θ	= polar angle	[radian]
μ	= viscosity	[g/cm·sec]
ρ	= density	[g/cm ³]
τ	= dimensionless stress acting at the surface of a drop	[—]
$\bar{\phi}$	= stream function	[cm ³ /sec]
<Subscripts>		
c	= continuous phase	
d	= dispersed phase	
r	= radial direction	
rr	= radial direction	
$r\theta$	= tangential direction	

Literature Cited

- 1) Boussinesq, J.: *J. Math. Pure Appl.*, **6**, 285 (1905)
- 2) Griffith, R. M.: *Chem. Eng. Sci.*, **12**, 158 (1960)
- 3) Hadamard, J.: *Compt. Rend.*, **152**, 1735 (1911)
- 4) Hamielec, A. E. and A. I. Johnson: *Can. J. Chem. Eng.*, **40**, 41 (1962)
- 5) Hamielec, A. E., S. H. Storey and J. M. Whitehead: *Can. J. Chem. Eng.*, **41**, 246 (1963)
- 6) Hozawa, M., T. Tadaki and S. Maeda: *Kagaku Kōgaku*, **34**, 315 (1970)
- 7) Lochiel, A. C. and P. H. Calderbank: *Chem. Eng. Sci.*, **19**, 471 (1964)
- 8) Marquardt, D. W.: *J. Soc. Indus. Appl. Math.*, **11**, 431 (1963)
- 9) Nakano, Y. and C. Tien: *Can. J. Chem. Eng.*, **45**, 135 (1967)
- 10) Ranz, W. E. and W. R. Marshall: *Chem. Eng. Progr.*, **48**, 141, 173 (1952)
- 11) Ruckenstein, E.: *Chem. Eng. Sci.*, **19**, 471 (1964)
- 12) Rybczynski, W.: *Bull. Acad. Sci. Cracov.*, **1**, 40 (1911)
- 13) Shiotsuka, T. and A. Hirata: *Kagaku Kōgaku*, **35**, 78 (1971)
- 14) Tadaki, T. and S. Maeda: *Kagaku Kōgaku*, **25**, 254 (1961)
- 15) Taylor, T. D. and A. Acrivos: *J. Fluid Mech.*, **18**, 466 (1964)
- 16) Ueyama, K. and J. Hatanaka: *Kagaku Kōgaku*, **35**, 605 (1971)

EFFECT OF SUB-COOLING ON FILM-BOILING HEAT TRANSFER

— DROPLET EVAPORATION ON A HOT PLATE* —

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The effect of liquid-subcooling on film-boiling heat transfer is estimated theoretically for the range

$$0.5 \leq Pr \leq 2.0, 0.03 \leq C_p'' \Delta T / h_{fg} \leq 4.0, 0.1 \leq Q \text{ (heat flux parameter)} \leq 2.0$$

It is observed from the results that the subcooled conditions of the liquid patch exert influence upon heat transfer rates for a practicable range of the heat capacity parameter. This aspect, hitherto not considered, reveals profound influence on the evaporation rates of the liquid patch, under spheroidal conditions. A comparison of the analysis with the experimental data reveals reasonably good agreement, substantiating the theory proposed.

Introduction

After the pioneering effort of Borishansky²⁾ on Leidenfrost phenomena, the problem of droplet evaporation was theoretically investigated by many^{1,4,5,7,9,17,18)}. Baumeister *et al.*¹⁾ and Gottfried *et al.*⁴⁾ solved the problem for two distinct physical models. The model of Baumeister¹⁾ assumes that the vapor-liquid interface is a horizontal plane separated from the hot plate by a thin layer of vapor. The flow of vapor is considered essentially to be two-dimensional laminar, the temperature being inde-

pendent of the radial coordinate. The evaporation rates are estimated on an analog computer. The model of Gottfried *et al.*⁵⁾ assumes total sphericity of the droplet and their solution includes possible molecular diffusion and radiation effects. In almost all the analyses, the effect of subcooled conditions of the droplet on evaporation rates is not considered. The analytical model of King and Scully⁷⁾ assumes that the total quantity of the vapor leaking out in the radial direction is supplied from the central core region of the patch of finite radius. But it is apt to assume that the evaporation takes place uniformly from the plane facing the hot plate. A semi-theoretical analysis is presented in Lit. 8), 9) to predict the partial evaporation rates considering the subcooled state of the liquid patch, evaporating under

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film-boiling conditions. The constant of proportionality of the equation is obtained by the method of best fit.

