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Effect of partial bypass on the heat transfer performance of dehumidifying coils ☆

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ABSTRACT

This study extends a so called "partial bypass" concept to dehumidifying fin-and-tube heat exchangers. Tests are performed for a 2-row and 4-row fin-and-tube heat exchangers having plain fin configuration. The inlet dry bulb temperature is fixed at 25 °C while the inlet relative humidities are 50% and 80%, respectively. Test results showed that for a 2-row coil under RH = 50%, the corresponding heat transfer augmentation ratio (Q_R) and pressure penalty ratio (P_R) decrease with the rise of bypass ratio (BR). At a smaller frontal velocity, the regime of appropriate bypass ratio where $Q_R > 1$ and $P_R < 1$ is more apparent. The effect of partial bypass decreases as the velocity increases. As the velocity gets close to 2 m s^{-1}, it shows that the performance at high bypass ratio becomes more efficient. At the same time, when bypass ratio becomes lager, the tendency of the decreasing of ΔP is more drastic than the decreasing of Q. Nevertheless, not for all the cases that the performances get better as the bypass ratio increases. For V_{fr} = 4 m s^{-1}, the experiment results show that the overall effect is less efficient than at V_{fr} = 2 m s^{-1}. The major reason is caused by the flow pattern. The rise of bypass airflow is quite large which may act as an air curtain to distort the main airflow, and result in higher pressure drop. This implies that the design of the bypass design is quite imperative in the practice of real applications. For a 4-row heat exchanger, the experiment data of Q_R and P_R for the 4-row coil performs better than a 2-row coil.
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1. Introduction

The air-cooled heat exchangers are widely used in a variety of industrial, commercial, and residential applications as coolers, heaters, evaporators and condensers. The most common configuration of the air-cooled heat exchangers takes the form as fin-and-tube where air normally flows along the fin (outside) and the other working fluids flow in the tube side. Depending on the applications and operational conditions, the air side thermal resistance can be as high as 70–90% of the total thermal resistance. As a consequence, it is imperative to reduce the associated resistance as far as the heat exchanger efficiency is concerned. The most effective reduction of the thermal resistance is via increasing voluminous surface but it also accompanies with gigantic pressure drop penalty. Hence, most studies had focused on increasing the air side heat transfer coefficient via enhanced geometries such as wavy, slit, louver, and vortex generator geometries, and some ingenious designs regarding to the fin patterns were reviewed by Wang [1,2].

Depending on the applications, multiple tube rows can be employed to increase surface area for increasing heat transfer rate. With multiple tube row designs, though the surface area increases linearly with the tube row, the heat transfer rate, unfortunately shows only moderately or only marginally increase with respect to the tube row [3,4]. This phenomenon becomes more and more severe when the number of tube row is further increased [5]. To make thing even worse, the pressure drop penalty is proportional to the number of tube row, meaning a linear increase of pumping power. It is quite often to employ air-cooled heat exchangers having deep row configuration. For instance, the commonly used ventilator, heat recovery system, and some large heat exchanging facility take form as deep row coils. Of course, one can always use highly interrupted surfaces to augment airside performance to offset the surface requirement, and accordingly a less number of tube row can be achieved. Nevertheless, highly interrupted surfaces, like louver, offset fin, slit fin, and the like are prone to fouling and are normally regarded inappropriate for industrial applications or large system [6].

In this regard, the present authors extend a newly proposed concept, "partial bypass", of heat transfer enhancement for this kind of air-cooled heat exchangers [7,8], and the concept can be extended to any highly compact heat exchangers as well. To explain the proposed concept for heat transfer augmentation, let's take the two samples from Wang et al. [8] using two air-cooled finned heat exchangers having a number

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Nomenclature

A	surface area (m^2)
BR	bypass ratio, defined in Eq. (1) (−)
c_p	specific heat (kJ kg^{-1} K^{-1})
C	heat capacity rate (W K^{-1})
$\dot{m}$	mass flow rate (kg s^{-1})
N	number of the tube row (−)
NTU	number of transfer unit, UA/C_{min} (−)
P_R	ratio of pressure drop (−)
Q_R	ratio of heat transfer rate (−)
Q	heat transfer rate (W)
RH	relative humidity (%)
T	temperature (K)
ε	effectiveness (−)
ΔT	temperature difference (K)
ΔP	pressure drop (Pa)
U	overall heat transfer coefficient (W m^{-2} K^{-1})
V_{fr}	frontal velocity (m s^{-1})

