Experimental Study of Performance Characteristics of Various Fin Types for Fin-Tube Heat Exchanger

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Key words : Fin-tube heat exchanger, Wave-slit fin, Slit fin, Louver fin, Heat transfer coefficient, Pressure drop, Friction factor

Abstract

Air side heat transfer and pressure drop for ϕ9.52 fin-tube heat exchanger with various types of slit and louver fins were measured, and compared with wave-slit fin. Longitudinal and transverse tube spacings of the heat exchangers are 21.65 mm and 25 mm respectively. Actual heat exchanger was tested using water, and the tests were performed for 2 row heat exchangers with 3 different fin spacings, 1.3, 1.5 and 1.7 mm. The overall performance of the enhanced fins was evaluated by comparing heat transfer coefficient with respect to fan power

Nomenclature

- A : heat transfer area [m²]
- A_{min} : min. free flow area [m²]
- c_p : specific heat [J/kg℃]
- d_o : tube outer diameter including fin collar [m]
- d_h : hydraulic diameter [m]
- f : fanning friction factor
- G_{max} : max. mass flux through Amin [kg/m²s]
- h : heat transfer coefficient [W/m² ℃]
- H : height of heat exchanger [m]
- j : Colburn j-factor
- k : thermal conductivity [W/m℃]
- L : length of heat exchanger along the flow direction [m]
- $\dot{m}$: mass flow rate [kg/s]
- N_R : number of tube rows
- Nu : Nusselt number
- ΔP : air pressure drop [Pa]
- p_f : fin spacing [m]
- Pr : Prandtl number
- Q : heat transfer rate [W]
- Re : Reynolds number

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S_L : longitudinal tube(row) spacing [m]
S_T : transverse tube(step) spacing [m]
T : temperature [℃]
t_f : fin thickness [m]
U : overall heat transfer coefficient [W/m² ℃]
W : width of heat exchanger [m]
u : velocity [m/s]

Greek symbols

μ : viscosity [kg/ms]
η_s : heat exchanger surface efficiency
ρ : density [kg/m³]

Subscripts

a : air
c : contact area
fr : front
in : inlet
out : outlet
r : water

1. Introduction

Cross-flow fin-tube heat exchanger is most widely used for air-conditioning equipment and one of the major components which affect the system performance. Fin-tube heat exchangers are often designated by tube outer diameter, and ϕ7 and ϕ9.52 are most widely used for domestic room air-conditioners. ϕ7 is mostly used for indoor unit of small and mid-capacity air-conditioners, and ϕ9.52 for the rest. In the present work performance of ϕ9.52(ϕ10.07 including fin collar after expansion) fin-tube heat exchangers of various fin types(slitted and louvered) were experimentally investigated and compared with baseline data for wave-slit fin.

To decide appropriate heat exchanger specification in designing a system, it is essential to know the air and refrigerant side heat transfer coefficients and pressure drops. Air side heat transfer coefficient and pressure drop depend on the fin type, tube diameter, longitudinal and transverse tube spacings, fin spacing, number of tube rows and etc. In general, enhanced fins such as slitted, louvered and wavy fins are widely used, and extensive works have been done for various types of enhanced fins. However, since configurations of enhanced fins are so much diversified and so are the flow and heat transfer characteristics, no generally applicable correlation exists.

In the present work air side heat transfer coefficient and pressure drop were measured for fin-tube heat exchangers with staggered tube array and of various fin types such as slitted and louvered shown in Fig. 1(b)-(f). To evaluate the fin performance considering both pressure drop and heat transfer, heat transfer coefficients were compared with respect to fan power.

It is known that the wavy fin shown in Fig. 1(a) enhances heat transfer by increasing heat transfer area and promoting turbulence. The slitted fins shown in Fig. 1(a)-(e) have several slits formed on the fin surface. By slitting the fin surface, heat transfer is enhanced as boundary layers are developed intermittently on the slit tips. The slitted fins can be categorized into 2 types. The one is single-direction slit where the slits are protruded in the same direction on the fin surface as shown in Fig. 1(a). The other is zigzag(Z) slit where the slits are protruded in both directions as shown in Fig. 1(b)-(e). Z-slit fin is generally known to have higher heat transfer coefficient than single-direction slit. The louvered fin shown in Fig. 1(f) has both effects, turbulence promotion and boundary layer interruption.

