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# Numerical study of air temperature distribution and refrigeration systems coupling for chilled food processing facilities

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**Abstract**

This paper presents an air temperature distribution and refrigeration system dynamic coupling model to assess the performance of air distribution systems used in chilled food processing areas and its energy consumption impact. The coupling consists of a CFD air flow/temperature distribution system model and a compression refrigeration system model developed in EES integrated in the TRNSYS platform. The model was tested and validated using experimental data collected from a scaled air distribution test rig in an environmental chamber showing a good agreement with the measured data (an hourly energy consumption error up to 5.3 %). The CFD/EES coupling model can be used to design energy efficient cooled air distribution systems capable to maintain the required thermal environment in chilled food processing facilities.


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*Keywords:* Air distribution systems; Chilled food factories; Refrigeration; CFD; EES; TRNSYS

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**Nomenclature**

| | |
|---|---|
| q | Heat flow [$W.m^{-2}$] |
| Ui | Overall heat transfer coefficient [$W.m^{-2}.K^{-1}$] |
| $\Delta T$ | Temperature difference [K] |
| P | Power consumption (kWh) |

## 1. Introduction

The UK chilled food market is one of the most highly developed in the world. Its growth over the last few years has been massive. According to the UK Chilled Food Association the prepared chilled food marked in the UK has grown from £7357 m in 2005 to its current £12280 m in 2015 [1]. Chilled food chain relies heavily on refrigeration for the maintenance of low temperatures during processing, transportation, and retail of chilled food products. In the UK, refrigeration systems in the cold food chain are estimated to be responsible for 16,100 GWh energy use and 13.7 $MtCO_2e$ Greenhouse Gas Emissions in 2010. This account for approximately 28 % of final energy use and 7 % of GHG emissions of the whole food-chain for 2010 [2]. Chilled food products have short shelf lives and processing takes place in facilities that are normally maintained at temperatures in the range between +4 to +12 ºC depending on the type of product, processing time and the desired minimum shelf time. Chilled food processing takes place in large spaces with high ceilings where cooling is normally provided by ceiling mounted fan coil units, air socks or diffusers. For the system to be effective, large air circulation rates and air velocities are required which, combined with the low temperatures cause high energy consumption and in some cases discomfort for the workers in the space. Therefore, the air distribution is an essential factor that needs to be carefully considered in order to create an environment capable of maintaining food quality without excessive worker discomfort. The air distribution system should create a temperature and humidity homogeneity around the food product to maintain its quality.

Air distribution modelling techniques have been developed to provide a better understanding of air flow patterns and temperature distribution in large spaces including cold rooms. The most powerful of these is the Computational Fluid Dynamics (CFD). Experimental results can be initially used for the CFD model validation. Once the CFD model is validated, it can be used to investigate different air distribution systems under different cooling conditions. This can lead to a better understanding of the air distribution in the space and optimum designs to improve product quality. Gowreesunker et al. [3] investigated numerically the energy performance and indoor environmental control of a displacement diffuser in an airport terminal space. A coupled TRNSYS-FLUENT model was developed and used to predict the performance of the building under two different control strategies. TRNSYS was used to simulate the air conditioning system, while FLUENT was used to simulate the indoor airflow and radiation. Ambaw et al. [4] reviewed the application of CFD for the modelling of post-harvest refrigeration processes. They identified the most common solution method to be the finite volume method with the upwind differencing scheme. In addition, it was reported that the Reynolds Stress Model (RSM) provides more accurate predictions compared to the conventional k-ε model but the k-ε model is more commonly used due to its lower computational requirements. Laguerre et al. [5] and Duret et al. [6], in order to avoid the computational time of a CFD model, created a simplified model using the knowledge obtained from experimental measurements. The model was separated into zones and heat balance equations for each zone were developed. The simplified model was found to predict the product cooling rate and the final product temperature at different positions in the cold room quite well. Delele et al. [7] developed a 3-D model in CFD in order to predict air-flow and heat transfer characteristics of a horticultural produce packaging system. In contrast to previous studies which considered the bulk of the product as a porous media due to limitations in computational power and time, Delele's study considered the detailed geometry and properties of the packaging material. The air flow in the space was solved using the Reynolds averaged Navier Stroke equations (RANS). The authors applied a transient simulation with a time step of 180 s (50 iterations per time step) and the governing equations were discretized using a second order upwind scheme. The SST k-ω was found to produce the most accurate predictions. This paper presents the evaluation of an integrated CFD air distribution and EES refrigeration systems model. The model was tested and validated using experimental data collected from a scaled air distribution test rig in the environmental chamber.