Subscript

1	first heat exchanger
2	second heat exchanger
air	air side
in	inlet
max	maximum value
min	minimum value
water	water side

of tube row of 4 and 8 respectively for illustration. A schematic showing the operation of these heat exchangers can be seen in Fig. 1(a) and (b). These two heat exchangers have the same longitudinal tube pitch, transverse tube pitch, fin collar outside diameter, and fin surface area and an identical frontal area, the inlet temperature of the air ($T_{air,in}$) and hot water ($T_{water,in}$) is also maintained at fixed temperatures, respectively. The heat exchangers are arranged in pure cross flow configuration with 4 and 8 circuitries for the tested 4-row and 8-row heat exchangers (see Wang et al. [8] for detailed circuitry arrangement). For a frontal velocity (V_{fr}) of 2 m s^{-1}, the corresponding heat transfer rate and pressure drop for the 4-row heat exchanger is about 4540 W and 43.2 Pa while the heat transfer rate is moderately increased to 5561 W accompanied with a nearly doubled pressure drop of 85 Pa for the 8-row coil. The results implicate adding more surfaces to augment heat transfer rate is comparatively ineffective. The results are not surprising for the maximum heat transfer rate ($Q_{max} = C_{min}(T_{water,in}–T_{air,in})$) for the heat exchangers are the same. Notice that the minimum capacity rate, normally C_{min} ($\dot{m}\,c_p$; $\dot{m}$: mass flow rate, c_p: specific heat), is on the airside for an air-cooled heat exchanger. Raising the number of tube row (N) leads to a rise of the number of transfer unit ($NTU = UA/C_{min}$, U: overall heat transfer coefficient, A: total surface area) and effectiveness ($\varepsilon = Q/Q_{max}$, Q: actual heat transfer rate).

The proposed concept is rather simple, focusing on increasing the Q_{max} at the rear part of the heat exchanger. For an illustration of this concept, Wang et al. [8] split the 8-row heat exchanger into two identical 4-row heat exchangers, and some spaces are between these two 4-row heat exchanger (some mixing mechanisms can be provided in the space if needed). However, the first heat exchanger is not tightly fitted into the duct area. Instead, some bypass area is

