Effect of Spacers on Perforated Plate Matrix Heat Exchanger Plate Surfaces

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***Abstract*— In the present work, copper perforated plate matrix heat exchanger (MHE) surfaces with paper spacers have been evaluated experimentally for heat transfer and flow friction characteristic by conducting single blow transient test. The transient experiment has been conducted with perforated plates having different spacer to plate thickness ratios (s/l) (0.5, 1.0 and 2.0). The test is conducted with perforated plates having rectangular and triangular perforations. The heat transfer and friction characteristics like Colburn factor (j) and Fanning friction factor (f) are plotted against Reynolds Number for various perforated plates.**

Keywords—Perforated plate matrix heat exchanger; single blow transient test; Colburn factor; friction factor; area goodness factor

I. INTRODUCTION

Heat exchangers are among the most vital components of any cryogenic refrigeration / liquefaction system and for these applications, heat exchangers should possess very high effectiveness. The performance of refrigerators, liquefiers and separation units is strongly dependent on the effectiveness of the heat exchangers used. If the effectiveness of the heat exchanger is below a certain critical value (less than 85%) [1], most cryogenic processes would cease to function. A decrease in heat exchanger effectiveness from 97% to 95% reduces the liquefaction by 12% [2]. The low values of attainable coefficient of performance (COP) [3] and the resulting high cost of refrigeration make it economically sensible to use a more effective and expensive heat transfer equipment. Apart from having a high effectiveness, cryogenic heat exchangers also need to be very compact, i.e. they must accommodate a large amount of surface area in a small volume. This helps in controlling heat exchange with the surroundings by reducing exposed surface area. Besides, a small mass means a smaller cooling load and a faster cooling time for refrigerators. This requirement is particularly important for small refrigerators operating at very low temperature. The requirement to attain high effectiveness and high degree of compactness together in one unit led to the invention of matrix heat exchangers [4].

Perforated plate matrix heat exchangers (MHE) essentially consist of a stack of perforated plates made of high thermal conductivity material alternating with spacers made of low thermal conductivity material. The stack of alternate high and low thermal conductivity materials is bonded to form a monolithic block. A schematic diagram of perforated plate MHE is shown as Fig. 1. The gaps in between the plates ensure uniform flow distribution (by continuous reheadering) and create turbulence which enhances heat transfer. Small perforations (diameter ranging from 1.5 mm to less than 0.4 mm) are made into the plates so that a large heat transfer coefficient and high surface area density (up to 6000 /m) is achieved.

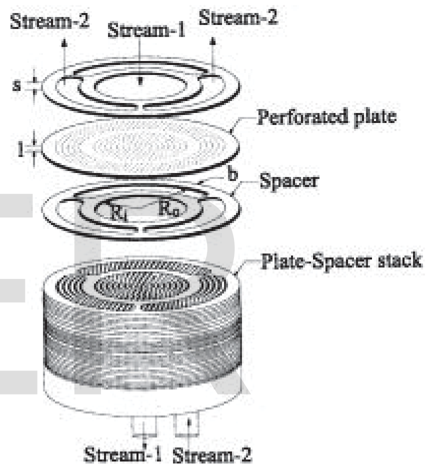

Fig. 1: Perforated plate matrix heat exchanger

The ratio of the plate thickness (length of the hole in the plate) to the diameter of the hole is on the order of 0.75, therefore, the thermal and hydrodynamic boundary layers do not become fully developed within the perforations, which results in high heat transfer coefficients and correspondingly high friction factors [5]. The flow within the small holes in the perforated plate is generally laminar. Due to their small hydraulic diameter and the low density of gases, the surfaces are usually operated in the Reynolds number range $500 < Re < 1500$ [6]. The spacers, being of low thermal conductivity material, also help in reducing axial conduction and consequent deterioration of performance. The spacers perform multiple roles such as reducing the longitudinal heat conduction through the walls, reducing the flow maldistribution by reheadering the flow in each spacer, interrupting the boundary layer and thus enhancing the heat transfer coefficients.