## 2. Procedure

The scope of this research aims to improve the efficiency of cold air-temperature distribution in chilled food processing areas. Improved air-temperature distribution should lead to the reduction of the overall energy consumption of the refrigeration plant. The first stage of this research focused on understanding the air flow and the temperature variation in existing chilled food production facilities [8]. In addition, the second stage of this research focused on the air temperature distribution evaluation in a scaled experimental test rig representing a section of the monitored chilled food processing facility in an environmental chamber [9]. The present stage of this research deals with the development of an CFD-EES coupling model to predict the performance of the refrigeration system under different conditions and air distribution systems. After the individual validation of the CFD air distribution model and the EES refrigeration model under steady state conditions, a transient simulation is implemented. The transient simulation success is evaluated through the integration of the CFD air distribution model and the EES refrigeration model.

### *2.1. Experimental test facility*

The experimental test rig was established using an environmental chamber constructed with insulated cold room panels. The chamber dimensions are 2.9 m (H) x 6.6 m (L) x 3.5 m (W). Cooling in the chamber was provided by an evaporator coil served by a R404a condensing unit situated outside the test rig in the ambient air. A schematic diagram of the test rig is shown in Fig. 1.

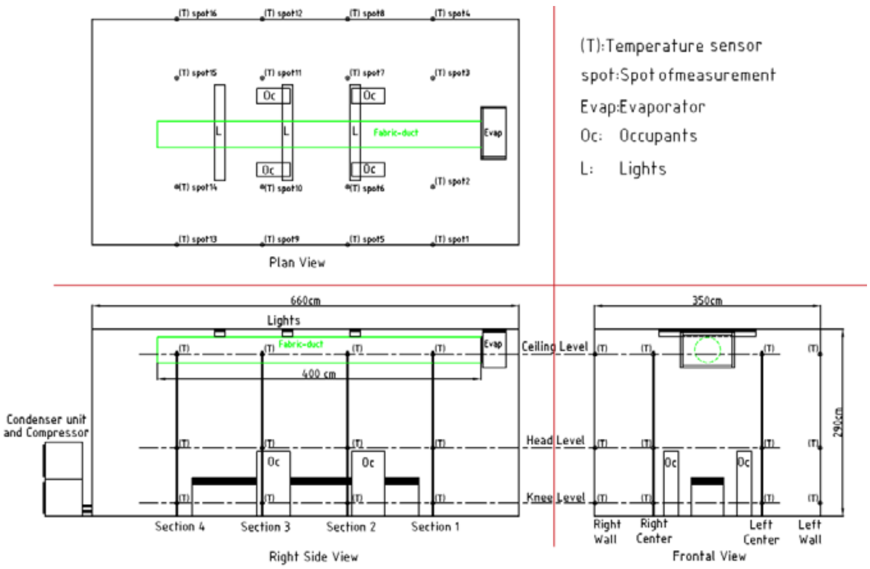


Fig. 1. Outline of the experimental test facility with air distribution via fabric duct at ceiling level, reference case.

The evaporator distributes the conditioned air into the space through a round fabric duct (measuring 400 mm in diameter and 4 meter in length) installed at ceiling level in the experimental facility as shown in Fig. 2. This configuration corresponds to an actual air distribution system installed in a chilled food processing factory. The fabric duct that was used for the experimental set-up was fabricated with the ‘KE - Low Impulse textile duct fabric’ of Ke-fibertec. The air temperature in the test chamber was controlled using a temperature controller with the thermostat located on the evaporator air suction side and set to 9.7 ºC. Also, a variable speed controller was used to control the supply air. Air temperature measurements in the chamber, using T-type thermocouples, were taken at 4 sections along the length of the chamber (see Fig. 1) and 4 sections along the width of the chamber (Right-wall, Right-centre, Left-centre and Left-wall) and at three heights, knee level, head level and ceiling level. Temperatures were also measured at the inlet and outlet of each component in the refrigeration system. In total, 58 temperature sensors were installed.