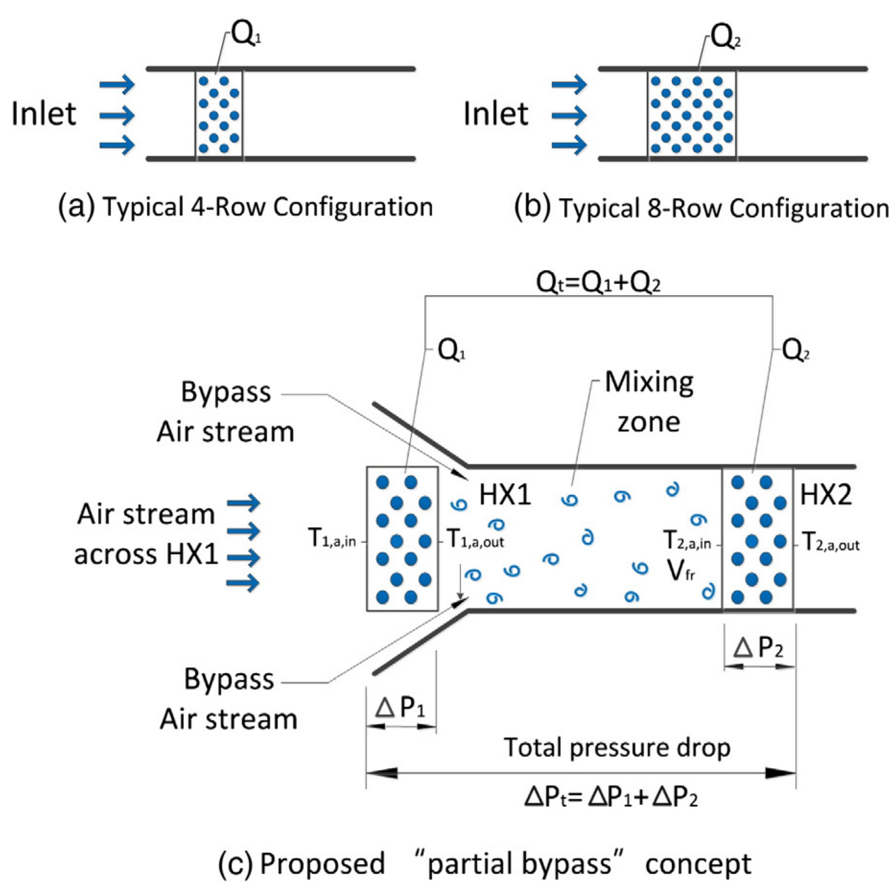
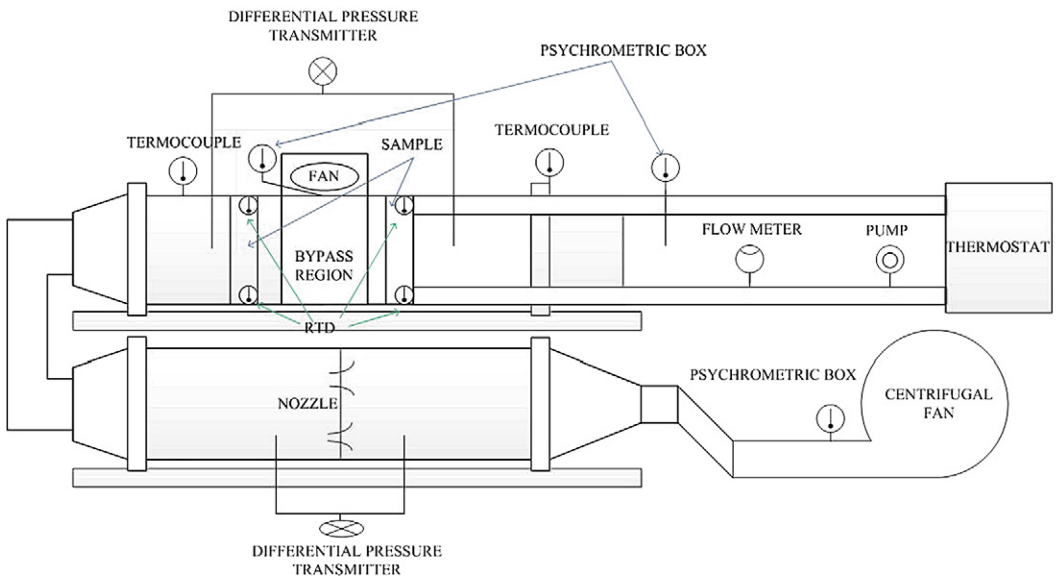
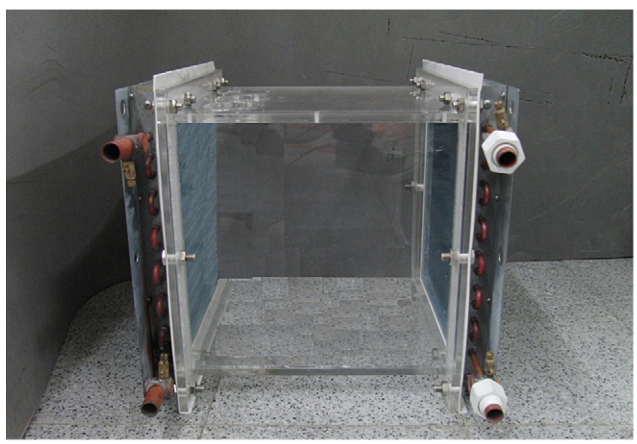


