooling capacity (W) *Generalized model*	Relative deviation (%)
44	6.43	6.09	5.4	6.10	5.4
46	6.06	6.56	8.2	6.45	6.3
48	6.14	6.24	1.5	6.25	1.8
51	6.64	6.32	5.0	6.32	5.0
53	5.56	5.47	1.8	5.82	4.7
56	6.44	6.43	0.2	6.21	3.7
58	6.10	6.10	0.1	5.97	2.1
61	6.15	5.82	5.8	6.11	0.7
67	6.44	6.16	4.4	5.98	7.5

Figure captions

Fig. 1. Schematic view of the diffusion-absorption refrigerator.

Fig. 2. Picture of the refrigerator investigated together with the auxiliary experimental equipment.

Fig. 3. Time evolution of the temperature of the weak solution leaving the generator for $\dot{Q}_{gen}$= 35W and 39W.

Fig. 4. (a) Time evolution of the temperature of the weak solution leaving the generator for $\dot{Q}_{gen} = 44 - 67$ W ; (b) Time required by the refrigerator to reach a steady-state regime vs. heat supply to the generator at a given temperature of the refrigerated room.

Fig. 5. Time evolution of the ambient air, refrigerated room and evaporator temperatures at different heat supplies to the generator.

Fig. 6. $\left[\varphi^2/_R\right]$ *vs.* $(T_{int} - T_{amb})$.

Fig. 7. Time evolution of the cooling capacity determined from experimental measurements and of the heat-rate-specific black box model at different heat supplies to the generator.

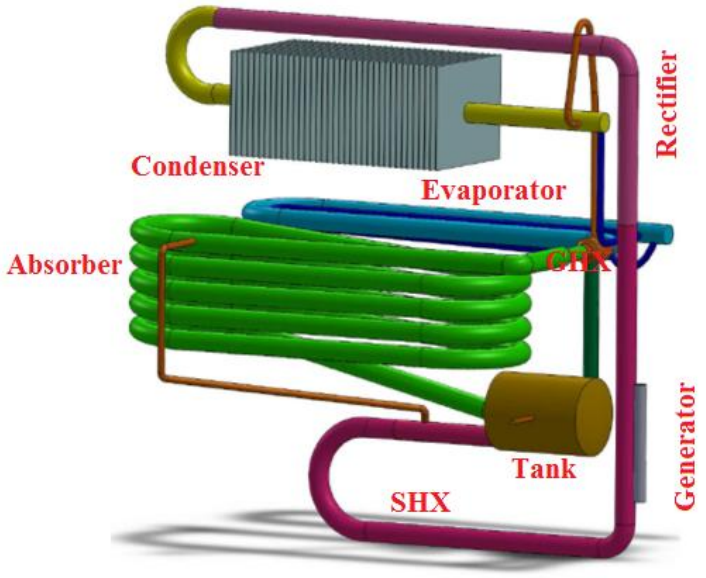
Fig. 8. K_P *vs.* $\dot{Q}_{gen}$.

Fig. 9. τ_d *vs.* $\dot{Q}_{gen}$.

Fig. 10. θ_P *vs.* $\dot{Q}_{gen}$.

Fig. 11. Schematic diagram of the black-box dynamic model in the Matlab Simulink® environment.

Fig. 12. Comparison of the experimental steady-state cooling capacity and that predicted by both the heat-rate-specific and generalized models as a function of the heat supply to the generator.