The thermal load from occupants (OC) was simulated by 4 rectangular boxes of 1.6 $m^2$ surface area each wrapped with trace heater elements, 150 W each. Air velocity measurements were also taken close to the position of each temperature sensor using an air flow meter TSI TA465-P with a thermo-anemometer probe 966. The refrigerant mass flow rate was measured using a Coriolis type mass flow meter Krohne Optimass 7300 C placed at the outlet of the condenser. The power consumption of the refrigeration system was recorded by a portable power metre Fluke 435 Series II while datascan modules 7020 were used for the temperature and refrigerant flow data logging. The logging interval of the data was set to 10 seconds. The time for each test was 17 hrs.

Fig. 2. Experimental test facility with air distribution via a round fabric duct.

## 2.2. CFD air-distribution model

A steady state 3-D CFD model was solved using the commercial ANSYS FLUENT® package. The dimensions of the developed 3-D CFD air distribution system model shown in Fig. 3a correspond to the actual size of the experimental test facility. The air supply from the air-sock and return air at the coil boundary conditions were defined as mass flow inlets and outlets, respectively. The supply and return air flow from the evaporator was set at 2825 $m^3hr^{-1}$. The air supply from the supply and return air sections at the coil boundary conditions were defined as mass flow inlets and outlets, respectively. The air supply temperature was set at 7 ºC.

The occupancy density of the investigated processing area was set at 4 occupants taking into account the scaled model area. Each occupant was defined as a rectangular box with a 1.57 $m^2$ surface area (1.2 m height) with a sensible thermal load of 150 W. The lighting heat load was set at 30 $Wm^{-2}$ floor area. Other heat sources into the processing area were neglected. The thermal boundary conditions of the surrounding walls were approached taking into account the thermal resistances (Fig. 3b) and heat flow (Eq. 1) from the exterior to the interior of the chamber, and considering an outdoor temperature of 20 ºC. The wall thickness was 0.1 m while the thermal conductivity of the wall was 0.023 $W.m^{-1}.K^{-1}$. The exterior and interior wall surfaces resistances were 0.13 $m^2.K.W^{-1}$ and 0.04 $m^2.K.W^{-1}$, respectively.

$$q = Ui \times \Delta T \ (\frac{W}{m^2}) \quad (1)$$

(a) (b)

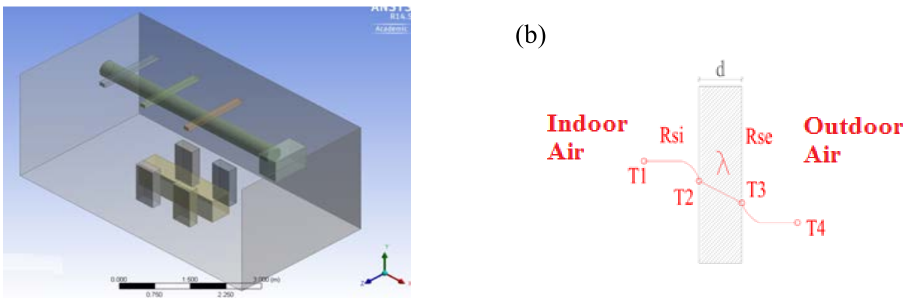


Fig. 3. (a) 3-D CFD model and (b) Heat flow through the walls.

The model was solved with the pressure based solution algorithm, second order upwind energy and momentum discretisation, 'Body-Force' weighted pressure discretisation, and SIMPLE pressure-velocity coupling. The air inside the processing area was considered compressible and the density was allowed to vary according to the ideal gas law to account for buoyancy effects. Other thermal properties (such as specific heat, thermal conductivity, and viscosity) were maintained constant. The SST-k-ω turbulence model developed by Menter [10] was selected since it was found to predict the measured data with better accuracy [9]. The computational domain was discretized with an automatic mesh method, with mainly tetrahedral cells (with hexahedral cells in the boundary layer). The mesh density was gradually refined near the building wall, the fabric duct and other surfaces in the space. The final mesh size consisted of 9.6 million elements. The element dimensions varied between 0.02-0.06 m (0.02 for lighting, occupants and sock. 0.04 for evaporator outlet. 0.06 for productions evaporator body and walls). The finer mesh sizes were located near the wall surfaces, where 4 inflation layers were also employed to capture the effects of the boundary layer. The final model mesh was generated following a mesh independency study. The simulation time was 8 hours, with an average of 3000 iterations, on a 2.5GHz, 64GB RAM, Intel Xeon Processor (2 processors) with 48 parallel threads. More details of the experimental test facility and CFD modeling procedure can be found in [9].