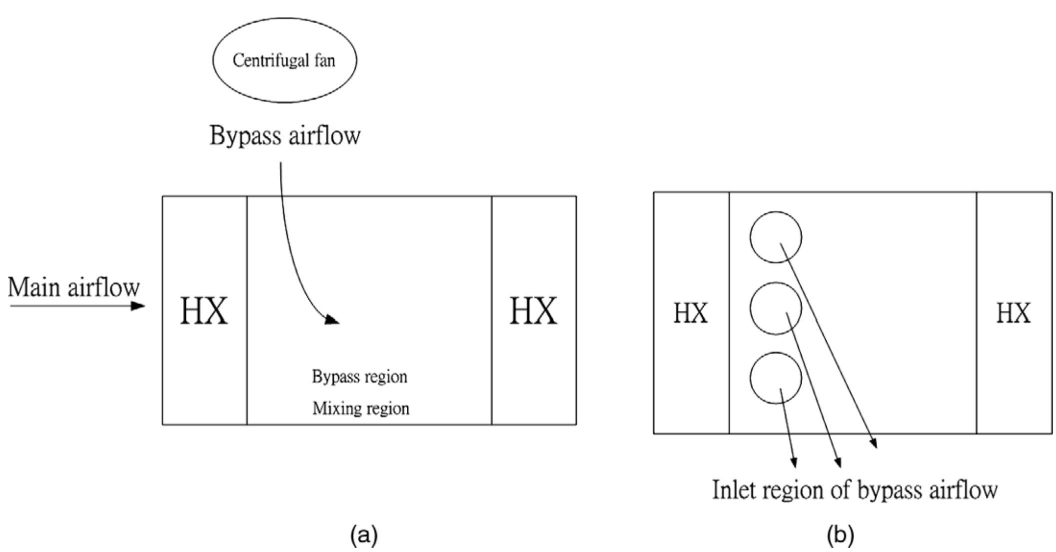
Fig. 1. Schematic of operation of the (1) 4-row heat exchanger; (2) 8-row heat exchanger; and (3) the proposed "partial bypass" heat exchanger.



(a) Schematic of the experimental setup



(b) Photo of the mixing device placed amid the first and the second heat exchanger.



(c) Schematic of the bypass flow into the bypass device.

Fig. 2. Schematic of the test facility and the partial bypass device.

designated as schematically shown in Fig. 1(c). The bypass ratio of the airflow across the first heat exchanger is designated as

$$\mathrm{BR} = \frac{\text{bypass flow rate (BF, bypass from the first heat exchanger)}}{\text{total flow rate (TF, evaluated at the second heat exchanger)}}. \quad (1)$$

With the foregoing design, the performance in the first part of the heat exchanger is degraded but the performance of the second part of the heat exchanger is appreciably enhanced. Wang et al. [8] had showed that the overall heat transfer performance is about the same but the pressure drop is significantly reduced. In this study, the present authors had extended the concept to examine the applicability in dehumidifying heat exchangers.

2. Experimental apparatus and data reduction

The schematic diagram of the experimental air circuit assembly is shown in Fig. 2(a). It consists of an open loop wind tunnel in which air is circulated by a variable speed centrifugal fan (3.7 kW, 5 HP). The air duct is made of steel sheet and has a 300 mm × 300 mm cross-section. The temperature and relative humidity of the environment are controlled by an environmental chamber. The air flow rate measurement station is set up with multiple nozzles. This setup is based on the ASHRAE 41.2 standard [9]. A differential pressure transducer is used to measure the pressure difference across the nozzles. The air temperatures at the inlet and exit zones across the sample heat exchangers are measured by three psychrometric boxes constructed based on the ASHRAE 41.1 standard [10]. Besides, the dry-bulb temperatures of the inlet and outlet air across the sample heat exchangers are also measured by pre-calibrated thermocouples, with a calibrated accuracy of ±1.1%. The pressure drop of the sample heat exchangers is measured by a differential pressure transmitter. The bypass device is made of acrylic plates as seen in Fig. 2(b) and is connected with the two heat exchangers. Bypass airflow is provided by another variable speed centrifugal fan (0.75 kW, 1 HP). The schematic of the bypass device is shown in Fig. 2(c). Main airflow is from the left side of the bypass device and the bypass airflow is from the top side of the device. Notice that the whole experimental setup is enclosed in an environmental chamber with controllable dry bulb and wet bulb temperatures.