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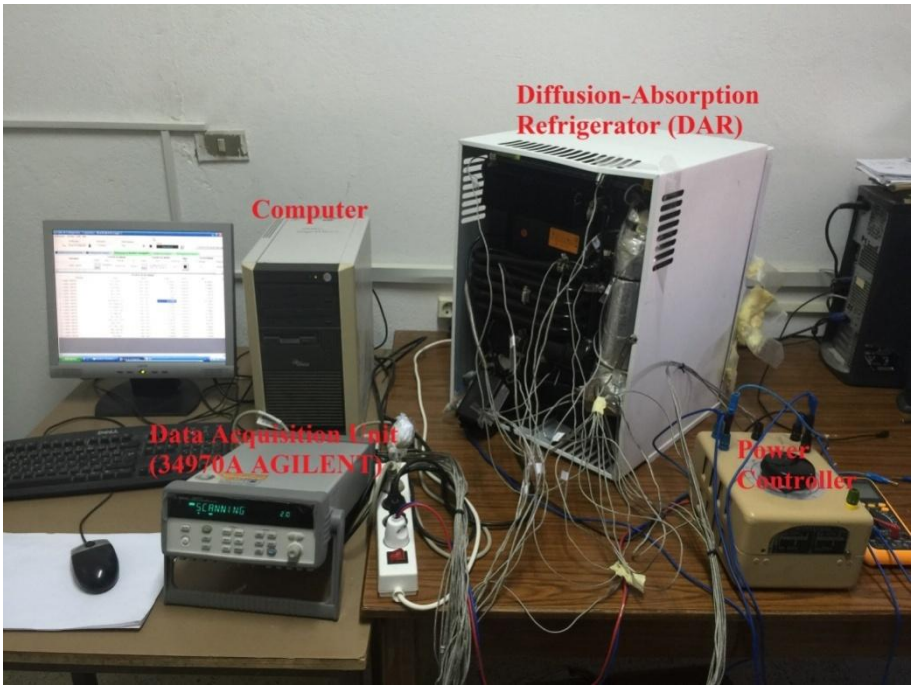

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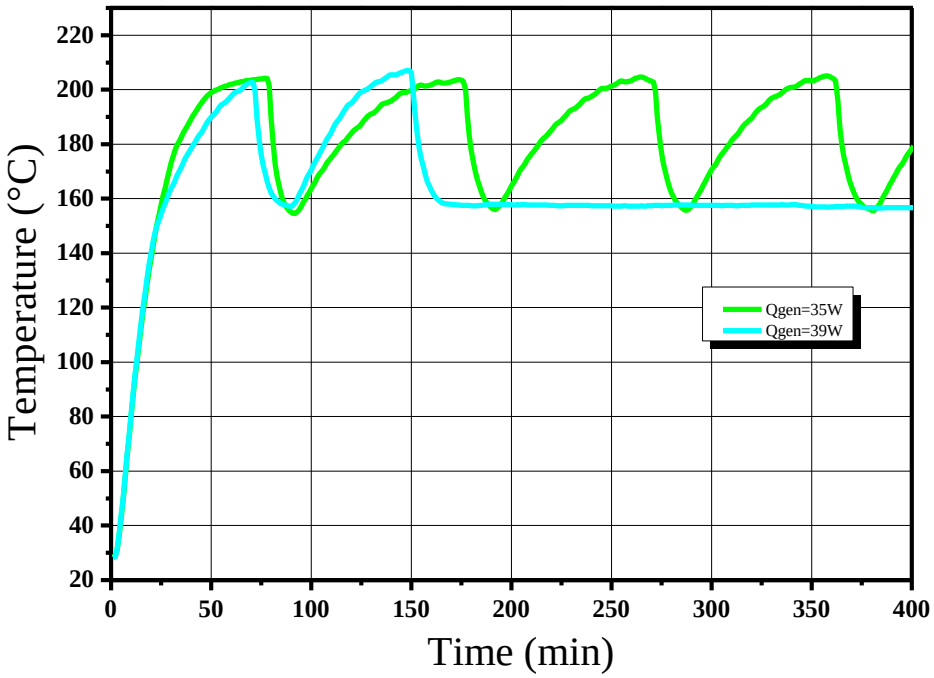

