DOI: 10.21134/a2j1s181

# ANALYSIS OF INFLUENCING PARAMETERS ON AN OFF-DESIGN INDIRECT EVAPORATIVE COOLING MODEL

**Manuel Andrés-Chicote[1]*, Ana Tejero-González[1] and Eloy Velasco-Gómez[1]**

1: Departamento de Ingeniería Energética y Fluidomecánica, Grupo de Termotecnia, ITAP, Escuela de Ingenierías Industriales, Universidad de Valladolid.
Paseo del Cauce, 59, 47011, Valladolid (Spain)
e-mail: manuel.andres.chicote@uva.es

**Abstract:** *Evaporative cooling has been considered for sustainable and efficient air-conditioning since ancient times. Nowadays, improved concepts have led to a large variety of applications from direct conditioning of indoor and outdoor environments up to sensible air-cooling through different configurations of Indirect Evaporative Cooling (IEC) schemes. However, better understanding and mathematical modelling of the process is still needed for design and operational optimization of such systems. This work addresses a MATLAB implementation of an off-design model for a crossflow IEC unit based on previous literature work. Then, multiple parametric runs are conducted to identify the influence of critical process characteristics and modelling aspects such as heat and mass transfer coefficients and surface wettability. Results provide insight to further modelling and simulation research, also extending the approach to other system designs and flow configurations.*

**Keywords:** Indirect Evaporative Cooling, Off-design model, Heat and mass transfer, surface wetability, HVAC.

## 1. INTRODUCTION

Evaporative cooling has been considered for sustainable and efficient air-conditioning since ancient times [1]. Nowadays, improved concepts have led to a wide variety of applications, ranging from direct conditioning of indoor and outdoor environments [2] up to sensible air-cooling achieved through different configurations of Indirect Evaporative Cooling (IEC) schemes [3]. Among these, applications to the rapidly growing data centre sector have recently gained interest, incorporating strategies for free cooling and evaporative cooling systems in pursuit of improved energy efficiency [4],[5].

Moreover, despite extensive research and application of evaporative cooling in the past, there remains a need for a better understanding and mathematical modelling of the process to optimize the design and operation of specific systems. Modelling of IEC systems is particularly challenging, due to the multiple possible combinations of components and design aspects (e.g. flow configurations, unit dimensions, type of materials or water distribution systems, among others), which influences the heat and mass transfer processes. These processes occur due to the interaction of two air streams and a wetted surface: a primary or process air stream flowing through dry channels of a heat exchanger, undergoing a sensible cooling, and a secondary air stream flowing through adjacent wet channels).

In this context, [6] presented a comprehensive compilation of different modelling approaches for IEC systems from previous literature, encompassing both analytical and numerical studies. It is revealed that models for various system concepts exist, and multiple approaches are possible with performance prediction errors generally falling within +/- 10%. Furthermore, considering parameters such as the wettability factor of the wet channel surface can improve result accuracy.

This work proposes and analyses a numerical model for a cross-flow Indirect Evaporative Cooling unit, building upon previous studies [7] and [8]. It examines the sensitivity of results to off-design operational conditions and modelling assumptions, with a particular focus on the wettability factor.

## 2. DESCRIPTION OF THE IEC SYSTEM AND MATHEMATICAL MODEL

The objective of this work is to propose a reference numerical model for IEC systems aiming at balancing model complexity (or computational effort) and model accuracy, as well as at facilitating the adaptation of the modelling approach to other flow configurations and unit dimensions. For this purpose, a reference IEC system from De Antonellis et al. [8] has been considered. It consists of a crossflow plate heat exchanger (HE) with 119 square plates (470 x 470 mm), accounting for 0.14 mm thickness each, 3.35 mm pitch and 220 $W \cdot m^{-1} \cdot K^{-1}$ thermal conductivity. This leads to a net face HE cross area of 0.089 $m^2$. The water supply for the wet channels is provided from the secondary air inlet section (in parallel flow with the air stream).

The proposed numerical model is based on a 2D spatial discretization of the HE plates surface, together with a Finite Difference Method (FDM) implementation of the heat and mass transfer governing equations for two adjacent channels on the plate heat exchanger under steady-state conditions: a dry channel for the primary air stream and the adjacent wet channel for the secondary air stream.

