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NUMERICAL STUDY OF HEAT TRANSFER AND FLUID FLOW IN LARGE-SCALE SPRAY COOLING SYSTEMS; EXAMPLE OF MINE-BULK-AIR COOLER

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ABSTRACT

Depletion of shallow mineral resources is forcing mining companies to exploit deeper deposits and design more complex mine access tunnels due to sophisticated nature of subsurface mining. Inevitably, this brings more complex mine ventilation networks, associated with higher energy profiles. Also, deep mines (i.e. deeper than 1km) and ultra-deep mines (i.e. deeper than 2.5km) are subjected to other heat loads, sourced by strata heat, auto-compression and equipment heat. These extreme heat loads result in further energy demands for the purpose of mine ventilation and air conditioning which is usually satisfied by grid power (or diesel generation in off-grid applications). Therefore, understanding the performance of mine ventilation and air conditioning systems is a necessity. For mine ambient air conditioning applications, bulk-air-coolers relying on spray cooling systems are commonly used. Although mine ventilation literature provides enough empirical design tools for industrial bulk-air-cooler system design, it still lacks a deeper numerical understanding to attain higher precisions in large-scale designs. Accordingly, this paper aims to provide a valid Computational Fluid Dynamics and Heat Transfer model to better understand the working principles of bulk-air-cooling systems. For this purpose, a previously validated CFDHT model was used to test the applicability of literature-ready, analytically expressed semi-empirical bulk-air-cooler design tools. Present study not only highlights the robustness of the introduced semi-empirical model, but also shows that the design tools used for modern bulk-air-cooler design purposes can capture an experimentally validated CFDHT model within ~7% agreement.

Keywords: advanced mine energy systems, bulk-air-coolers, deep-mine cooling, spray cooling systems, heat transfer, mine ventilation

NONMENCLATURE

Abbreviations	
HVAC	Heating, vent. and air-cond.
BAC	Bulk-air-cooler
OPEX	Operating expense
CFD	Computational fluid dynamics
WB	Wet-Bulb
DB	Dry-Bulb
Symbols	
m_a	Air mass flow rate (kg/s)
m_w	Water mass flow rate (kg/s)
S_{out}	Sigma of outlet air (J/kg)
S_{in}	Sigma of inlet air (J/kg)
C_w	Specific heat for water (J/^{o}C/kg)
$t_{w,in}$	Water inlet temperature (^{o}C)
$t_{w,out}$	Water outlet temperature (^{o}C)
η_w	Water efficiency (Dimensionless)
η_a	Air efficiency (Dimensionless)
R	Cooler Capacity Factor (Dimensionless)
R*	Capacity Factor (for R>1) (Dimensionless)
F	Factor of Merit (Dimensionless)
E	Cooling effectiveness (Dimensionless)
N	Number of Staging (Dimensionless)

1. INTRODUCTION

Mining operations are very energy intensive and their energy reliance is growing bigger due to rapid depletion of shallow deposits. As Vergne [1] states a typical underground mine may need 100 kWh per tonne ore mined and processed. For a well-developed, fully functional subsurface mine, 30-50% of this overall energy is used for mine heating, ventilation and air conditioning (HVAC) systems [2].

According to Ranjith et al. [3], average depth of South African mines has already reached 2 km including cases which are already deeper than 4.5 km. China has 47 coal mines reaching +1 km depth [3] and Canada has one mine deeper than + 3 km [4] with another 5 planning to go below 3 km depth in future. Therefore, it is reasonable to say that mine HVAC systems are gaining even more importance as the mine ventilation networks and underground tunnel systems are getting more complex due to deeper mine designs.

Cooling effect of mechanical ventilation remains insufficient and mine air conditioning by other methods becomes mandatory after 2 km mine depth [1], due to inclusion of several sources of heat i.e. auto compression, strata heat and mine machinery [5]. American Society of Heating, Refrigerating and Air-conditioning (ASHRAE) states that all North American mines are demanded not to exceed 27-29 ^{o}C wet-bulb temperature [6] to prevent heat stress and hyperthermia issues. To provide cooling to their subsurface operations, modern mines usually prefer to deploy conventional refrigeration systems that could total up to $50 million in upfront capital investment which is also accompanied by large annual operational expenses (OPEX) [7]. Agnico Eagle's La Ronde mine, a Canadian example, decently highlights the size of a typical ultra-deep mine cooling system. With almost ~1.7 million cfm air provision La Ronde needs almost 30 MW of underground mine cooling [8], [9].