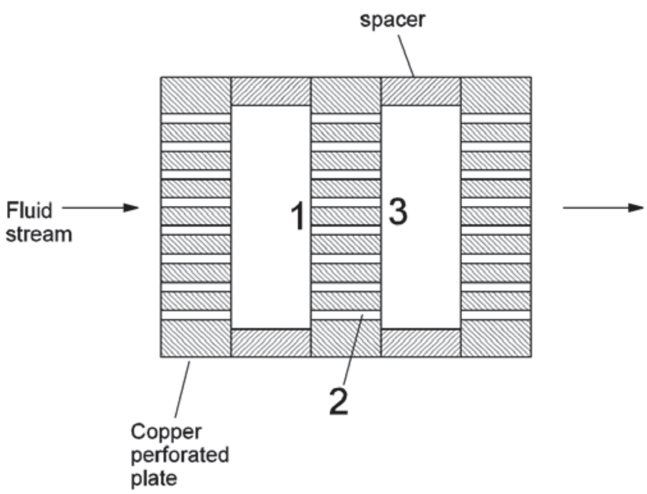

Fig. 2: Convective heat transfer area of perforated plate

Conventional effectiveness NTU approach methods cannot be used for MHEs because of the discontinuity in the heat transfer surface in the axial (longitudinal) direction [7]. New rating and sizing methods have been developed [8] that treat the MHE as a distinct set of plate - spacer pairs. The convective heat transfer in MHE has been considered by a number of authors [9, 10]. The single blow transient test is most appropriate for determining the heat transfer coefficients of perforated plate MHEs. Lot of methods have been developed in literature for calculating the heat transfer coefficients from the single blow test data and to take into account the non-ideal conditions that exist during the testing.

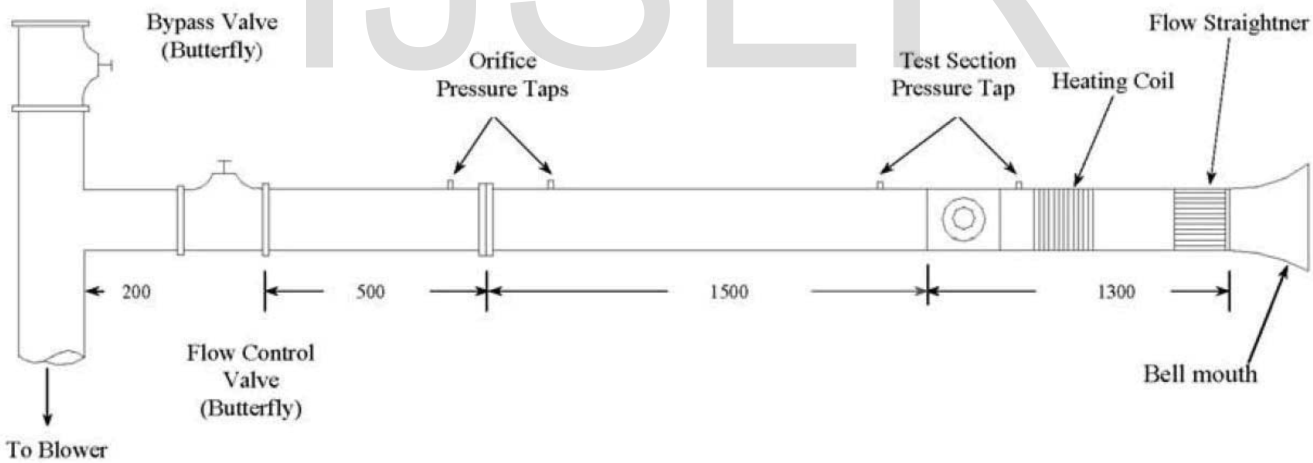

Fig. 3: *Experimental setup to conduct single blow transient test*

The single blow test consists of flow through a wind tunnel with facility to vary flow parameters to observe changes in the flow. In this kind of test usually only one fluid is allowed through the wind tunnel or channel and hence the named as single blow transient test and the characteristics of plate surfaces of the heat exchanger is studied. The convective heat transfer and flow friction characteristics are strongly depending on a number of geometric parameters such as plate porosity, spacer thickness plate thickness, perforation diameter, shape of perforations etc. The overall heat transfer coefficient depends on the convective heat transfer area, which consists of
(i) unperforated region of any plate facing the flow (ii) the unperforated region at the back of each plate, in the wake region of the flow and (iii) the tubular portion of each plate as referred in Fig. 2. Many researchers [11, 12, 13] have expressed the heat transfer coefficient as a function of porosity of matrix alone.