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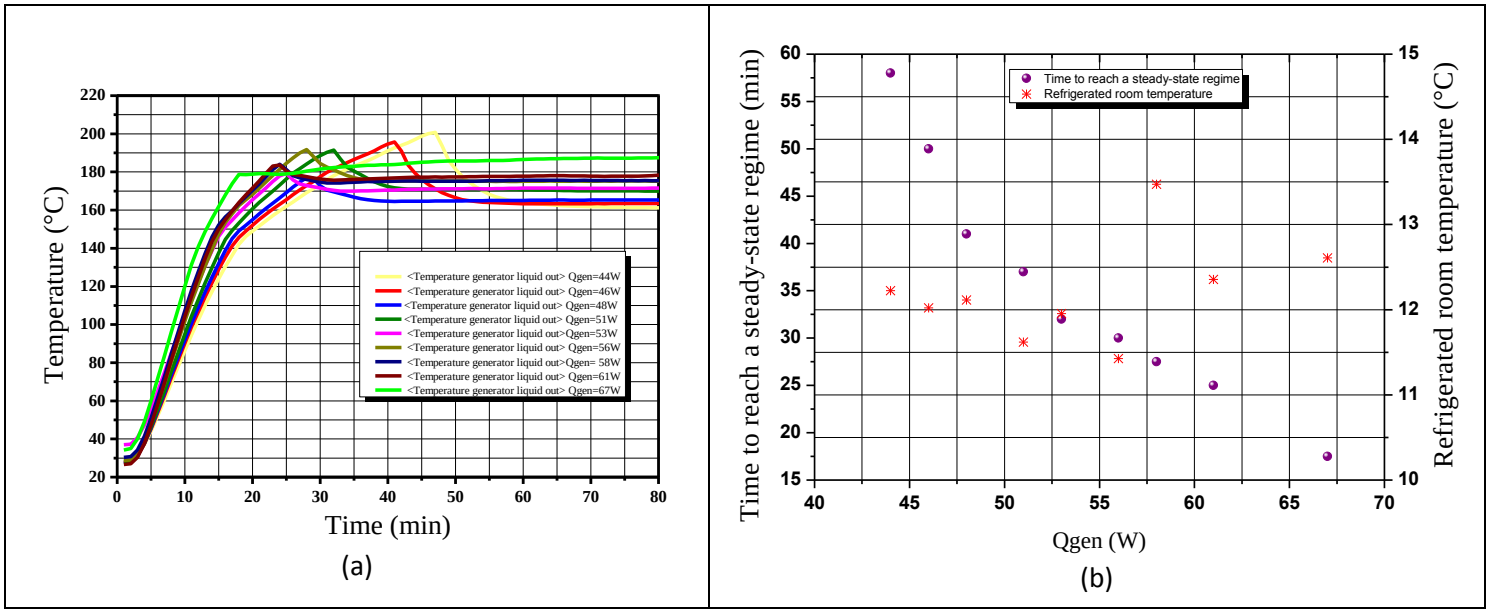