The working medium on the tube side is cold water. A thermostatically controlled reservoir provides the cold water at selected temperatures. The temperature differences on the water side are measured by pre-calibrated RTDs. The water volumetric flow rate is measured by a magnetic flow meter. All the temperature measuring probes are resistance temperature device (Pt100), with a calibrated accuracy of ±1.3%. The test conditions of the inlet air are as follows:

Dry-bulb temperatures of the air: 25 ± 0.5 °C;
Inlet relative humidity for the incoming: 50 and 80%;
Inlet air velocity: From 0.5 to 4.0 $\mathrm{ms}^{-1;}$
Inlet water temperature: 2 ± 0.5 °C; and
Water mass flow rate inside the tube: 1–4 L/min.

3. Results and discussion

To illustrate the proposed partial bypass concept applicable to dehumidifying coils, experiments were conducted for 2- and 4-row heat exchangers subject to a dry bulb temperature of 25 °C and RH = 50 and 80%, respectively. Experiments were made with fixed frontal velocities into the second heat exchanger of 0.5, 1, 2, and 4 ms^{-1}, respectively. The results shown in the figure are in terms of ratios of heat transfer ($Q_R = Q_t/Q_2$ or $Q_R = Q_t/Q_4$) and of pressure drop ($P_R = \Delta P_t/\Delta P_2$ or $\Delta P_t/\Delta P_4$) where the subscripts 2 and 4 denote experimental results for the 2-row heat exchanger and 4-row heat exchanger having the same fin pitch respectively. Hence enhanced heat transfer is achieved when $Q_R > 1$. On the other hand, $P_R < 1$ denotes no pressure drop penalty. As far as the best performance is concerned, it would be better to achieve $Q_R > 1$ and $P_R < 1$ simultaneously. The experimental data of Q_R and P_R for the 2-row coil are shown in Fig. 3(a) and (b) under RH = 50%. As expected, both Q_R and P_R decrease with the rise of bypass ratio (BR). In the meantime, the region with $Q_R > 1$ and $P_R < 1$ depends on the frontal velocity. At a smaller frontal velocity, the regime of appropriate bypass ratio where $Q_R > 1$ and $P_R < 1$ is more apparent. Take $V_{fr} = 0.5\ \mathrm{ms}^{-1}$ for example, the value of Q_R is higher or almost equal to 1 and in the meanwhile P_R decreases nearly 6% at BR = 0–0.2. The results can be explained from the relation of ε-NTU as seen in Fig. 4. With a smaller frontal velocity, it is expected that the C_{min} is reduced. In the meantime, U is also reduced but the degree of reduction is smaller than that of C_{min}. In this sense, the number of transfer unit, NTU, actually increased, leading to a higher effectiveness ε. Therefore, ε can be easily improved via increasing the Q_{max} when it is operated at a higher value of NTU. In this regard, the region for $Q_R > 1$ increased. However, the effect of partial bypass decreases as the velocity increased. As the velocity gets close to 2 m s^{-1}, it shows that the performance at high bypass ratio becomes more efficient. At the same time, when the bypass ratio becomes lager, the tendency of the decreasing of ΔP is more drastic than the decreasing of Q. This effect shows that the major benefit of this operation is to reduce