In this paper, the results obtained with different s/l ratios for both rectangular and triangular perforated plate geometry at same porosity using transient single blow tests are presented. An improved maximum slope method has been used to reduce the test data.

II. EXPERIMENTAL DESCRIPTION

Experimentations were carried out in an open circuit wind tunnel particularly designed for measuring the heat transfer and flow friction characteristics of perforated plate matrix heat exchanger surface. The schematic diagram of the investigational system is shown in the Fig. 3. The wind tunnel of 50 mm diameter is coupled to the suction side of a centrifugal blower. A bell mouth is used at the inlet of the tunnel for distortion free suction. The flow straightners compose the flow uniform. The flow from beginning to end of the tunnel is controlled by means of a butterfly valve. For uniform heating the matrix, spirally coiled nichrome coil is used as heater, which is controlled by an autotransformer. The rate of flow is measured using Fluke 992 Air flow meter. The temperature at the inlet and outlet of the test matrix is measured using two T- type thermocouples of size 32 gauge. The temperature difference between the values measured by thermocouples is within the measurement accuracy (0.1K). A data acquisition system, Agilent 34970A was used to record the temperature history for the period of transient testing method. The plates were detained jointly by mechanical pressure, and sealed on the outside by means of adhesive tapes to put a stop to air leakage. The entire stack was insulated with a 2 mm thick polyurethane insulation. The plates were stacked randomly.

III. EXPERIMENTAL METHODOLOGY

The experimentation course of action consists of varying the temperature of air flow to the test section abruptly and observing the thermal response at the outlet of the test section (perforated plate matrix heat exchanger). A sudden temperature perturbation is provided at the inlet section of the fluid flow and the temperature history of the single blow transient test method is obtained using the data acquisition system. This experimental method normally use single fluid flow throughout the wind tunnel and therefore known as single blow transient test method. The heat transfer coefficient is obtained from the exit temperature response. Different methods such as the maximum slope method, curve matching method etc, have been opted in literature to find out the heat transfer coefficient from the test data. In the current work, a modified maximum slope method in which both the maximum slope and the time at which maximum slope is used concurrently to forecast the heat transfer coefficient as well as the longitudinal heat conduction parameter experimentally. The longitudinal heat conduction parameter is experimentally forecasted in this work. The contact resistance between the plate and spacers will differ from experiment to experiment. Heat transfer coefficient of the test core is forecasted by matching the slope and time at maximum slope of the theoretical and exit temperature variations.

The experimentation is done to examine the effect of perforation diameter and porosity on the heat transfer characteristics. The details of plate sets are tabulated in table 1. The perforated plates, shown in figure 4, used were 50mm diameter and made of copper using photo chemical milling process. Paper spacers was used the paper spacers were made using die punch sets. The s/l ratios used were 0.5, 1.0 and 2.0. Thirty number of plate - spacer pair is stacked together to form the test matrix. Porosity of the perforated plates are kept constant as 0.3.

TABLE I. SPECIFICATION OF PLATE SET USED FOR EXPERIMENTS

Plate set	s/l ratio	Perforation shape
1	0.5	Rectangular
2	1.0	Rectangular
3	2.0	Rectangular
4	0.5	Triangular
5	1.0	Triangular
6	2.0	Triangular

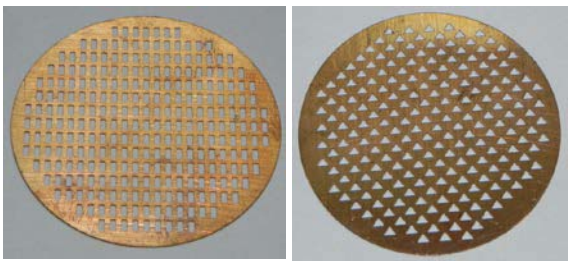

Fig. 4 : Rectangular and triangular perforated plate

The experiment was conducted for Reynolds number in the range of 50 to 1300. Reynolds number is based on the perforation diameter. A varying input was provided at the inlet section and the exit temperature response curve was obtained. The slope of this curve was calculated and the maximum slope and corresponding time at the maximum slope were noted. In this experiment, modified maximum slope method [14] was used to estimate the heat transfer coefficients. Based on the modified maximum slope method, the number of transfer unit (ntu) values were obtained. The corresponding Colburn factor (j) was calculated from ntu using governing Eq. (1-6). The governing equations for the solid and fluid can be obtained in any plate by an energy balance across a control volume as follows, where n denotes number of plates and j denotes j^{th} plate. Eq. (1-7):