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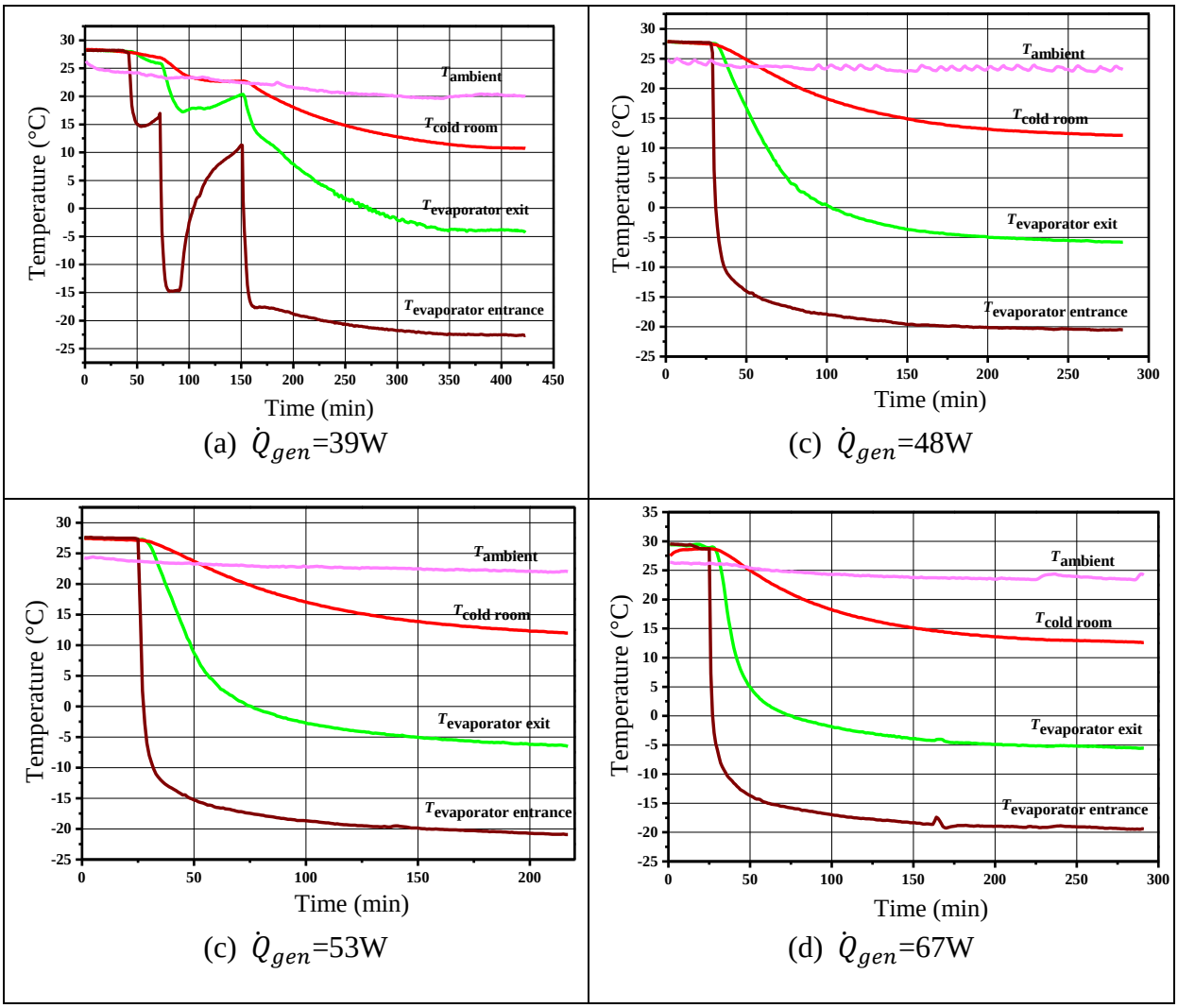

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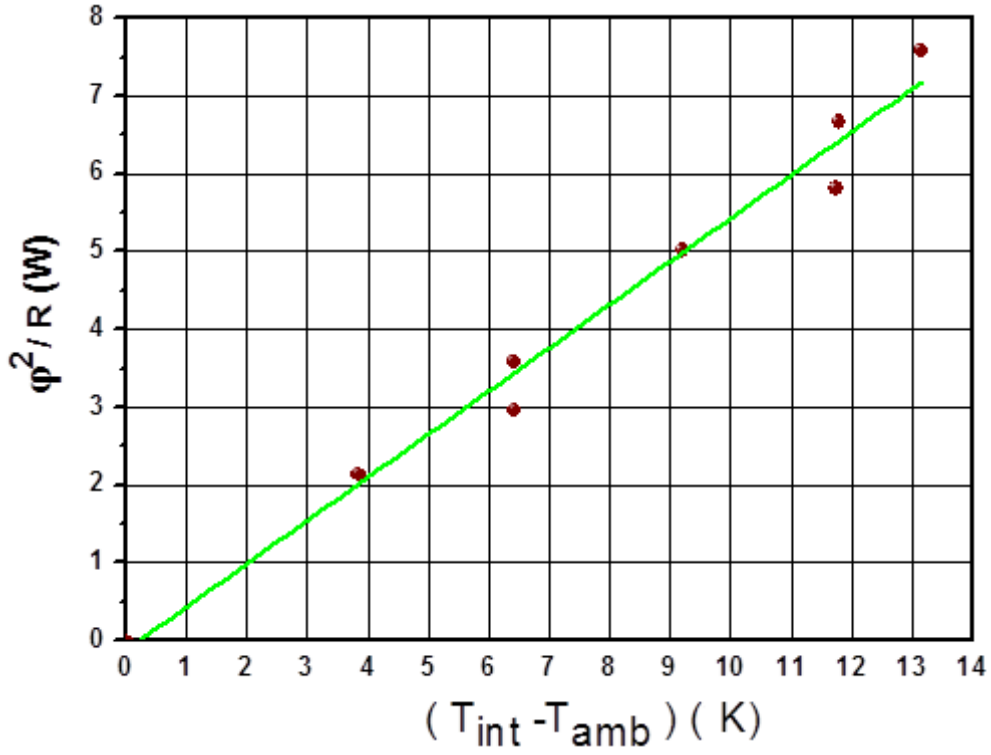