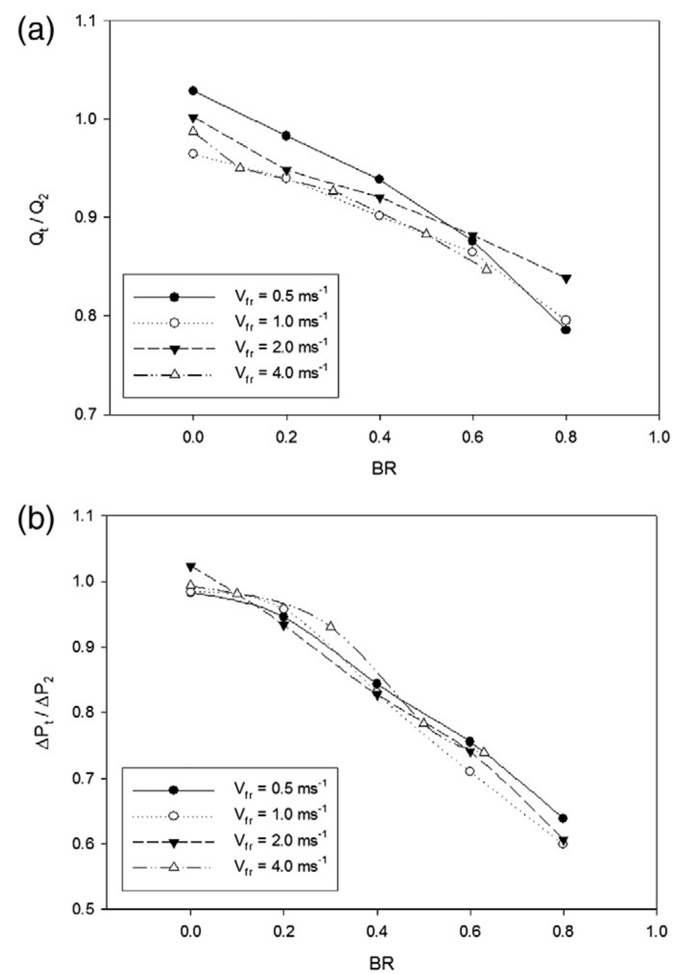


Fig. 3. (a) Experimental results for Q_R vs. bypass ratio for a 2-row coil under RH = 50%; (b) experimental results for P_R vs. bypass ratio for a 2-row coil under RH = 50%.

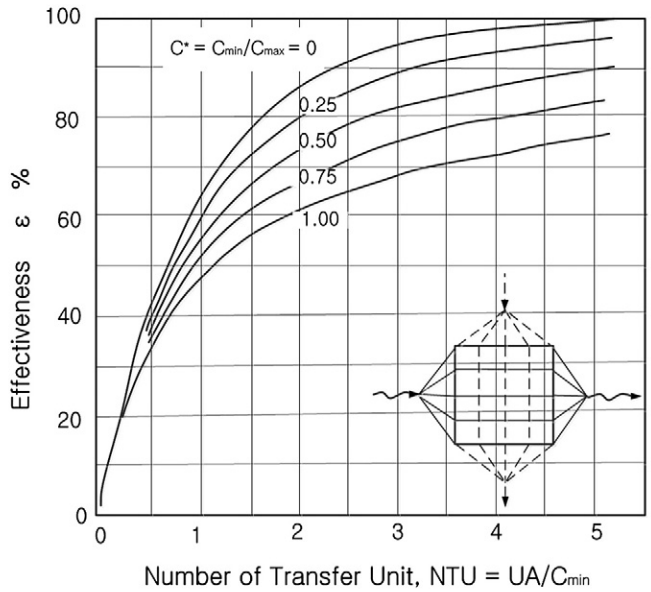


Fig. 4. Relation of the ε-NTU for unmixed/unmixed cross flow arrangement subject to variation of C*.

pressure drop. The obvious result for $V_{fr} = 2$ m s^{-1} is that even though Q decreased by 17%, the corresponding reduction of ΔP is about 40% at BR = 0.8.