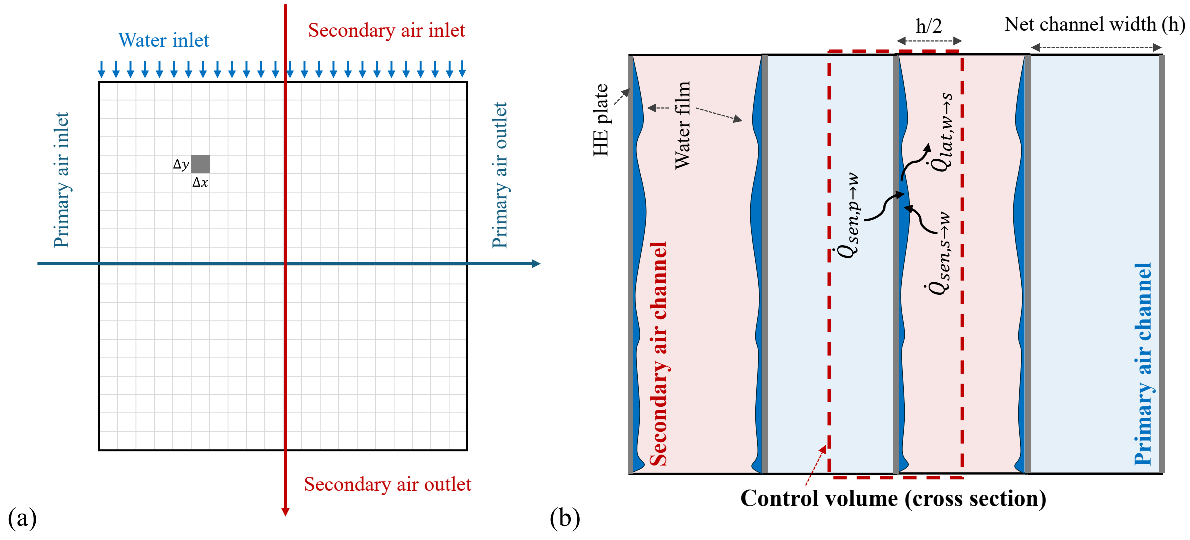



Figure 1. (a) 2D view of the IEC unit plates; (b) detailed view of 2 adjacent air channels.

Figure 1(a) shows a 2D representation of the discretized IEC unit plates including air and water flows. Figure 1(b) shows a detailed view of two adjacent channels including the interfacing HE plate and water film, as well as the relevant heat and mass transfer flows.

The governing equations included in the model are:

$$\frac{dT_p}{dx} = \frac{U_{T,p}\left(T_w - T_p\right)}{\rho_a v_{N,p} c_{p,p} \frac{h^*}{2}} \quad (1) \qquad \frac{dT_s}{dy} = \frac{h_{T,s}(T_w - T_s)}{\rho_a v_{N,s} c_{p,s} \frac{h^*}{2}} \quad (2)$$

$$\frac{dX_s}{dy} = \frac{h_{M,s}(X_w - X_s)\sigma}{\rho_a v_{N,s} \frac{h^*}{2}} \quad (3) \qquad \frac{d\dot{m}_w}{dy} = \frac{h_{M,s}(X_s - X_w)\sigma}{h^*/2} \quad (4)$$

$$h_{M,s}\left(\lambda + c_{p,s} T_s\right)(X_s - X_w) + h_{T,s}(T_s - T_w) + U_{T,p}\left(T_p - T_w\right) = 0 \quad (5)$$

The following modelling assumptions, which are consistent with those detailed in [8], are also adopted:

- No heat losses to the surroundings
- Negligible axial heat conduction and water diffusion in the air streams
- Negligible heat conduction in the HE plates
- Uniform inlet conditions for the primary and secondary air streams
- Homogeneous temperature within the water film
- Constant Lewis number
- Use of a calibrated correlation for the convective heat transfer coefficients

On the other hand, differences in respect to [8] involve:

- The effect of a direct evaporative pre-cooling of the secondary air stream due to partial evaporation of the water sprayed at the inlet section is not modelled. Depending on the water distribution system, this effect could be modelled apart (e.g. based on saturation efficiency equations) and the corresponding secondary air inlet conditions could be updated accordingly within the present model.
- Heat and mass transfer equations have been revised and corrected from [8] to avoid the contribution of the latent heat transfer to the sensible energy balance, which is linked to the secondary air dry-bulb temperature variation along the HE.
- A surface wettability factor (accounting for the fraction of total plate area that is actually wetted) is considered in the governing equations as suggested by [7] and [8]. However, different approaches are considered for the modelling of this factor. A reference scenario will be studied based on the calibrated correlation provided by [8] depending on the water specific mass flow rate and the secondary air nominal velocity. Other scenarios will consider constant values of the wettability factor ranging from 0 to 1.