Film-boiling studies on various geometries of the heater element as reported by many investigators^{6,8,10,11,13-17} reveal profound influence of the subcooled conditions of the bulk on film-boiling heat transfer coefficient.

Thus, the present paper intends to investigate theoretically the effect of subcooled conditions of the liquid droplet on evaporation rates over a practicable range of parameters and to verify the evaporation rates as predicted by the theory with the experimental data. Another possibility, viz. the constant heat flux condition of the wall, is also indicated and analyzed.

Analysis

The physical configuration assumed is a liquid patch under subcooled conditions, separated from the hotwall by a thin vapor layer. Intensive vaporization occurs at the vapor-liquid interface (which is always at the saturation temperature though the bulk of the liquid is under subcooled conditions), the flow of vapor being two-dimensional and laminar. In the absence of bubble generation in the bulk of the liquid patch, the flow is symmetrical about the z -axis. The conservation of mass, momentum and energy equations are, respectively

$$\frac{\partial u}{\partial r} + \frac{u}{r} + \frac{\partial w}{\partial z} = 0 \quad (1)$$

$$\rho \left(u \frac{\partial u}{\partial r} + w \frac{\partial u}{\partial z} \right) = \frac{\partial p}{\partial r} + \mu \left(\frac{\partial^2 u}{\partial r^2} + \frac{1}{r} \frac{\partial u}{\partial r} - \frac{u}{r^2} + \frac{\partial^2 u}{\partial z^2} \right) \quad (2)$$

$$u \frac{\partial T}{\partial r} + w \frac{\partial T}{\partial z} = \alpha \frac{\partial^2 T}{\partial z^2} \quad (3)$$

In writing the above system of equations it is assumed that the equation of conservation of momentum in z -direction can be neglected for the solution of the problem by an order of magnitude analysis. The boundary conditions can be enunciated for the problem as

$$u=0; \quad w=0; \quad T=T_w$$

$$\text{or } q=q_w \text{ (constant heat flux) at } z=0$$

$$u=0; \quad T=T_s \text{ at } z=\delta$$

The above set of equations are rendered non-dimensional by introducing the following solutions and similarity variables:

$$u = r f'(z) = r \phi'(\eta) \left[\frac{2 \Delta p}{\rho_v R^2} \right]^{1/2} \quad (4)$$

$$w = -2f(z) = -2\phi(\eta) \left[\frac{2 \Delta p \nu_v^2}{\rho_v R^2} \right]^{1/4} \quad (5)$$

$$\eta = z \left[\frac{2 \Delta p}{\rho_v \nu_v^2 R^2} \right]^{1/4} \quad (6)$$

where

$$\Delta p = P_s - P_o$$

$$\frac{T_w - T}{T_w - T_s} = \Psi(\eta) \quad [\text{constant wall temperature}] \quad (7)$$

or

$$T - T_s = \left[\frac{\rho_v \nu_v^2 R^2}{2 \Delta p K^4} \right]^{1/4} q_w \Psi(\eta) \quad [\text{constant heat flux}] \quad (8)$$

Further, the pressure can be assumed to be a sum of two independent functions of r and z . For Hiemenz-type of flow¹² the pressure variation in radial direction can be assumed parabolic. Thus, we have

$$\frac{P_s - P}{P_s - P_o} = \left[\left(\frac{r}{R} \right)^2 + CF(z) \right] \quad (9)$$

wherein $CF(z)$ is an unknown and can only be obtained by solving the conservation of momentum equation in z -direction. This aspect of the problem is not considered in the present analysis. As a matter of fact the assumption of a parabolic pressure distribution in the radial direction can be shown to be valid by neglecting the inertia terms for a while in the conservation of momentum equation and solving the balance between the pressure forces and viscous forces. Further, the boundary condition mentioned ($u=0$ at $z=\delta$) at the vapor-liquid interface does not include the behavior of the convective cells within liquid patch configuration considered in the analysis. As a matter of fact, a more meaningful version of the boundary condition can be written as

$$\mu_v \frac{\partial u}{\partial z} \Big|_{z=\delta} = \mu_L \frac{\partial U}{\partial z} \Big|_{z=\delta}$$

where U refers to the radial component of velocity in the liquid patch. As the equal shear condition at the vapor-liquid interface leaves the problem mathematically incomplete, we tentatively make use of the boundary condition $u=0$ at $z=\delta$ as adopted by Baumeister¹.