For measuring air side heat transfer coefficient, naphthalene sublimation test[1], model test[2,3], actual heat exchanger test[4 9] are widely conducted. Among these, actual heat exchanger test using water were chosen in the present work because it was considered to give the most practical results and most widely used.

Rich[4,5], McQuiston[6], Seshimo and Fujii[7] and Kayansayan[8] presented experimental results for plane fin-tube heat exchanger. Nakayama and Xu[10] conducted experimental study on plane and slitted fins, and proposed a heat transfer correlation predicting the effect of slit size from numerical analysis. Kang and Kim[2], Jung et al.[3] and Yun et al.[11] conducted ϕ7 model heat exchanger test with various types of fin. Yoon et al.[12] conducted actual heat exchanger test of ϕ7 slitted configuration.

As mentioned above few works on ϕ9.52 configuration have been reported and no previous work on the specific fin type dealt with in the present work is available to the authors' knowledge.

2. Experiment

2.1 Experimental apparatus

Figure 2 is a schematic drawing of the heat exchanger experimental apparatus used in the present work, which is composed of test chamber, cord tester, constant temperature water bath and air-conditioning equipment controlling the temperature of the test chamber. Sample heat exchanger is installed at the inlet of the cord tester, and water is supplied to heat exchanger from the outer water bath and recirculated.

Temperatures of the test chamber and water bath and flow rates of air and water are automatically controlled. Air flow rate is calculated from the measured pressure drop across nozzles installed inside the cord tester. Water flow rate is measured using positive displacement flow meter. Inlet and outlet air and water temperatures are measured using 100Ω Pt sensors, and pressure drop across heat exchanger are measured using diaphragm type pressure transducer. Other details about the apparatus are explained in Youn et al.[9]

2.2 Heat exchanger samples

Specifications of the heat exchangers tested are listed in Table 1. All the heat exchanger samples are made of inner-grooved copper tubes of 9.52 mm outer diater and aluminium fins of 0.11 mm thickness(t_f), and have staggered tube array. The tubes were mechanically expanded to secure contact between tube outer wall and fin collar. Outer diameter (d_o) of the tube including fin collar after expansion is 10.07 mm, and longitudinal(S_L) and transverse tube spacings(S_T) are 21.65 mm and 25 mm respectively. Width(W) and height (H) of the heat exchangers are 400 mm and 250 mm respectively, and number of tube rows (N_R) are 2.

Experiments were conducted for 5 types of fin, 4 zig-zag slit fins and 1 louvered fin, as listed in Table 1 and shown in Fig. 1(b)-(f). All the slits are arranged in radial direction around tube as shown in Fig. 1(b)-(f). The zig-zag slit fins are denoted by ZS1-ZS4 and louvered fin by L. 3 fin spacings(p_f), 1.3, 1.5

and 1.7 mm were tested in the present work.

The ZS1 fin shown in Fig. 1(b) is composed of 22 slits whose width and height are 1 mm and 0.6 mm respectively. In the ZS2 fin shown in Fig. 1(c), 4 side slits are removed from the ZS1 fin. In the ZS3 fin shown in Fig. 1(d), the slit height is increased by 0.1 mm and the rest is the same as the ZS2 fin. The ZS4 fin shown in Fig. 1(e) is composed of 9 slits whose width and height is 1.5 mm and 0.8 mm respectively. The L fin has 20 louvers of 0.6 mm width and 0.3 mm height.

The present results were compared with those of wave-slit fin(WS2) shown in Fig. 1(a) reported by Youn et al.[13] The WS2 fin surface is wavy and has 7 single-direction slits as shown in Fig. 1(a).

All the heat exchangers have cross-counter flow configuration, and all the tubes were connected in a series comprising a single circuit. Fifteen different configurations were tested, and 2 heat exchangers were tested for each configuration to obtain averaged results. Total number of heat exchangers tested were 30.

2.3 Test method and conditions

Heat exchanger sample is installed at the inlet of the cord tester and edges are carefully sealed. Unfinned U-bend area of the heat exchanger is insulated to minimize heat loss.