Under these considerations, the FDM scheme was implemented in MATLAB following an explicit method based on forward differences for the primary air dry-bulb temperature, the secondary air dry-bulb temperature and humidity ratio, the water mass flow rate, alongside with an energy balance at each surface element.

## 3. DEFINITION OF TEST CASES AND PARAMETRIC RUNS

The proposed model was applied to a reference test case based on the IEC unit described in Section 2 operating under a wide range of off-design conditions. Three different analyses were conducted, always keeping the primary air inlet fixed at relevant conditions for process air in data centre applications (i.e. dry-bulb temperature = 35ºC; relative humidity = 30%).

**Study 01**: First, nominal primary and secondary air flow rates of 1800 $m^3/h$ in normal conditions ($v_{N,p} = v_{N,s} =$ 5.67 m/s) were fixed and the secondary air stream inlet was varied from 5 to 25 g/kg humidity ratios and from the saturation condition (at each humidity ratio level) to 45ºC air dry-bulb temperature. Water supply mass flowrates were varied from 0 (dry operation) to 120 kg/h (0.45 $kg_{d.a.}/(m^2s)$).

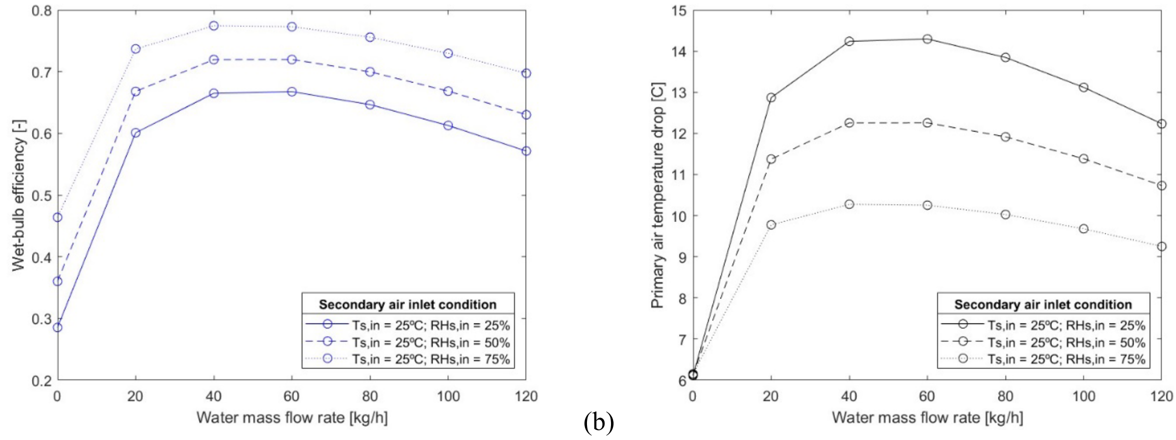
**Study 02**: Then, for a reference scenario with secondary air conditions fixed at 25ºC dry-bulb temperature and 40% relative humidity, primary and secondary air flowrates (in normal conditions) were varied from 600 $m^3/h$ ($v_{N,s}$ = 1.87 m/s) to 3000 $m^3/h$ ($v_{N,s}$ = 9.36 m/s), while water mass flowrates were varied from 0 to 140 kg/h.

**Study 03**: Finally, the same parametric runs from Study 01 were calculated under the consideration of constant wettability factors ($\sigma$=0.2 and $\sigma$=1).