Given the regulatory and operational importance of mine HVAC systems, engineering design of these systems cannot be underestimated. On the other hand, they are very large in scale and therefore, substantially expensive and should require to be designed precisely. Malcolm McPherson's highly esteemed book [10] provides very robust and accurate design tools for mine refrigeration system design (Chapter 18, pp.651-738). Another one of the most reliable authorities in mine HVAC systems, ASHRAE uses McPherson's methodology as well [6]. It is known that the empirical method proposed by McPherson [10] is highly accurate and very robust in conventional mine bulk-air-cooler (BAC) design. However, to the best of the authors' knowledge there is a lack in the number of numerical studies testifying this methodology in the mining literature. Given the reliability of computational fluid dynamics (CFD) and its flexibility in parametric controls, it is highly believed that CFD simulations could add further value to McPherson's design methods by evaluating the effects of new parameters; i.e. nozzle characteristics (size and angle), droplet characteristics and size distribution, spray velocity and etc.

To simulate McPherson's analytic BAC design equations, the present paper uses the experimental results of a residential scale spray cooling system that was initially put forward by Sureshkumar et al. [11], [12] and was further modeled and validated through CFD platform by Montazeri et al. [13]. Throughout this work, the agreement between these two approaches are highlighted and further improvements and recommendations are offered.

2. METHODOLOGY

2.1 McPherson's model

A single spray, single stage cooling system was designed, and same inputs were solved both analytically and numerically. For numerical CFD simulation *Fluent 19.1* was used. For analytical calculations, the design steps listed in McPherson's method were employed (Chapter 18, pp.651-738) [10]. Here, the mathematical relations used for analytical solution are shown as in [10].

Using energy balance between air and spray water following equation can be derived.

$$m_a(S_{out} - S_{in}) = m_w C_w(t_{w,in} - t_{w,out}) \quad (1)$$

Then, knowing the ideal conditions should yield an equality between $t_{w,out}$ and $t_{air,in}$, one can conclude that a non-dimensional ratio can be derived as in Eqn. (2) to represent the water efficiency.

$$\eta_w = \left(\frac{t_{w,in} - t_{w,out}}{t_{w,in} - t_{a,in}}\right) \quad (2)$$

Using the same approach one can establish a similar relationship for air as well. Then,

$$\eta_a = \left(\frac{S_{out} - S_{in}}{S_{w,in} - S_{in}}\right) \quad (3)$$

Consequently, a non-dimensional capacity factor can be derived to represent the cooler performance as shown in Eqn. (4).

$$R = \left(\frac{\eta_a}{\eta_w}\right) \quad (4)$$

Note that, McPherson [10] develops following logical statement to estimate the cooling plant effectiveness, E.

$$E = \eta_a \; if \; R \geq 1 \; \& \; E = \eta_w \; if \; R \leq 1 \quad (5)$$

As seen in the logical statement shown below (6) Whillier introduces a flexibility to the capacity factor R as cited in [10], by introducing R* for the cases that R factor overpasses the unity. This allows inversing the R factor for more precise curve fitting during factor of merit calculations.

$$R^* = R \text{ when } R \leq 1 \;\&\; R^* = 1/R \text{ when } R \geq 1 \quad (6)$$

Logically, McPherson [10] attaches E to R* and F as follows:

$$E = F^{R^*} \quad (7)$$

Then, accordingly McPherson [10] summarizes the staging as:

$$N = \frac{F}{R^{0.4}(1-F)} \quad (8)$$

Here, it is worthwhile to note that, merit factor is a bulk-air-cooler performance parameter and ranges between 0 and 1 [10]. For typical merit factors McPherson [10] or ASHRAE HVAC Handbook can be referred [6].

2.2 Numerical model

A wind tunnel with (0.585m x 0.585m x 1.9m) dimensions was built to mimic the conditions listed by McPherson's design model [10]. The spray simulation was developed using the design characteristics and geometry listed as in Montazeri et al. [13]. For this design, *Fluent 19.1* integrated meshing tool was used. A snapshot of meshing and some mesh statistics are shown on Fig 1.