Thus, introducing Eqs. (5), (6), (7), (8) and (9) in Eqs. (2) and (3) we get

$$(\phi')^2 - 2\phi\phi'' = 1 + \phi''' \quad (10)$$

$$\Psi'' + 2\Psi'\phi Pr = 0 \quad (11)$$

the boundary conditions being

$$\phi=0; \quad \phi'=0; \quad \Psi=0 \quad (\text{constant wall temperature})$$

$$\text{or } \Psi' = -1 \quad (\text{constant heat flux) at } \eta=0 \quad (12)$$

$$\phi'=0, \quad \Psi=1 \quad (\text{constant wall temperature})$$

$$\text{or } \Psi=0 \quad (\text{constant heat flux) at } \eta=\eta_\delta \quad (13)$$

Thus, Eqs. (10) and (11) with the boundary conditions stated are mathematically complete and are to be solved simultaneously.

The process peculiarity associated with the vapor-

zation taking place at the interface can be written as

$$\rho_v w(\delta) [h_{fg} + C_p' \Delta T_L] = -K_v \frac{dT}{dz} \Big|_{z=\delta} \quad (14)$$

or, in terms of transformed variables we have for constant wall temperature

$$\frac{2\phi(\eta_\delta)}{\Psi'(\eta_\delta)} = \frac{H}{1 + C_p' \Delta T_L / h_{fg}} \quad (15)$$

and for constant heat flux

$$\frac{2\phi(\eta_\delta)}{\Psi'(\eta_\delta)} = \frac{q_w X^{1/4} / \rho_v h_{fg}}{1 + C_p' \Delta T_L / h_{fg}} \quad (16)$$

where

$$H = C_p'' \Delta T / h_{fg} \cdot Pr$$

$$X = (\rho_v R^2) / 2 \Delta p \nu_v^2$$

Equations (15) and (16) are obtained on the assumption that the total heat conducted to the interface is equal to the gain in enthalpy by a fraction of the mass of the droplet evaporating from the subcooled to the saturation conditions and thence the latent heat of vaporization. The difficulty in estimating the temperature gradient in the liquid medium at the interface is obviated by such an approximation. At the first instant Eqs. (15) and (16) suggest that the case of a constant heat flux and constant wall temperature may turn out to be one and the same. But the change in the boundary conditions, as is obvious from Eqs. (12) and (13), will yield different solutions for the two cases under consideration.

Solutions of Eqs. (10) and (11)

Equations (10) and (11) are solved numerically by the Runge-Kutta method on an IBM 1130 for the following range of parameters:

$$0.5 \leq Pr \leq 2$$

$$0.03 \leq C_p'' \Delta T / h_{fg} \leq 4$$

$$0.1 \leq Q \text{ (heat flux parameter)} \leq 2$$

The results obtained are interpreted with the aid of Eqs. (15) and (16) to arrive at meaningful conclusions.

Velocity and Temperature Profiles

Figure 1 represents the velocity and temperature profiles for a constant heat flux condition on the wall. It is obvious that the velocity and temperature profiles are profoundly influenced by the subcooling parameter $C_p'' \Delta T_L / h_{fg}$. For constant heat flux conditions of the wall, as the parameter $C_p'' \Delta T_L / h_{fg}$ increases, the velocity of vaporization at the vapor-liquid interface decreases.

Vapor Film Thickness

The vapor film thickness is monotonically increasing with the wall temperature or heat flux. However, for a given Pr and the heat capacity parameter $C_p'' \Delta T / h_{fg}$,

Fig. 1 Velocity and temperature profiles

Fig. 2 Dimensionless film thickness as a function of heat capacity parameter

Fig. 3 Dimensionless film thickness as a function of heat flux parameter

an increase in subcooling parameter shows marked decrease in the vapor film thickness. These observations can be made from Figs. 2 and 3.