## 4. RESULTS

Results from Study 01 reveal that the model is capable to capture the performance sensitivity to the variation of the water mass flowrate supplied to the secondary air channels. Figure 2 represents wet-bulb efficiencies, $\varepsilon_{wb}$ (see Eq. (6)) and primary air temperature drops for different secondary air inlet humidities at $T_{s,in}$ = 25ºC. It is observed that optimal performance occurs for a water mass flowrate range of 40-60 kg/h in this IEC unit.

$$\varepsilon_{wb} = \frac{T_{p,in} - T_{p,out}}{T_{p,in} - T_{wb,s,in}} \qquad (6)$$




(a) (b)

Figure 2. Wet-bulb efficiency (a) and primary air temperature drop (b) for different water mass flowrates and three different secondary air inlet conditions with $T_{s,in}$ = 25ºC.

According to Figure 2 wet-bulb efficiency increases, and primary air temperature drop decreases with increasing humidity levels. This happens for a constant $T_{s,in}$ of 25ºC. Figure 3 and 4 show the values of these performance parameters for the whole range of the studied psychrometric conditions. Wet-bulb efficiency results reveal higher values for secondary air inlet conditions closer to the saturation line (i.e. decreasing evaporative cooling potential) and decreasing, but positive, sensible cooling potential ($T_{p,in} - T_{s,in}$).

However, primary air temperature drops increase for colder and drier secondary air inlet conditions according to Figure 4. These results are in line with those presented in [8] but provide slightly smaller temperature drops due to the presented model adaptations and the disregarded pre-cooling effect of direct evaporative cooling produced by sprayed water supply before the entrance of the secondary air stream to the heat exchanger (HE).

Heat transfer coefficients obtained for Study 01 tests reveal almost constant values around 85 $W \cdot m^{-2} \cdot K^{-1}$.

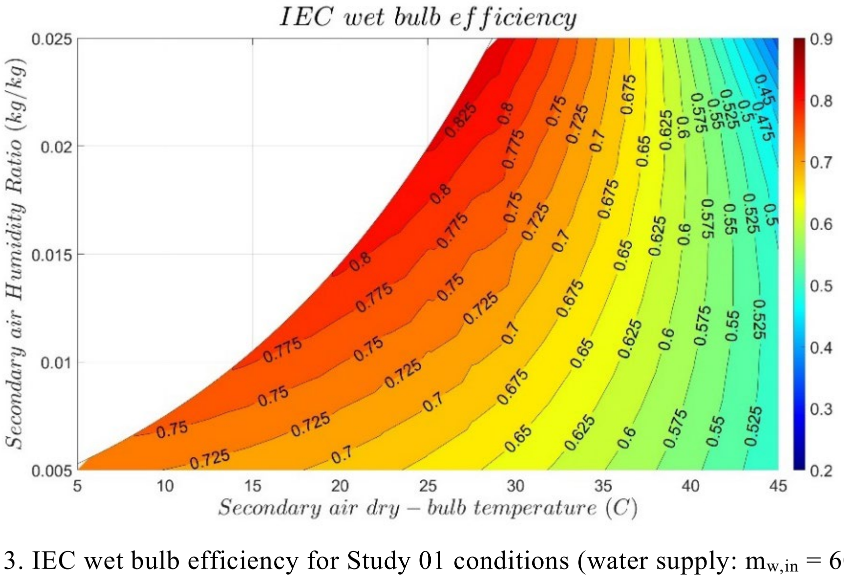



Figure 3. IEC wet bulb efficiency for Study 01 conditions (water supply: $m_{w,in}$ = 60 kg/h)