## *2.3. EES refrigeration system model*

The numerical model developed for the vapour compression refrigeration system is mainly based on correlations for prediction of the thermodynamic properties of the refrigerant R404a [11], and heat transfer correlations for boiling mode [12,13] and condensation mode [14]. The resolution procedure follows a step by step process in which energy balances in every component are solved sequentially. Convergence is obtained when the sub-cooling criterion is satisfied and the calculated refrigerant temperature at the evaporator inlet is in equilibrium with the energy balance on this component. The software used was the Engineering Equation Solver (EES). Validation was conducted taking into account the data collected from the test facility room. The model inputs are the evaporator fan velocity, ambient temperature, inlet air-side temperature to the evaporator, evaporator thermal load, Super-heating, sub-cooling and the index n for the compressor. Parameters compared are as follow: Refrigerant flow rate (m_refrig), air-side temperature at the evaporator outlet (Te,air,out), condenser load (Qcond) and electrical instantaneous power to the compressor (Inst. power). Table 1 shows a resume of the model validation.

Table 1. Experimental measurements and EES model results comparison.

| Runs | Fan Vel % | m_refrig (kg/h) | | | Te,air,out (ºC) | | | Qcond (kW) | | | Power (kW) | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Exp. | Model | Error | Exp. | Model | diff. | Exp. | Model | Error | Exp. | Model | Error |
| 1 | 70 | 153,5 | 146,7 | 4,4% | -3,3 | -4,4 | 1 | 7,8 | 7,4 | 4,9% | 3 | 2,5 | 14,8% |
| 2 | 80 | 145,1 | 138 | 4,9% | -2 | 1,1 | 3,1 | 7,4 | 6,6 | 11,5% | 3 | 2,7 | 8,2% |
| 3 | 90 | 145,8 | 128,9 | 11,6% | 0,7 | 3,2 | 2,5 | 7,1 | 6,3 | 11,6% | 3 | 2,5 | 18,1% |
| 4 | 100 | 148,7 | 144,7 | 2,7% | 8,5 | 3,5 | 5,0 | 7,5 | 6,9 | 7,3% | 3 | 2,9 | 3,3% |

## *2.4. CFD/EES dynamic model*

As described in section 2.2, CFD modelling was used in order to simulate the air-distribution in the space, whereas the EES model was developed to simulate the refrigeration system. The dynamic coupling was achieved within TRNSYS platform. The TRYNSYS platform was used to control the simulation procedure and exchange data between CFD and EES at the end of each time step. In order to succeed that, a TRNSYS component was programmed in FORTRAN and compiled in TRNSYS. In addition, TRNSYS-EES coupling was achieved through data exchange via clipboard as indicated in Fig. 4. TRNSYS controls the procedure, calling and exchanging data between CFD and EES for each time step until the simulation duration is completed. Within the simulation duration, TRNSYS calls CFD to model the air distribution in the space. Once the CFD modelling is finished, TRNSYS calls EES to model the refrigeration system. If the room temperature is higher than the set point, EES calculates the new supply temperature and the work done form the refrigeration system. Otherwise, EES indicates that the work done from the refrigeration system equals with zero and the new supply temperature equals with the room temperature. In this way, a loop is implemented for the selected simulation duration.

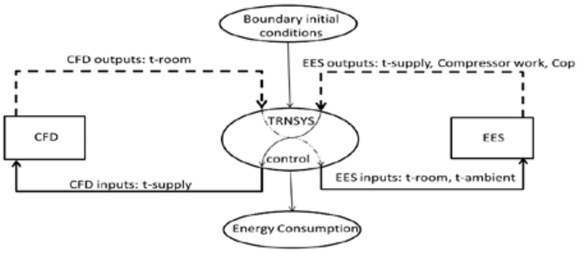


Fig. 4. CFD-TRNSYS-EES dynamic simulation data exchange.