Fig. 4 Effect of Prandtl number on heat transfer coefficient

Fig. 6 Effect of subcooling on $\psi'(\eta_\delta)$

Heat Transfer Coefficient

For the case of constant wall temperature, the heat transfer coefficient from the wall to the liquid patch can be computed from

$$h = \frac{-K}{T_w - T_s} \frac{dT}{dz} \Big|_{z=0} \quad (17)$$

or

$$h = \left[\frac{2\Delta p K^4}{\rho_v \nu_v^2 R^2} \right]^{1/4} \Psi'(0) \quad (18)$$

Since $\Psi'(0)$ is strongly dependent on $C_p'' \Delta T / h_{fg}$; $C_p'' \Delta T_L / h_{fg}$ and Pr we have

$$hN^{1/4} = F \left[\frac{C_p'' \Delta T}{h_{fg}}; \frac{C_p'' \Delta T_L}{h_{fg}}; Pr \right] \quad (19)$$

The exact relationship of Eq. (19) is shown in Figs. (4) and (5). **Figure 4** is plotted to bring out the Prandtl dependency and, obviously, as Prandtl number increases for a given heat capacity parameter the heat transfer coefficient increases. **Figure 5** brings out the effect of subcooling parameter on heat transfer coefficient for different Prandtl numbers. Evidently with the increase of $C_p'' \Delta T_L / h_{fg}$ the heat transfer coefficient increases. But for higher values of heat capacity parameter ($C_p'' \Delta T / h_{fg} \leq 1.5$) the change in heat transfer coefficient is not significant.

Estimation of Evaporation Rates

For the confirmation of theory, we take up the

Fig. 5 Effect of subcooling parameter on heat transfer coefficient

partial evaporation of the liquid patch for which data are readily available.

Thus, we have

$$-\frac{dM}{d\mathcal{T}} [h_{fg} + C_p' \Delta T_L] = -K_v \frac{dT}{dz} \Big|_{z=\delta} \pi R^2 \quad (20)$$

The solution of Eq. (20) gives the variation of mass with time. For a patch, we have

$$\pi R^2 \delta_p \rho_L = M \quad (21)$$

where δ_p = thickness of the liquid patch.

Introducing Eq. (21) in Eq. (20) and integrating within proper limits the mass time dependency is obtained as follows.

$$1 - \left(\frac{M}{M_0} \right)^{1/4} = \frac{\Psi'(\eta_\delta)}{4(1+B)} \left(\frac{\rho_v}{\rho_L} \right)^{3/4} \left(\frac{C_p'' \Delta T}{h_{fg}} \right) \alpha \mathcal{T} / L^{*2} \quad (22)$$

where

$$B = C_p' \Delta T_L / h_{fg}; \quad L^{*2} = \left[\frac{V_0 \nu_v^2 \delta_p^2}{4\pi g} \right]^{1/4}$$

and $\Psi'(\eta_\delta)$ can be obtained from **Fig. 6** or from an approximation

$$\begin{aligned} & [\Psi'(\eta_\delta) + 0.6] \\ &= 1.12 [C_p'' \Delta T / h_{fg}]^{-0.181} [1 + C_p' \Delta T_L / h_{fg}]^{0.191} \end{aligned} \quad (23)$$

In all the computations ΔT_L , which represents the mean subcooled state of the liquid patch during the evaporation period, is calculated from the expression³⁾

$$\frac{\bar{\theta}_m}{\theta_o} = 1 - \frac{0.83}{A \mathcal{T}_m} [1 - \exp(-A \mathcal{T}_m)] \quad (24)$$

where A (dimensional constant) = $2.46 \alpha / \delta_p^2$

$$\bar{\theta}_m = \bar{T}_B - T_L$$

$$\theta_o = T_s - T_L$$

$\mathcal{T}_m$ = time of stay of liquid patch on the hot plate.