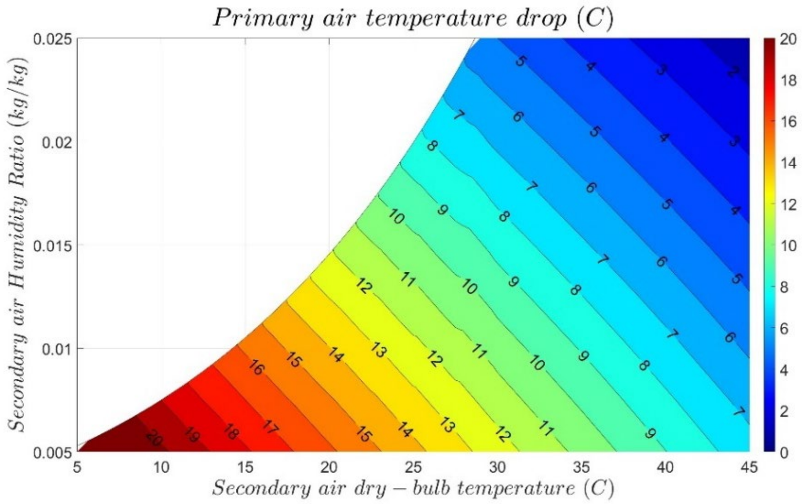
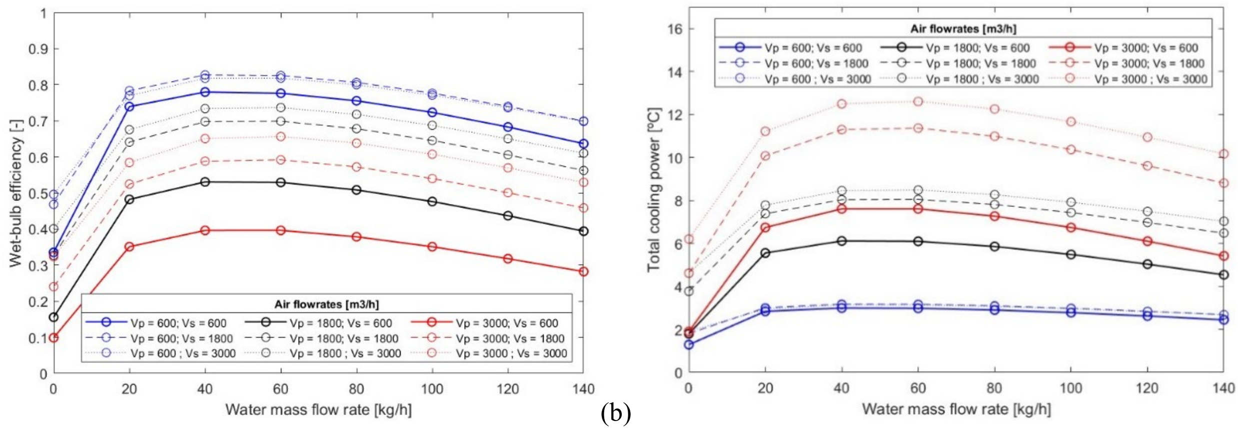



Figure 4. Primary air temperature drops for Study 01 conditions (water supply: $m_{w,in}$ = 60 kg/h)

Results from Study 02 show how the variation of the air flowrates modifies the thermal performance of the IEC unit. Wet-bulb efficiency values increase with decreasing primary air flowrates and increasing secondary-to-primary air ratios (Figure 5(a)). As expected, the total cooling power of the IEC system increases with increasing primary air flow rates and increasing secondary-to-primary air flow ratios (Figure 6(b)). Heat transfer coefficients obtained for Study 02 tests reveal a dominant dependence with air flow velocities with values ranging from 30 $W\cdot m^{-2}\cdot K^{-1}$ (at $v_N$ = 1.87 m/s) up to 137 $W\cdot m^{-2}\cdot K^{-1}$ (at $v_N$ = 9.36 m/s)




(a) (b)

Figure 5. Influence of air flowrate variations on the system's wet-bulb efficiency (a) and total cooling power (b)

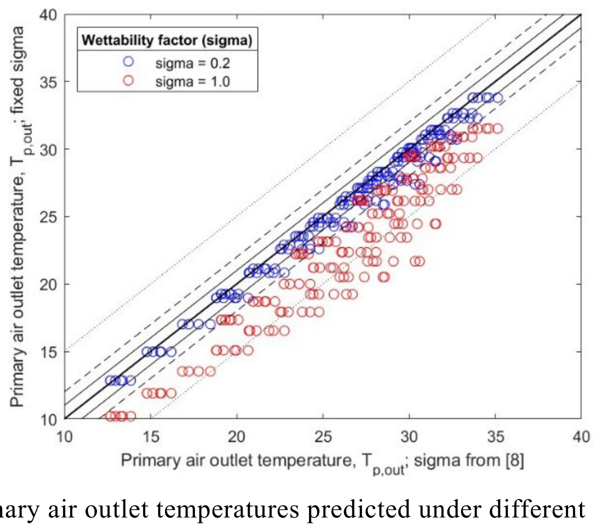



Figure 6. Comparison of primary air outlet temperatures predicted under different modelling assumptions for the wettability factor: model from [8] (x-axis); constant wettability factor (y-axis)

Results from Study 03 enable to preliminary evaluate the effect of surface wettability modelling into the overall IEC unit model. Figure 6 shows primary air outlet temperature results derived from considering the wettability factor model described in [8] compared to results based on constant wettability factor considerations. It is evident that assuming a perfectly wetted secondary air channel surface leads to relevant underprediction of primary air outlet temperatures. However, maintaining a constant wettability factor of 0.2, regardless of the air conditions and water mass flow rate, results in minor deviations (+/- 1ºC), while also reducing the modelling effort. Nonetheless, further analyses and validation tests are required to confirm the potential benefits of this proposed model simplification.