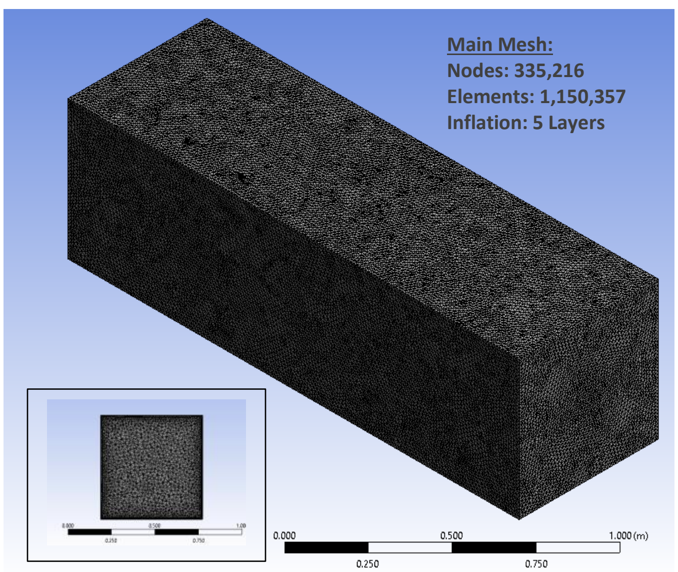


Fig 1 – Mesh conditions and statistics

Also, Table 1 summarizes the boundary conditions used for the numerical simulation. Inputs are taken from a real-life Canadian example to represent reality. Applicable inputs used for the analytical model are also marked on Table 1. Note that; realizable, k-Ɛ model with scalable wall functions was used as viscous model. Also, 'escape' boundary condition was used for the discrete phase.

For the analytical solution, as a typical, horizontal BAC input, 0.7 merit factor is estimated. Also, in typical BAC applications nozzle pressure should not exceed 2-3 bars [10] with max 6 m/s water velocity at nozzle outlet.

Table 1 – Spray cooling model parameters

Parameter	Numerical	Analytical	Source
Inlet Air (DB)	23ºC	23ºC	
Inlet Air (WB)	18ºC	18ºC	
Inlet Water	6ºC	6ºC	[14]
Air Flow Rate	3 m/s	3 m/s	
Water Flow Rate	12 l/min	12 l/min	
Merit Factor	N/A	0.7	
Min. Droplet Size	74 µm		
Max. Droplet Size	518 µm		
Mean Droplet Size	369 µm		
Rosin-Rammler Spread Parameter	3.67	N/A	[13]
Hollow-Cone Angle	18º		
Nozzle Size	4 mm		
Turbulent Intensity	10%		
Turbulent Length Scale	0.041 m		

In order to compare the results of both models, outlet air conditions (*i.e. DB Temperature, H_2O molar fraction, RH % and heat flux* $_{inlet\text{-}outlet}$) are compared and shown. In order to understand the mesh dependency of the numerical model a grid sensitivity was also conducted and presented. Fig 2 compares three mesh sizes with their corresponding mesh statistics.

3. RESULTS AND DISCUSSION

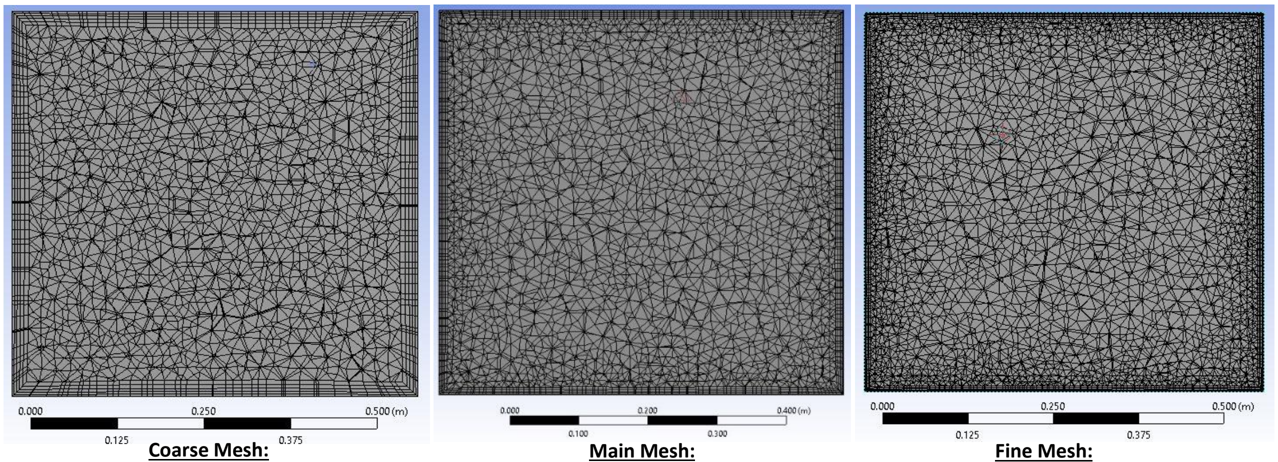
Inputs listed in Table 1 are introduced to both models to observe the differences in results. Also, numerical model was run for three different grid sizes to estimate the impact of the of the mesh size on the output conditions.