For n=1,

$$ntu\ \frac{\partial \tau_1}{\partial \varsigma} + \lambda_p(\tau_1 - \tau_2) = \theta_1 - \theta_2 \tag{1}$$

$$\theta_2 = \theta_1 - \varepsilon_p(\theta_1 - \tau_1) \tag{2}$$

For $2 \leq j \leq n$,

$$ntu\ \frac{\partial \tau_j}{\partial \varsigma} + \lambda_p(2\tau_j - \tau_{j-1} - \tau_{j+1}) = \theta_j - \theta_{j+1} \tag{3}$$

$$\theta_{j+1} = \theta_j - \varepsilon_p(\theta_j - \tau_j) \tag{4}$$

For j=n,

$$ntu\ \frac{\partial \tau_n}{\partial \varsigma} + \lambda_p(\tau_n - \tau_{n-1}) = \theta_n - \theta_{n+1} \tag{5}$$

$$\theta_{n+1} = \theta_n - \varepsilon_p(\theta_n - \tau_n) \tag{6}$$

where $\tau = \frac{(T - t_{in})}{(T_0 - t_{in})}$, $\theta = \frac{(t - t_{in})}{(T_0 - t_{in})}$,

$$\lambda_p = \frac{k_s A_c}{smc_p},\ \varsigma = \frac{(mc_p)}{(MC)}\xi$$

$$\varepsilon_p = 1 - \exp(-ntu),\quad ntu = \frac{(hA)}{(mc_p)} \tag{7}$$

The basic performance data for a perforated plate matrix heat exchanger surface are often shown as curves of the Colburn factor ($j = St.Pr^{2/3}$), and the Fanning friction factor (f), plotted versus Reynolds number. The exit temperature response is used for the experimental determination of ntu and λ. Dimensionless temperature and dimensionless time are determined. The slope is determined from the dimensionless temperature and time plot. The maximum slope and time at which the maximum slope occurs are used in this method for the determination of ntu and longitudinal heat conduction factor.

The heat transfer characteristics data is presented in the form of Colburn factor (j), vs Reynolds number (Re) and the flow friction data is presented in the form of fanning friction factor (f), vs Reynolds number (Re). The fanning factor (f) was determined from isothermal pressure drop data and both factors (j and f) were determined from the following expressions (Eq. 8):

$$j = ntu\frac{Ac}{A}\Pr^{\frac{2}{3}} \quad \text{and} \quad f = \frac{2\Delta p\rho d}{4LG^2} \tag{8}$$

IV. RESULTS AND DISCUSSION

The experiment was conducted for different s/l ratios for triangular and rectangular shaped perforated plates having same porosity. The j, f and j/f plots against Re for rectangular perforated plates having porosity 0.3 and l/d ratio 0.25 were plotted in fig. 5, 6 and 7. The j, f and j/f plots against Re for rectangular perforated plates having porosity 0.3 and l/d ratio 0.25 were plotted in fig. 8, 9 and 10. The values of j as well as f are found to be decreasing as Reynolds number increases. As the values of Re increases, the residence time available for the fluid inside the matrix for transferring heat to the plates reduces and hence the heat transfer coefficient decreases. Since the friction factor is inversely proportion to the square of the velocity, the friction factor reduces with increase in Reynolds number. It is found that the area goodness factor, j/f is above the value of 0.1 as in fig. 7 and 10. This shows that at higher Re, performance of the heat exchanger increases. The area goodness factor (j/f) is inversely proportional to the square of flow area. An increase in j/f indicates a decrease in flow area and this leads to a more compactness for the heat exchanger. This aspect is highly desirable for a heat exchanger.