Temperature inside the test chamber, air velocity, temperature and mass flow rate of water and etc. are set to specified test conditions, and it usually takes about 90-120 minutes for the system to converge to a steady state. The system convergence is automatically determined by taking 200 sets of data for major parameters such as air and water temperatures and flow rates for 200 seconds and comparing with specified values. Further details are explained in Youn et al.[9]

Table 2 summarizes the test conditions of the present work. Tests were conducted for 5 different air velocities ranging from 0.75 to

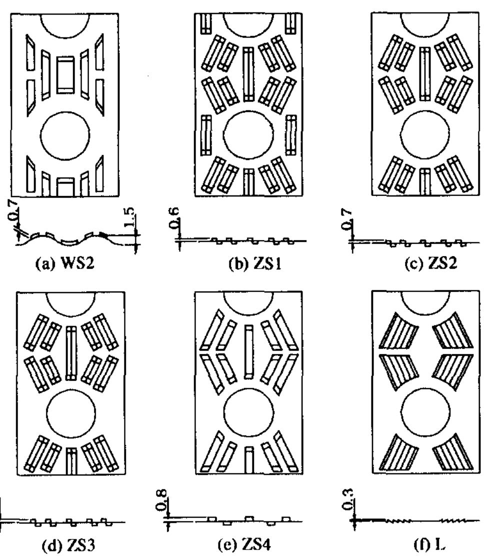


Fig. 1 Fin samples.

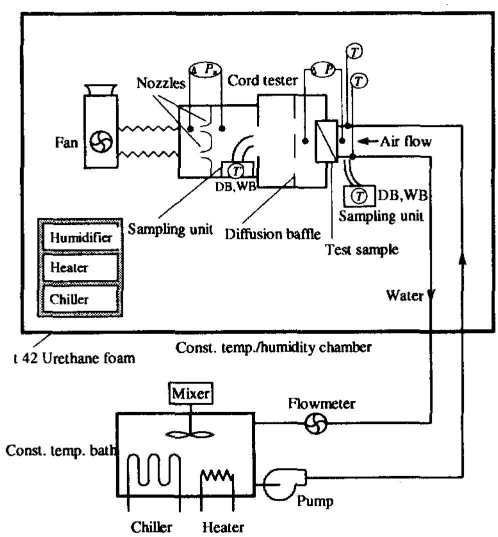


Fig. 2 Schematic of heat exchanger experimental apparatus.

2.5 m/s. Inlet temperatures of the air and water were set to 21℃ and 45℃ respectively, and water flow rate was 0.091 kg/s. These test conditions were selected considering actual operating conditions, limitations of apparatus, uncertainties of test results and etc.

3. Data reduction

Thermal resistance for fin-tube heat exchanger consists of water and air side convection resistances, tube wall and fin collar conduction resistances and contact resistance. Conduction resistances are negligible since the tube and the fin are very thin and thermal conductivities are relatively high. And thus the overall heat transfer coefficient can be expressed as follows.

$$UA=\left(\frac{1}{h_rA_r}+\frac{1}{h_cA_c}+\frac{1}{h_aA_a\eta_s}\right)^{-1} \tag{1}$$

where h and A are heat transfer coefficient and heat transfer area respectively, and subscripts r, c and a denote water, contact and air respectively. ηs is surface efficiency of heat exchanger.

Fin efficiency of staggered fin-tube heat exchanger can be calculated using the approximate equation proposed by Schmidt.[14] For the water side heat transfer coefficient(h_r), the correlation developed by Park et al.[15] was used. They presented heat transfer coefficient of water inside the inner-grooved tube identical to the one used in the present work. Youn et al.[17] suggested a correlation of contact heat transfer coefficient(h_c) based on the experimental data of Sawai et al.[16]

To calculate UA from experimental data it is usual to use LMTD or ε-Ntu method. In general, fin-tube heat exchanger is not a pure cross-flow exchanger when 2 or more tube

Table 1 Specifications of heat exchanger samples

Notation	ZS1	ZS2	ZS3	ZS4	L
Fin type	Slit	Slit	Slit	Slit	Louver
d_o(mm)	10.07				
S_T(mm)	25.0				
S_L(mm)	21.65				
t_f(mm)	0.11				
No. of slits	22	18	18	9	20
Slit height(㎜)	0.6	0.6	0.7	0.8	0.3
W(mm)	400				
H(mm)	250				
p_f(mm)	1.3/1.5/1.7				
N_R	2				
No of samples	6	6	6	6	6