Here, it is shown that results obtained from McPherson's design model [10] is very capable in capturing the CFD model results and yields very realistic results as supported by the CFD runs conducted in *Fluent 19.1*.

Table 2 below summarizes the outlet conditions captured with McPherson's design introduced in the previous section [6], [10] and CFD model presented here. Note that, following CFD results are the results obtained from the 'main mesh' model.

It is important to note that, model proposed by McPherson [10] is designed to size cooling wattage and water needed for a prospective plant. However, hence the water content for the given case is constant (12 l/min) a reverse calculation by using Excel's goal-seek function was used to estimate the outlet WB temperature for the given equations in *Chapter 2.1*. Also, during CFD runs, it was observed that all the 6000 discrete elements generated (300 streams x 20 drops) during DPM iteration was fully escaped from the outlet boundary, yielding no evaporation, allowing the overall H_2O mass fraction constant. This was used to define the outlet dry-bulb temperature of McPherson's model [10].

CFD setup within ~7% difference. Note that spray position and droplet characteristics are of a prime importance for the heat transfer process and needs to be further discussed in a more elaborate way. For this study only the given geometry discussed by Sureshkumar [11], [12] and Montazeri [13] was used. To explain further, the



Coarse Mesh: **Nodes: 106808** **Elements: 430549**

Main Mesh: **Nodes: 335216** **Elements: 1150357**

Fine Mesh: **Nodes: 588908** **Elements: 1983501**

Fig 2 – Mesh statistics used for the grid sensitivity analysis

Table 2 – Outlet air condition comparison between analytical and numerical model

Parameter	Numerical	Analytical [10]
Water Flow (kg/s)	**0.204**	**0.204**
Out. Air (DB)	18.65 ºC	17.80ºC
Out. Air (WB)	16.50 ºC	16.04ºC
Out. Rel. Hum.	83.81 %	84.00 %
Out. Air H_2O Mass Fr.	0.0108 kg/kg	0.0108 kg/kg
Tot. Coolth Delivered	**6436 W**	**6918 W**

As it can be observed from the results, McPherson's analytical model [10] is in agreement Montazeri's [13]

spray was positioned at the center of the inlet face in a parallel flow fashion. See Fig 3 below showing the temperature distribution across the middle plane of the given geometry in Fig 1. On the other hand, Fig 4 represents the vertical sections of the wind-tunnel geometry for the given positions. Here, it is important to observe the distribution of the static temperature across the planes and weighted average temperature across the given sections.