The effect of s/l ratio on the heat transfer and flow friction characteristics has been studied. Three triangular perforated plate and rectangular perforated plate with different s/l ratio with same porosity are selected for analysis. The j and f decreases as the spacer thickness increase. Jet impingement heat transfer decreases with an increase in the distance between the spacer to impingement surface ratio. Lytle et al. [15] provided a correlation for average local heat transfer coefficients as

$$Nu = 0.424\ Re^{0.57}\left(\frac{s}{l}\right)^{-0.33} \tag{9}$$

The value of j decreases as the spacing between the plate increases for MHE. An increase in spacer thickness results in the increase in overall volume without any increase in heat transfer area. An increase in spacer thickness leads to an increase in weight of the heat exchanger and a decrease in its compactness. But the overall longitudinal conduction decreases with an increase in spacer thickness.

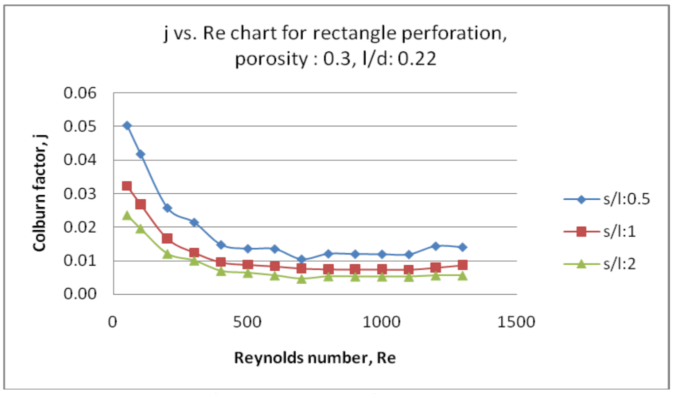

Fig. 5: j vs. Re plot for rectangular perforation with p 0.3 and l/d 0.22

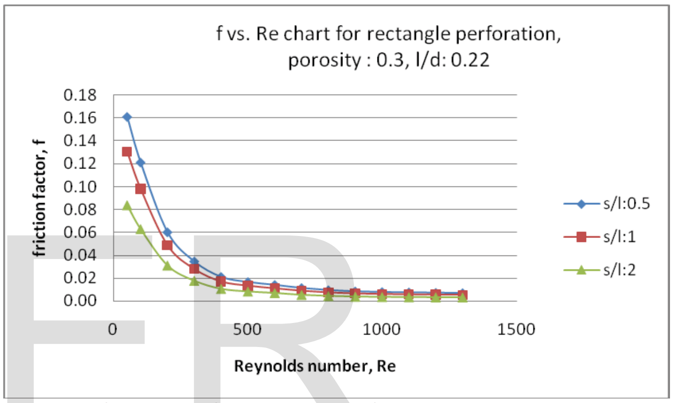

Fig. 6: f vs. Re plot for rectangular perforation with p 0.3 and l/d 0.22

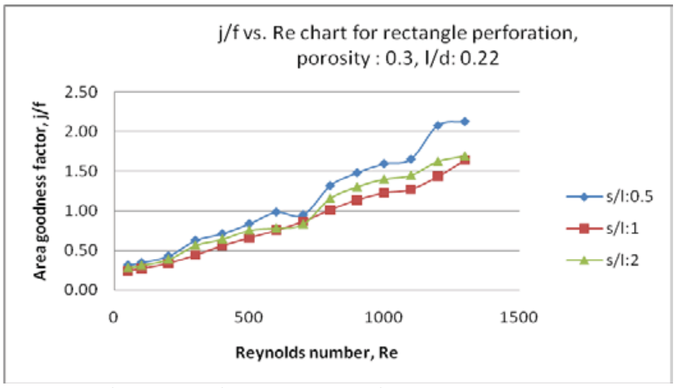

Fig. 7: j/f vs. Re plot for rectangular perforation with p 0.3 and l/d 0.22

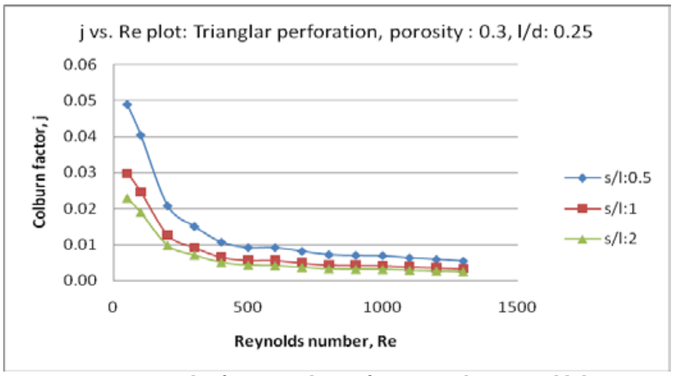

Fig. 8: j vs. Re plot for triangular perforation with p 0.3 and l/d 0.25

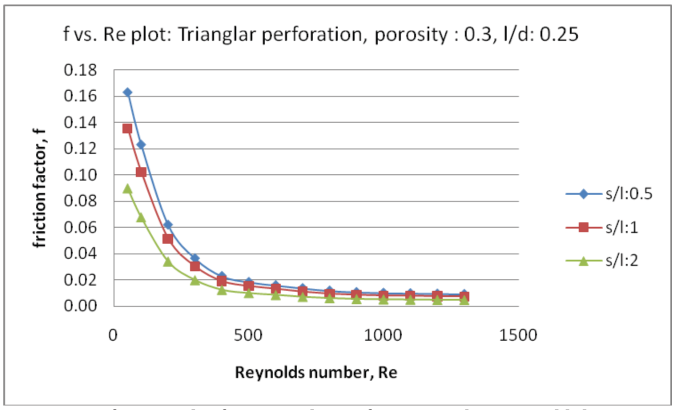

Fig. 9: f vs. Re plot for triangular perforation with p 0.3 and l/d 0.25

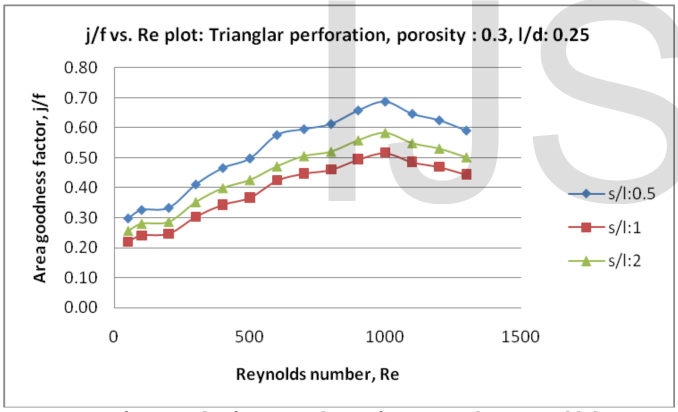

Fig. 10: j/f vs. Re plot for triangular perforation with p 0.3 and l/d 0.25

V. ERROR ANALYSIS