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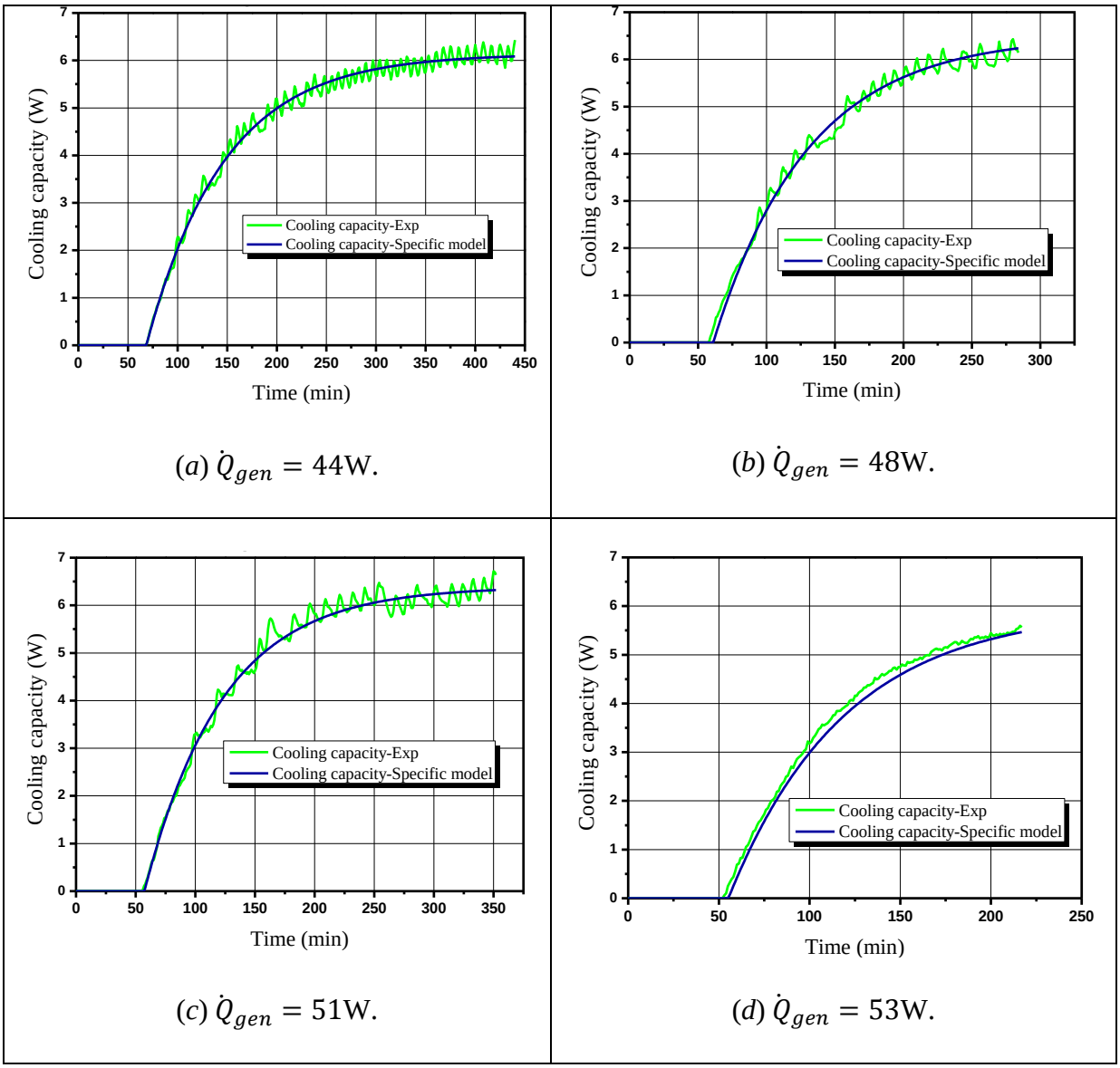