Table 2 Summary of test conditions

Inlet air		inlet water	
Temp.	Frontal vel.(m/s)	Temp.	Mass flow
21℃	0.75/1.0/1.5/2.0/2.5	45℃	0.091 kg/s

rows exist, and ε-Ntu relation is dependent upon tube circuit configuration. In the present work, ε-Ntu relations for 2 row cross-counter flow configuration were obtained using tube-by-tube method explained in Youn et al.[17] It was verified in the present work that the discrepancy in heat transfer coefficient could be as much as 20% according to the choice of ε-Ntu relation for data reduction.

From the measured inlet and outlet temperatures of water and air, water and air side heat transfer rates are calculated such that

$$Q_r=\dot{m}_rc_{pr}(T_{r,\mathrm{in}}+T_{r,\mathrm{out}}) \tag{2}$$

$$Q_a=\dot{m}_ac_{pa}(T_{a,\mathrm{out}}+T_{a,\mathrm{in}}) \tag{3}$$

In the above equations $Q, \dot{m}, c_p$ denote rate of heat transfer, mass flow rate and specific heat respectively. The rate of heat transfer is finally calculated by taking arithmetic average of air and water side such that

$$Q=(Q_a+Q_r)/2 \tag{4}$$

Several characteristic lengths, e.g, tube outer diameter, longitudinal tube spacing or hydraulic diameter, have been used for the fin-tube heat exchanger. However, it is hard to tell which one is the most appropriate, and it is likely to be chosen according to author's point of interest. In the present work hydraulic diameter(d_h) was chosen, which is based on the minimum free flow area(A_{min}) of air and defined as follows.

$$d_h=\frac{4A_{min}L}{A_a} \tag{5}$$

In the above equation, A_a denotes the total air side heat transfer area, and L is the length of heat exchanger along the flow direction. Reyn olds number is defined based on the maximum air flux(G_{max}) through the minimum free flow area and hydraulic diameter such that

$$\mathrm{Re}=\frac{G_{max}d_h}{\mu_a} \tag{6}$$

In the above equation μ_a denotes the viscosity of air. Also Nusselt number and Colburn j-factor is defined such that

$$\mathrm{Nu}=\frac{h_a d_h}{k_a} \tag{7}$$

$$j=\frac{\mathrm{Nu}}{\mathrm{Re}\,\mathrm{Pr}_a^{1/3}}=\frac{h_a}{G_{max}c_{pa}}\mathrm{Pr}_a^{2/3} \tag{8}$$

In the above equation k_a, Pr_a denote the thermal conductivity and Prandtl number of air respectively. From the measured air pressure drop between the inlet and outlet of heat exchanger the fanning friction factor is calculated using the following equation.

$$f=\frac{\rho_a d_h \Delta P}{2G_{max}^2 L} \tag{9}$$

In the above equation ρ_a denotes the density of air.

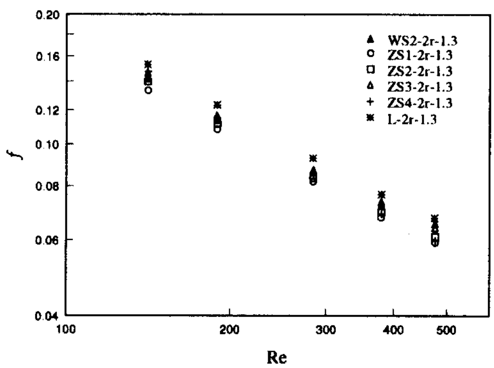
4. Results and Discussion

Uncertainty of friction factor data in the present work was estimated to be about 10% with 95% confidence level, and it was mostly dependent upon the accuracy of pressure differential gauge across heat exchanger.