## 5. CONCLUSIONS AND FUTURE WORK

This work presented a 2D off-design model of an Indirect Evaporative Cooling system based on the numerical solving of the heat and mass transfer governing equations. Previous work from literature was taken as reference, revised, and adapted for the purpose. The IEC unit performance was evaluated for a typical primary air condition in data centre applications and a wide variety of secondary air temperature and humidity values. Air flow rates were varied to off-design conditions observing a dominant dependence of heat and mass transfer coefficients with air velocities. The use of a constant wettability factor is suggested for model simplification.

Results are aligned with those from literature, but detailed validation is needed. Results also provide insight to further modelling and simulation research, extending the approach to other system designs and flow configurations (e.g. counterflow IEC, regenerative IEC schemes, dew-point evaporative coolers, etc.)

### Nomenclature

| | | | |
|---|---|---|---|
| $c_p$ | Specific heat capacity [in kJ/kg] | $\lambda$ | Latent heat of vaporization [in kJ/kg] |
| $h^*$ | Air channel width [in m] | Subscripts and superscripts | |
| $h_T$ | Heat transfer coefficient [in $Wm^{-2}K^{-1}$] | $p$ | Primary air |
| $h_M$ | Mass transfer coefficient | $s$ | Secondary air |
| $\dot{Q}$ | Energy flow [in kW] | $w$ | water (film) |
| $\rho$ | Densidad [in kg(m$^3$] | $sen$ | sensible |
| $U$ | Global heat transfer coefficient [in $Wm^{-2}K^{-1}$] | $lat$ | latent |
| $v$ | Velocity [in m/s] | $N$ | Normal conditions |
| $X$ | Specific humidity (humidity ratio) [in kg/kg] | $a$ | air |
| $\sigma$ | Surface wettability factor [-] | in | inlet |
| $T$ | Dry-bulb temperature [in ºC] | out | outlet |

### Acknowledgements

This work has been developed within the research project TED2021-129652A-C22, funded by MCIN/AEI/10.13039/501100011033 and the European Union through the "NextGenerationEU"/PRTR.

## REFERENCES

[1] J.R. Watt, Evaporative Air Conditioning Handbook, Springer, 1st ed., New York, NY, 2011.
[2] Tejero-González, A., Franco-Salas, A. Direct evaporative cooling from wetted surfaces: Challenges for a clean air conditioning solution. WIRE: Energy and Environment, 2022, 11(3), e423.
[3] Duan, Z., Zhan, C., Zhang, X., Mustafa, M., Zhao, X., B., Hasan, A. Indirect evaporative cooling: Past, present and future potentials. Renewable and Sustainable Energy Reviews, 2012, 16(9), 6823-6850.
[4] Isazadeh, A., Ziviani, D., & Claridge, D. E. Cooling technologies in datacom facilities: An overview and perspectives. 19th Int. Refrigeration and Air Conditioning Conference, Purdue, USA, 2022.
[5] Chu, J., Huang, X. Research status and development trends of evaporative cooling air-conditioning technology in data centers. Energy and Built Environment, 2023, 4(1), 86-110.
[6] Caruana, R., De Antonellis, S., Marocco, L., Guilizzoni, M. Modeling of Indirect Evaporative Cooling Systems: A Review. Fluids, 2023, 8(11), 303.
[7] Chengqin, R., Hongxing, Y. An analytical model for the heat and mass transfer processes in indirect evaporative cooling with parallel/counter flow configurations. International Journal of Heat and Mass transfer, 2006, 49(3-4), 617-627.
[8] De Antonellis, S., Joppolo, C. M., Liberati, P., Milani, S., Romano, F. Modeling and experimental study of an indirect evaporative cooler. Energy and Buildings, 2017, 142, 147-157.