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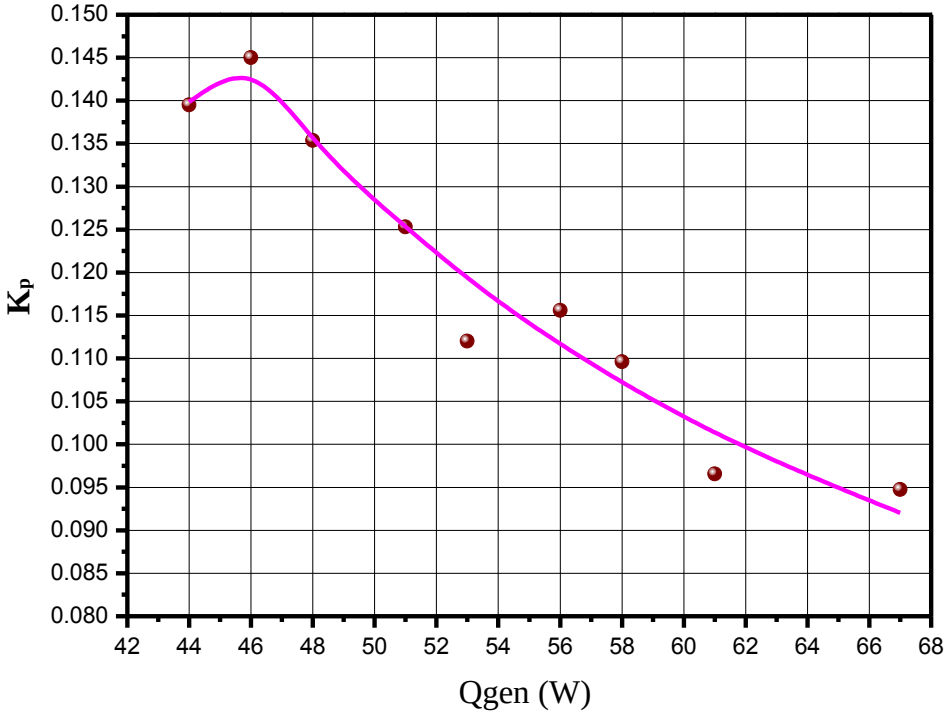