From Eq. (24) the value of ΔT_L is calculated from the expression

$$\Delta T_L = T_s - \bar{T}_B$$

Equation (24) is derived³⁾ for a specific set of thermal conditions imposed on a slab which is initially at

uniform temperature and is subjected to a sudden change of temperature on one face, the other being insulated. The same equation is utilized to calculate the mean temperature of the bulk of the liquid patch on the assumption that the vapor-liquid interface remains at a constant temperature T_s and from the other face no mass transfer takes place. Equation (22) is compared with experimental data²⁾ and the reasonable agreement, as is evident from Fig. 7, justifies the validity of the analysis presented. However, it is to be readily admitted that the semi-empirical equation of Kutateladze^{8,9)} agrees better, the reason being that the constant factor of multiplication is derived from the data. At this stage it is anticipated that a more accurate analysis which includes the equal shear condition at the vapor-liquid interface may lead to better agreement.

Conclusions

The following conclusions can be drawn from this theoretical investigation.

1. The heat transfer coefficient increases with increase in Prandtl number for a given heat capacity parameter.
2. The evaporation rate $dM/d\mathcal{T}$ is strongly dependent on $C_p'' \Delta T/h_{fg}$; $C_p' \Delta T_L/h_{fg}$ and Eqs. (22), (23) can be utilized to estimate the mass-time dependency.

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Nomenclature

C_p'	= specific heat of liquid	[Kcal/Kg·°C]
C_p''	= specific heat of vapor	[Kcal/Kg·°C]
g	= acceleration due to gravity	[M/hr ²]
h	= film heat transfer coefficient	[Kcal/M ² ·hr·°C]
h_{fg}	= latent heat of vaporization	[Kcal/Kg]
K	= thermal conductivity	[Kcal/M·hr·°C]
M	= mass of the liquid patch	[Kg]
P	= pressure	[Kg/M·hr ²]
P_o	= system pressure	[Kg/M·hr ²]
P_s	= pressure at stagnation point	[Kg/M·hr ²]
Δp	= $P_s - P_o$	[Kg/M·hr ²]
Pr	= vapor Prandtl number	[—]
q	= heat flux	[Kcal/M ² ·hr]
R	= radius of the liquid patch	[M]
T	= temperature	[°C]
$\bar{T}_B$	= mean bulk temperature of the liquid during evaporation	[°C]
T_L	= initial temperature of the liquid	[°C]
ΔT	= $T_w - T_s$	[°C]
$\Delta \bar{T}_L$	= $T_s - \bar{T}_B$	[°C]
$\mathcal{T}$	= time	[hr]

Fig. 7 Verification of the analysis

u	= radial component of vapor velocity	[M/hr]
w	= vertical component of vapor velocity	[M/hr]
z	= axial space variable	[M]
α	= thermal diffusivity	[M ² /hr]
δ	= vapor film thickness	[M]
η	= similarity variable defined by Eq. (6)	[—]
η_δ	= dimensionless vapor film thickness	[—]
μ	= absolute viscosity	[Kg/M·hr]
ν	= kinematic viscosity	[M ² /hr]
ρ	= density	[Kg/M ³]
ϕ	= dimensionless variable defined by Eq. (5)	[—]
ϕ	= dimensionless temperature defined by Eq. (7)	[—]

<Subscripts>

v	= vapor
L	= liquid
s	= saturation conditions
w	= plate

Literature Cited

- 1) Baumeister, K. J., T. D. Hamill, F. L. Schwartz and G. J. Schoessow: NASA-TMX 52103 (1965)
- 2) Borishansky, V. M.: "Vaprosi Teplo Obmena Pri Izmenii Agregatnaya Sostoyaniya Veshitva", Eds., S. S. Kutateladze, Govt. Publishing House, Moscow, U.S.S.R. (1953)
- 3) Eckert, E. R. G. and R. M. Drake: "Analysis of Heat and Mass Transfer", McGraw Hill, New York (1972)
- 4) Gottfried, B. S. and K. J. Bell: *Ind. Eng. Chem., Fundamentals*, **5**, 561 (1966)
- 5) Gottfried, B. S., C. J. Lee and K. J. Bell: *Int. J. Heat & Mass Transfer*, **9**, 1167 (1966)
- 6) Hamill, T. D. and K. J. Baumeister, NASA-TNO 3925 (1967)
- 7) King, P. J. and D. B. Scully: *J. Appl. Chem. (London)*, **16**, 258 (1966)
- 8) Kutateladze, S. S.: AEC Tr. 3770 (1952)
- 9) Kutateladze, S. S.: "Osnovi Teorii Teploobmena", 2nd ed. p. 365, (in Russian) Moscow (1962)
- 10) Merte, H. and J. A. Clark: *J. of Heat Transfer ASME*, **26**, (Series C) 351 (1964)
- 11) Merte, H. and E. W. Lewis: Boiling of Liquid Nitrogen in Reduced Gravity Fields with Subcooling, Heat Transfer Lab., Dept. of Mech. Eng., Univ. of Michigan, Ann Arbor, Michigan (1967)
- 12) Schlichting, H.: "Boundary Layer Theory", 4th ed., p. 78, McGraw Hill, New York (1962)
- 13) Seetharmu, K. N., "Studies on Film Boiling", Ph. D. Thesis, I.I.T. Madras (1973)
- 14) Siviour, T. B. and A. J. Ede: 4th Int. Heat Transfer Conf.,