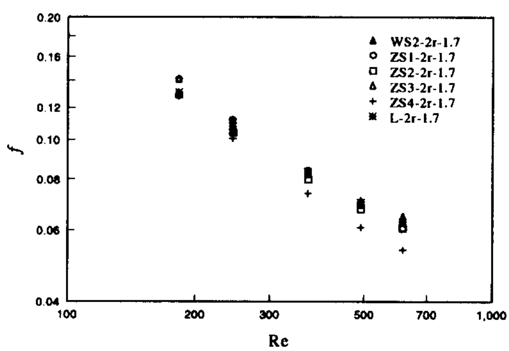
Uncertainty of water and air side heat transfer rate was estimated to be 5% and 2.5% respectively. However, uncertainty of heat transfer coefficient was found to be largely dependent upon the uncertainties of water side and contact heat transfer coefficients cited from the references rather than the measured heat transfer. Uncertainty of water side heat transfer coefficient was 10%, however uncertainty of contact heat transfer coefficient was not stated in the reference. Assuming the uncertainty of contact heat transfer coefficient is 10%, the uncertainty of the heat transfer coefficient data was estimated to be about 12%. Ratio of water side heat transfer to air side ranged from 96% to 100%, which went higher consistently as the heat transfer rate increased.

In Figs. 3 and 4 respectively the friction factors and heat transfer coefficients of the ZS fins and L fin are compared with WS2 fin data of Youn et al.[13] WS2-2r-1.3 denotes WS2 fin, 2 rows and fin spacing 1.3 mm. Looking at Fig. 3, the friction factors did not show a discernable differences among the fins considering the uncertainty of the results. In case of fin spacing 1.3 mm, the friction factor of the ZS1 fin is about 7% lower than that of the WS2 fin, which is the lowest, and L fin showed the highest friction factor(3~8% higher than the WS2 fin). In case of fin spacing 1.7 mm, the friction factor of the ZS4 fin is 1-17% lower than that of the WS2 fin, which is the lowest, and the ZS1 and ZS3 fins showed about 9% higher friction factor at lower velocity, but almost the same as WS2 fin at higher velocity.

It is demonstrated in Fig. 4 that the heat transfer coefficients of the ZS fins and L fin are at almost the same level showing a considerable(about 20-25%) improvement compared with WS2 fin. ZS4 fin showed the best performance in heat transfer at the air velocity of 1 m/s, whose heat transfer coefficient is about 25% higher than that of the WS2 fin. ZS4 fin showed relatively better performance than the other fins especially at lower velocity. ZS4 fin has only 9 slits which is the fewest and slit height is 0.8 mm which is the

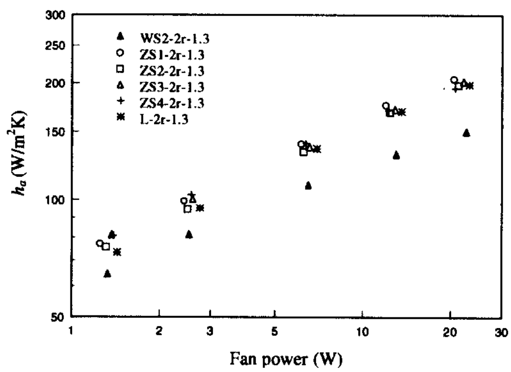


(a) Fin spacing 1.3 mm

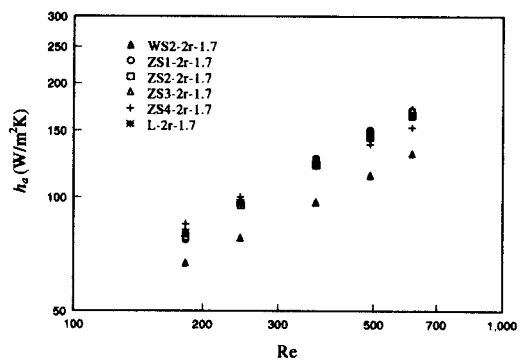


(b) Fin spacing 1.7 mm

Fig. 3 Comparison of friction factor for ZS1-ZS4 and L fins with WS2 fin.



(a) Fin spacing 1.3 mm



(b) Fin spacing 1.7 mm

Fig. 4 Comparison of heat transfer coefficient for ZS1-ZS4 and L fins with WS2 fin.

highest. From this result it can be said that the increase in number of slits does not necessarily guarantees improvement of heat transfer, but the appropriate choice of number of slits and size is important.