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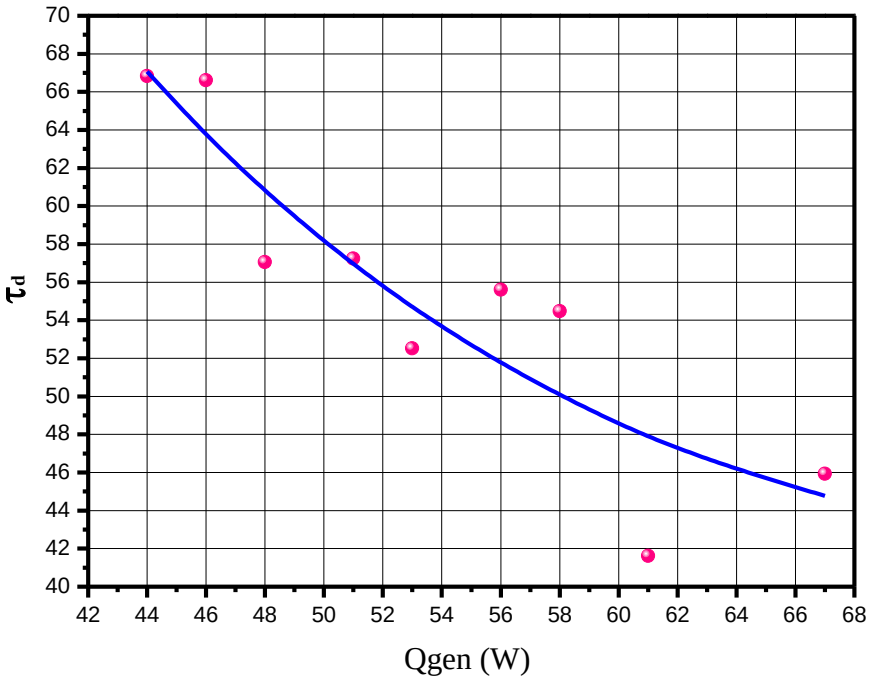


Fig. 9. τ_d *vs.* $\dot{Q}_{gen}$.

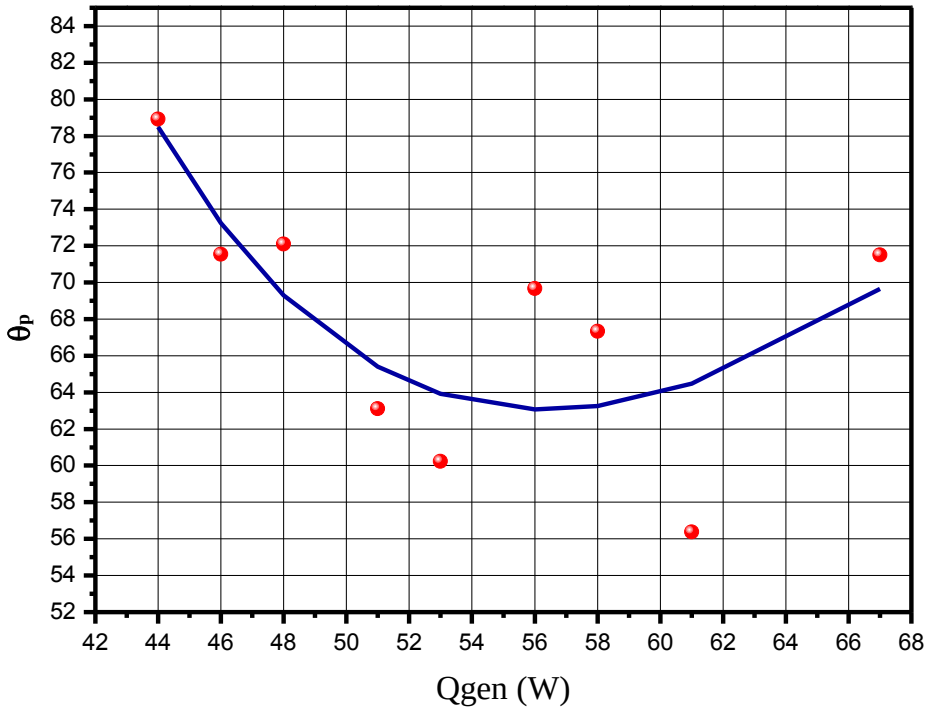
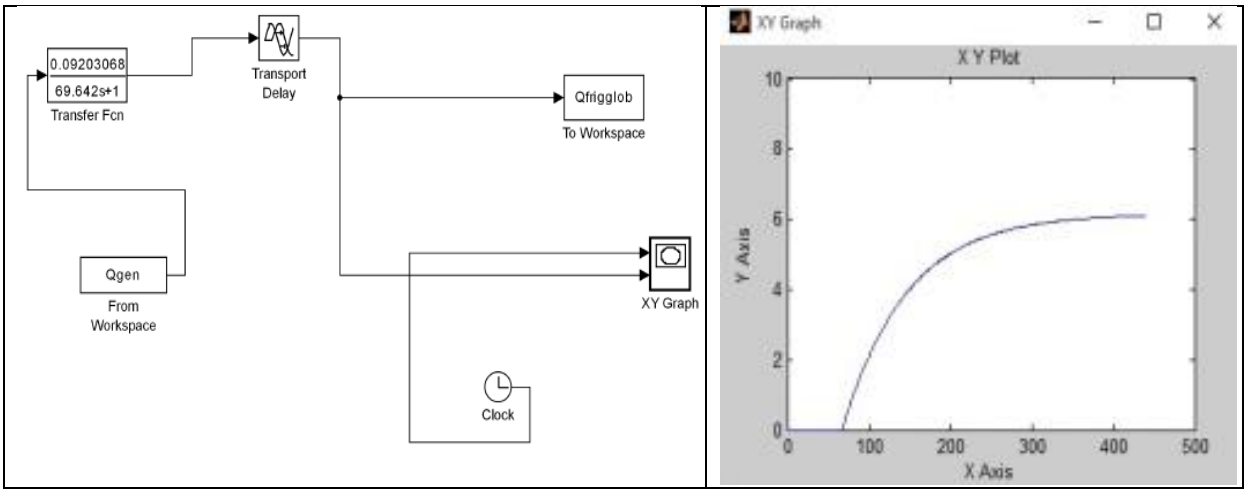