Paris, Vol. 5, 3.12 (1970)

- 15) Sparrow, E. M. and S. R. D. Cess: *J. of Heat Transfer ASME*, **84**, (Series C), 149 (1962)
16) Tachibana, F. and S. Fukui: *Int. Develop. in Heat Transfer*, p. 219, ASME (1961)

- 17) Tamura, Z. and Y. Tanasawa: 7th Symposium (International) on Combustion, Butterworths London, p. 509 (1959)
18) Wachters, L. H. J., H. Bonne and H. J. Van Nouhuis: *Chem. Eng. Science*, **21**, 923 (1966)

BATCH MEASUREMENT OF ADSORPTION RATE IN AN AGITATED TANK — PORE DIFFUSION KINETICS WITH IRREVERSIBLE ISOTHERM* —

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A simple technique is proposed to determine the intraparticle diffusivities of adsorbents from concentration change in an agitated tank. Pore diffusion and rectangular isotherm are assumed to derive the theoretical solutions, which are then compared with the experimental results. The theory gave a good prediction for the transient concentration profile in the tank. The effective diffusivities of 2-dodecylbenzene sulfonate in four different activated carbon particles are determined by the proposed technique.

Introduction

Adsorption from a solution by a solid adsorbent has attracted considerable attention in connection with advanced treatment of waste water. One of the important problems met in the course of designing adsorption systems is the determination of contact time in adsorption equipment. In the case of aqueous systems, the rate of adsorption is controlled by diffusion of the solute into adsorbent particles.

For adsorption in aqueous solutions, we frequently meet strongly non-linear isotherms such as a rectangular one which means strong adsorption and limited adsorption capacity. For determination of intraparticle diffusivity in such a case, the most common method is to apply the theories developed by Kawazoe⁶⁾ or Vermeulen *et al.*⁴⁾ to the concentration breakthrough for flow through a packed bed of adsorbent.

However, this type of experiment sometimes requires a long time to finish and then the difficulty lies in getting data for different conditions such as different samples, particle sizes or temperatures.

Batch measurements seem to solve these difficulties. In the case when isotherm is given by Henry's relation, the theories developed by the previous investigators^{1-3,5)} are applicable, but no available theory has been

presented for non-linear isotherm systems with pore diffusion kinetics.

The purpose of this note is to show a way to rapidly estimate the intraparticle diffusivity by batch measurement for a rectangular isotherm system which, as mentioned before, is common in adsorption from aqueous solutions. Adsorption rates of 2-dodecylbenzene sulfonate on four different activated carbons were measured and the proposed technique was applied to the estimation of the intraparticle diffusivities.

Development of a Model

The following assumptions are made:

- (1) Isotherm is rectangular. Then the adsorption site is either vacant or saturated with the adsorbate, depending on whether the liquid-phase concentration is zero or finite.
- (2) The particle is homogeneous.
- (3) Accumulation of adsorbate in the liquid phase in a particle is negligible.

The above assumptions lead to an idea similar to the shrinking-core model for the case of gas-solid reactions¹⁰⁾. Namely, adsorption takes place only at adsorption front, $r=r_i$, where concentration of liquid phase is considered to be zero. Solid phase in the shell between adsorption front and the external surface of the particle ($r_i < r \leq R$) is saturated by adsorbate, and then the velocity of movement of adsorption front toward the center of the particle is controlled by the

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