To evaluate overall performance of heat exchanger, it is necessary to consider heat transfer and pressure drop at the same time, and the concept of fan power is introduced, which is defined as a multiplication of pressure drop and air volume flow rate such that

$$\text{Fan power} = u_{\text{fr}} A_{\text{fr}} \Delta P \qquad (10)$$

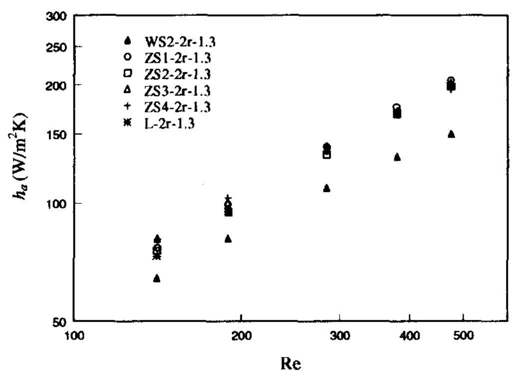
In the above equation u_{fr} and A_{fr} denote the frontal velocity of air and the heat exchanger frontal area respectively.

In Fig. 5 the heat transfer coefficients are compared with respect to the fan power. It is demonstrated in Fig. 5 that the performance of the ZS fins and L fin considerably improved compared with the WS2 fin, but the differences among themselves are indiscernable within the accuracy of the test results. Strictly speaking, ZS4 fin showed relatively a little better performance at lower velocity than the other fins. In general, enhancement of heat transfer by modifying fin surface is usually accompanied by an increase in pressure drop. However, the ZS fins and L fin proposed in the present work demonstrated an improvement of more than 20% in the heat transfer coefficient compared with the WS2 fin while maintaining almost the same level of pressure drop.

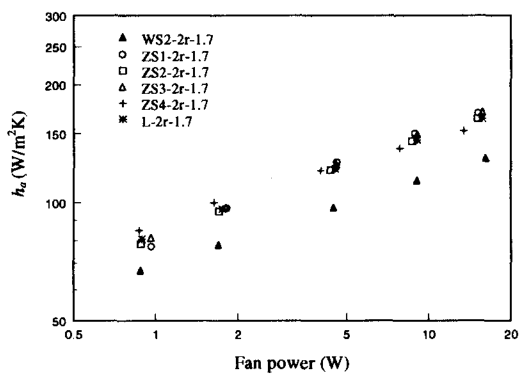
Considering number of slits for ZS fins and L fin, they were expected to show higher pressure drop than the WS2 fin, however it was almost the same. It is thought to be because the wavy fin surface only increases pressure drop making little contribution to heat transfer enhancement.

The ZS2 fin was modified from the ZS1 fin to reduce flow resistance and induce more air flow around tube by removing the 4 side slits. However, the friction factor of the ZS2 fin is 0.8% lower than ZS1 fin on an average, and the heat transfer coefficient is lower by 3%. Thus it is thought that the ZS2 fin did not show the intended design purpose.

The ZS3 fin was aimed to improve heat transfer performance by increasing the slit height of ZS2 fin from 0.6 mm to 0.7 mm. The friction factor and heat transfer coefficient of the ZS3 fin is about 5% and 4% respectively higher than the ZS2 fin on an



(a) Fin spacing 1.3 mm



(b) Fin spacing 1.7 mm

Fig. 5 Heat transfer coefficient vs. fan power.

average. Fig. 5 indicates that the ZS3 is a little better in its overall performance than the ZS2 fin, however the difference is thought to be indiscernable.

The L fin was expected to show relatively lower pressure drop than the others because the louver height is only 0.3 mm. However Fig. 3 indicates that the friction factor of the L fin is rather higher than the others in case of fin spacing 1.3 mm, and it did not show any desirable characteristic.

5. Conclusion

In the present work air side performance of various enhanced fins, 4 of which are with radial zig-zag slits and 1 with radial louvers, for ϕ9.52 fin-tube heat exchanger was experimentally investigated and compared with baseline data of wave-slit fin. All the fins studied showed about 20-25% improvement in heat transfer coefficient while maintaining pressure drop almost at the same level.

The comparison of ZS4 fin with the other fins showed that an increase in the number of slits does not necessarily improve heat transfer. Combination of appropriate number of slits and slit height is thought be important. Also the comparison of ZS1-ZS3 fins showed that the removal of side slits and increase in slit height did not make any contribution to overall performance.

The various fins studied in the present work demonstrated considerably better performance than the wave-slit fin. However, there was almost no observable difference among themselves.

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