Fig. 10. θ_P *vs.* $\dot{Q}_{gen}$.



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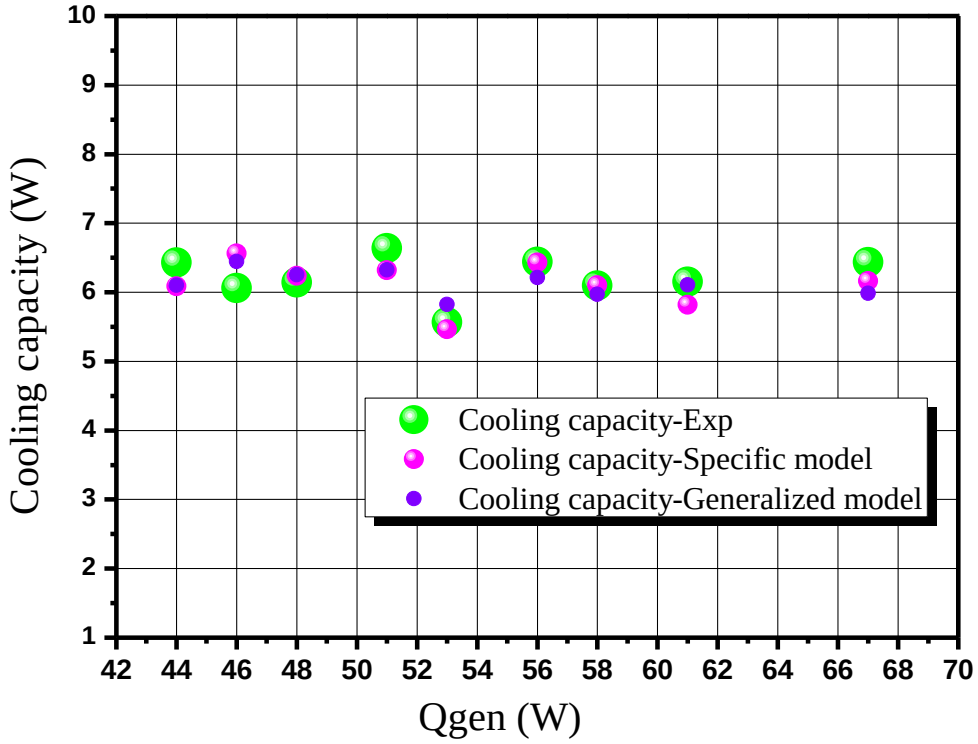

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