## 3. Results and discussion

### *3.1. CFD model results and validation*

Fig. 5 shows the velocity and temperature distribution at 4 cross sections along the space. Air velocities were very low and varied between 0.01 and 0.3 m.s$^{-1}$. Regarding the air temperature distribution over the test facility, with a supply temperature from the air sock at 7 ºC, the temperature in the space varied between of 8.0 ºC and 13.4 ºC. Temperature stratification was observed, with lowest temperatures measured at knee level and highest at ceiling level. To determine the validity of the model, the temperature predictions from the model were compared against temperature actual measurements obtained from the experimental facility. Fig. 6 presents the average temperature data collected including the measurement uncertainty and the modelling results. In general, the model shows a good level of prediction for the air temperature trend and distribution achieved in the space. From all results, the average absolute error across all test points was found to be 0.95 ºC.

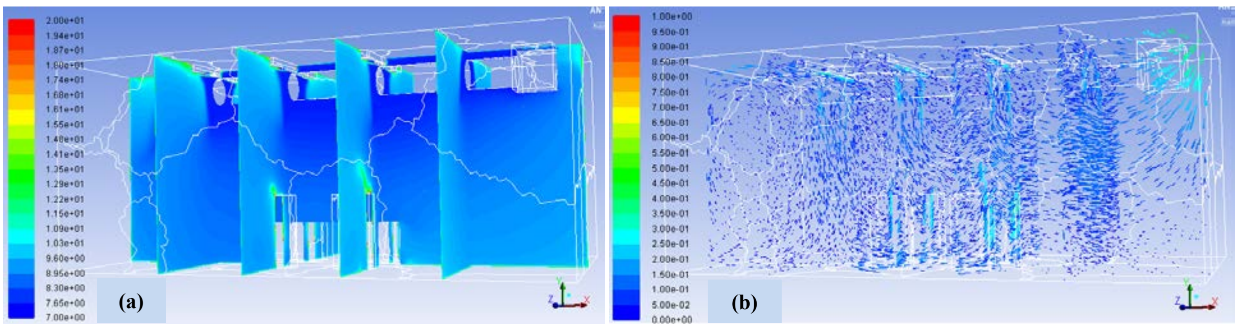


Fig. 5. CFD Simulation results of (a) air temperature in the space and (b) air velocity (m.s$^{-1}$).

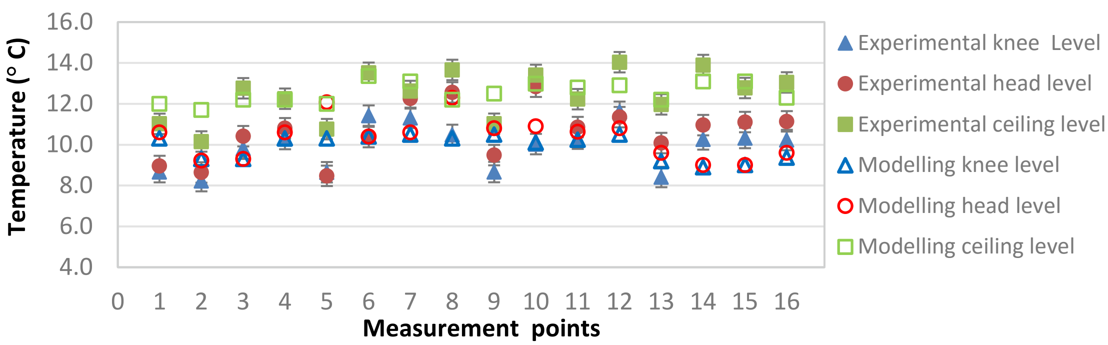


Fig. 6. Experimental and CFD modelling air temperature comparison.

### *3.2. CFD/EES coupling model results*

The CFD-EES coupling model was validated considering the power and the air temperature measurements that were carried out in the experimental chamber. Due to high computational time, the transient simulation was conducted for 1 hour of real time operation. Fig. 7 show a comparison between predicted and experimental values for the air supply temperature, room temperature (return temperature) and instant power consumption. Modelling results show that supply and return air temperatures fluctuate over time from 5 °C to 12 °C and from 9.5 °C to 12 °C respectively, whereas experimental values show that supply and return air temperatures fluctuate from 4 °C to 11.5 °C and from 9.0 °C to 12 °C respectively. In addition, the predicted instant power consumption estimated at 2.95 kW shows a good agreement compared with the experimental instant power consumption (Table 2).