Based on the uncertainties in the measurements, the uncertainty in the estimation of j was 10% and for friction factor, 2.4 %. This was calculated by the Kline and McClintock method. In this work, velocity, static pressure, fluid temperature etc. are the measure qualities and the error in this work is due to the uncertainty in the measurements and the uncertainty in the value of physical properties. The uncertainty bands that may be placed on the reduced data (j and f) are functions of the uncertainties in the measurements (pressure difference, temperature and temperature difference, ambient pressure and time) made during the experiments.

VI. CONCLUSION

The effect of shape of perforations on heat transfer and flow friction characteristics is studied for various perforation diameter and porosity.

1. It is observed that Colburn factor j decreases as porosity and perforation diameter increases.
2. As Reynolds number increases, j decreases as the residence time for the heat transfer decreases.
3. As Reynolds number increases, the value of f also decreases.
4. An increase in j/f indicates a decrease in flow area and this leads to a more compactness for the heat exchanger.

Nomenclature

A	Heat transfer area, m^2
A_c	Cross sectional area , m^2
C	Specific heat of the matrix, J/kgK
c_p	Specific heat of fluid, J/kgK
d	Perforation diameter, m
f	Fanning friction factor
j	Colburn factor
j/f	Area goodness factor
M	Mass of the matrix, kg
m	Fluid flow rate, kg/s
ntu	Number of Transfer Units of plate
G	Mass velocity of the fluid in the perforation
L	Length of the heat exchangers, m
Pr	Prandtl number ($\mu\ c_p$ / k)
Re	Reynold Number
S	Thickness of spacer, m
T	Temperature of the matrix, K
T_o	Temperature of test section at time, $\xi = 0$
t	Temperature of the fluid, K
t_{in}	Fluid temperature maintained at the entry of the bed, K
Δp	Fluid static pressure drop
ρ	Fluid density

Subscripts

p	Plate
f	fluid
in	Inlet

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