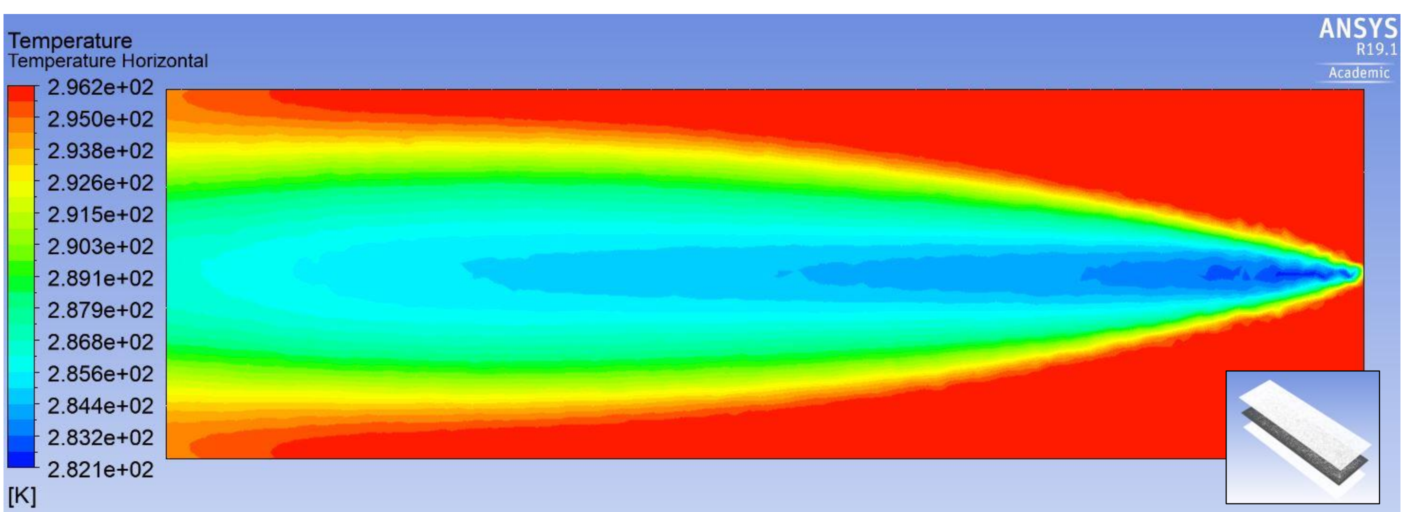


Fig 3 – Mesh statistics used for the grid sensitivity analysis

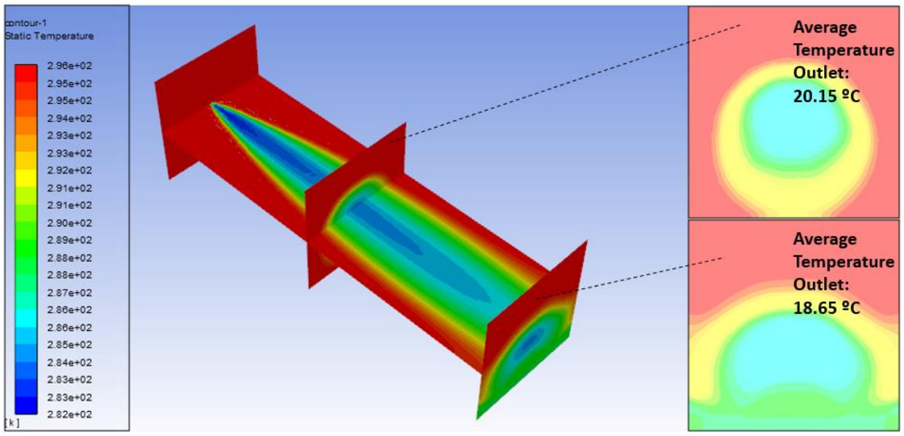


Fig 4 – Temperature distribution at different faces (Vertical Sections at x=0, x=0.95 and x=1.90m Horizontal Section at x=0.2925m)

Finally, water velocity at the nozzle outlet is set to ~6 m/s as widely accepted by the conventional bulk-air-cooler designs found in mining applications. This parameter can be very critical in terms of droplet behavior and it is believed that it could have very critical impact on the final performance of the cooler. Certainly, inclusion of discrete phase interaction equations is very difficult to implement with analytical approaches and requires employment of mathematically complex equations. On the other hand, CFD can reliably help solving these complex interactions and create an opportunity to