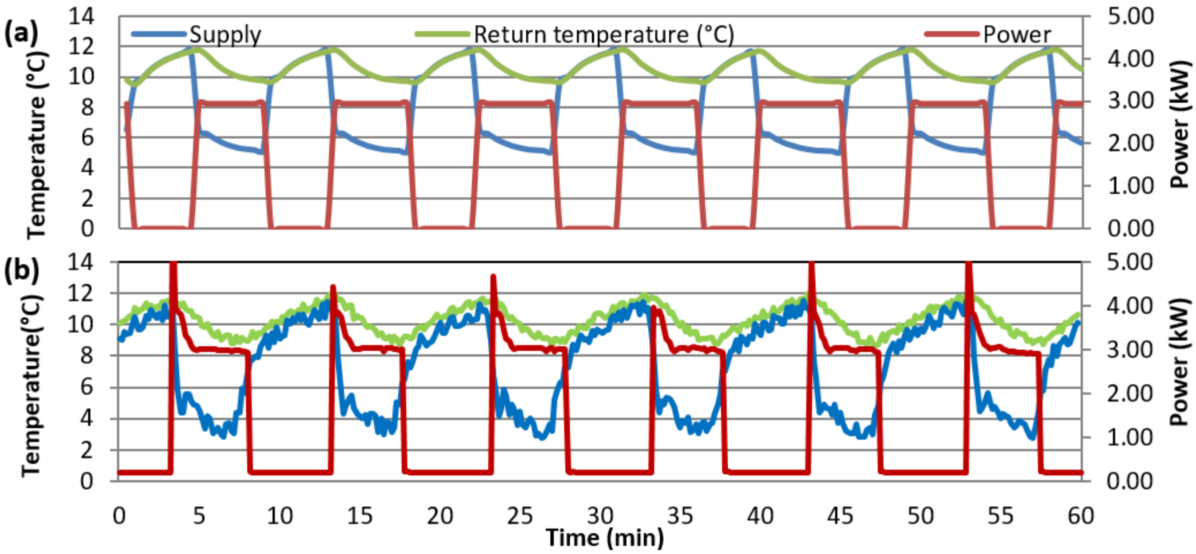


Fig. 7. Dynamic Model Validation: (a) CFD/EES Modelling Results (b) Experimental Values.

The power consumption (P) of the refrigeration system was estimated by considering the sum of the instant power consumption of the refrigeration system during the running period ($P_{on}$) and the energy consumption of the condenser and evaporator fans and control system during the Stop period ($P_{off}$). Data collection was every 10 seconds.

$$P = \sum P_{on} + \sum P_{off} \tag{2}$$

Comparing the results, the number of operation cycles and their duration are in a good agreement. Table 2 shows that predicted hourly consumption was 1.57 kWh whereas experimental measurements showed an average hourly power consumption at 1.49 kWh. In general, the integrated CFD/EES model modelling showed a good agreement with the measured data.

Table 2. CFD/EES coupling model validation.

| 1 Hour running time | | | | | |
|---|---|---|---|---|---|
| | Experimental | | Modelling | | Error |
| Number of on Cycles | 6 | cycles | 6.3 | cycles | |
| Refrigeration System ‘on’ Period | 4.26 | minutes | 4.5 | minutes | 5.6 % |
| Hourly Consumption (kWh) | 1.49 | kWh | 1.57 | kWh | 5.3 % |
| Instant Power Consumption during operation (kW) | 3.22 | kW | 2.9 | kW | 9.9 % |

## 4. Conclusion

This paper outlines research that aims to investigate and improve the efficiency of air distribution and temperature control systems in chilled food manufacturing facilities. A scaled experimental test rig was developed to represent an existing chilled food facility.

A 3-D CFD model was developed, validated, and used to predict the air and temperature distribution of different air distribution systems over the scaled facility.

An EES simulation model capable to predict the energy consumption and the refrigeration cycle characteristics of the system was developed.

Both the CFD and EES refrigeration system models can be used individually to optimise the air distribution and refrigeration system respectively. In order to predict the energy impact of each different air distribution systems, CFD air distribution model and EES refrigeration system model was integrated.

A CFD/EES dynamic coupling model was developed with the usage of TRNSYS platform and FORTRAN programming. The integrated CFD/EES model was validated against experimental results showing an hourly energy consumption error up to 5.3 % with respect to the measured data.

The integrated CFD/EES model can be used to design an optimum energy efficient air distribution system that will create an environment capable to maintain the food product quality at production facilities.

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