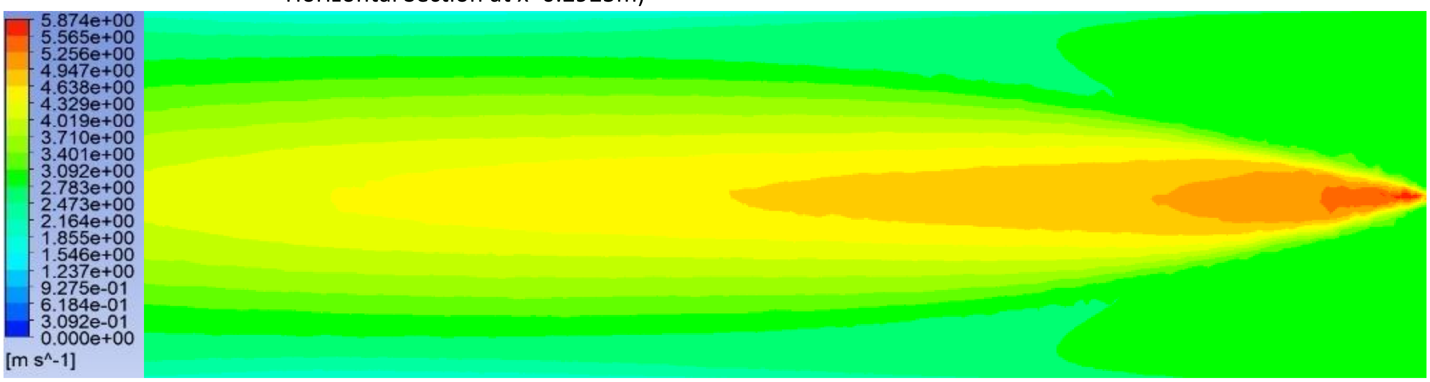


Fig 5 – Discrete phase velocity (m/s)

3.1 Grid sensitivity study

To observe the impact of the mesh sizing on the results obtained from the numerical runs a grid sensitivity analysis was conducted with mesh statistics shown on Fig 2. During this comparison, coolth wattages of each mesh scenario was compared and no significant change was observed with a maximum relative difference attained no more than ~5% between each grid scenario. Table 3 below shows the coolth wattages obtained from each run with different mesh sizes and signifies the relative percentage differences compared with McPherson's analytical model [10].

Table 2 – Cooling wattages attained by each grid scenario

Parameter	Flux (W)	Diff (%)
Analytical Run (W): 6918		
Coarse Mesh	6462	6.59
Main Mesh	6436	6.96
Fine Mesh	6444	6.86

include these critical factors into account during design of conventional mine air cooling systems with relatively less complexity.

Fig 5 above shows the discrete phase velocity across the middle plane introduced in Fig 3 and Fig 4. At this point, it should be remarked that one other significant improvement to this study would be modeling of multiple nozzle systems to observe the ultimate effect of complex nozzle systems on the overall system design by investigating their internal interaction.

4. CONCLUSION

It is observable that subsurface mining operations are growing deeper with more complex tunnel networks. Nowadays, not only the ventilation of these mines is getting more complex, but their refrigeration could be problematic as well. Mining literature is heavily reliant on analytical interpretations obtained from the empirical works done in the past. However, as computational techniques and methods evolve, more sophisticated yet elegant problem-solving techniques are also being developed.

Computational fluid dynamics techniques are very prominent and reliable tools used to estimate fluid behavior in complex domains with high precision. On the other hand, the analytical equations developed based on previous studies are also very robust and usually provide industry-accepted results.

In this study, it is aimed to bring together two approaches for the same problem and compare the results obtained from both solution techniques. For this, a numerically and experimentally validated, single spray based cooling system was used as a baseline to test the capability of McPherson's cooling design equations listed in his book [10], (Chapter 18, pp.651-738). The results of this comparison has shown that McPherson's equations [10] can reliably capture what was found by the CFD runs with an agreement of ~7%.

The main purpose of this study was to introduce new parameters to the design process in order to have further flexibility by still achieving reliable results. It is believed that these parameters like spray conditions such as size distribution, nozzle aperture size, spray angle, cone characteristics and so on, can bring indispensable value to the precision of conventional bulk-air-cooler design methods used in the mining industry.

5. RECOMMENDATIONS

The results of this study can be further strengthened with several parametric studies on spray characteristics. Droplet distribution parameters can also be further examined and studied. Moreover, impact of different spray angles and spray geometries (i.e. solid cone) can also be studied in the future. Lastly, multiple spray systems in a bulk-air-cooling system can be studied and optimized as a future work.

Most of these parameters are empirically estimated in the modern bulk-air-cooling system design process and can be further improved or investigated in a CFD environment.

The scope of this study is aimed to be further expanded with these steps in the future.

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