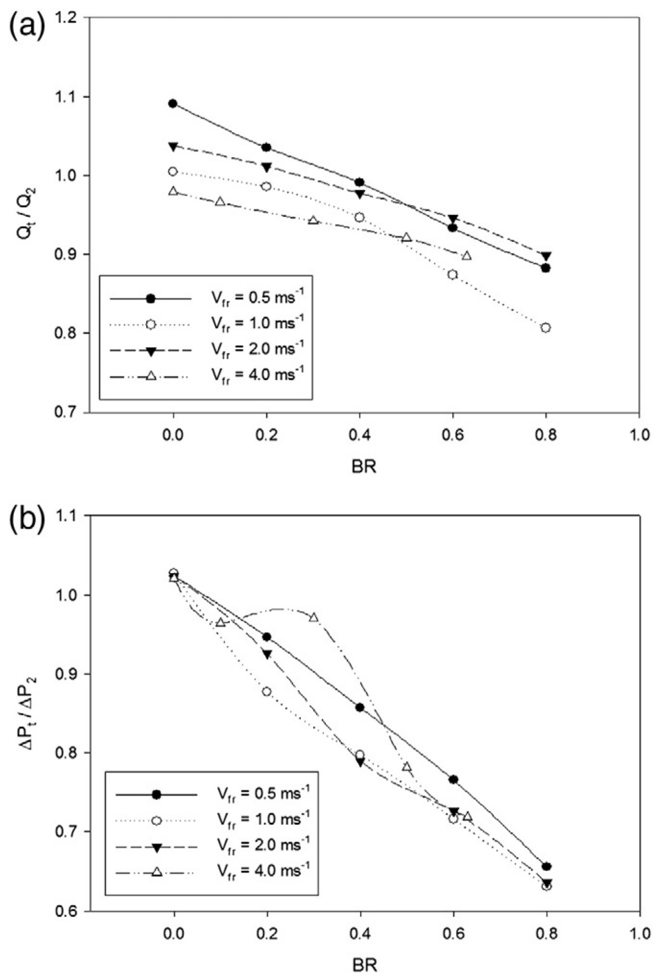


Fig. 5. (a) Experimental results for Q_R vs. bypass ratio for a 2-row coil under RH = 80%; (b) experimental results for P_R vs. bypass ratio for a 2-row coil under RH = 80%.

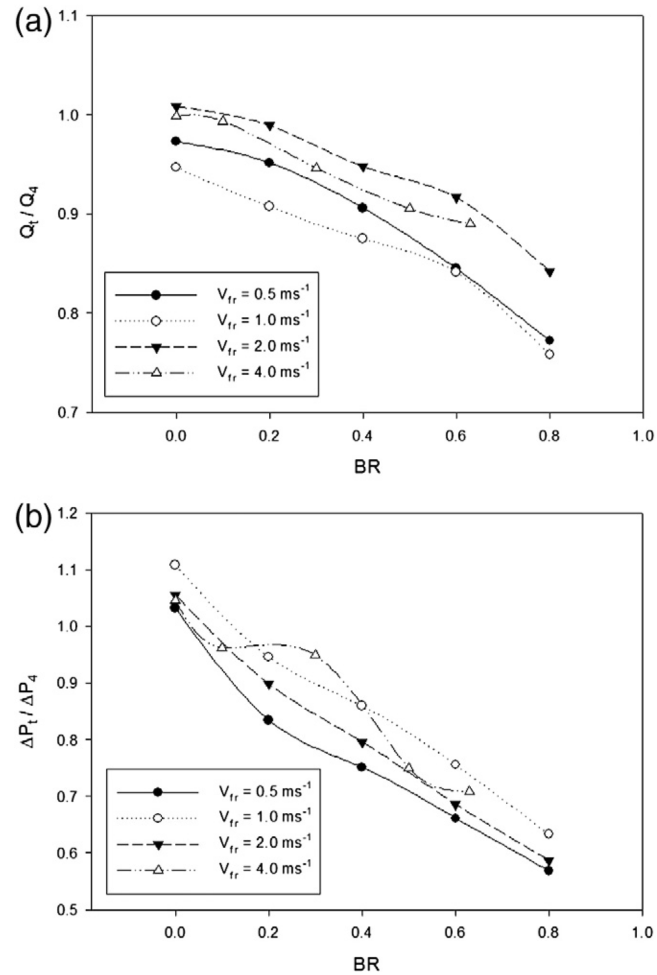


Fig. 6. (a) Experimental results for Q_R vs. bypass ratio for a 4-row coil under RH = 50%; (b) experimental results for P_R vs. bypass ratio for a 4-row coil under RH = 50%.

Nevertheless, it is not applicable for all the cases that the performances get better as the bypass ratio increased. For $V_{fr} = 4$ m s^{-1}, the experimental results show that the overall effect is less efficient than that at $V_{fr} = 2$ m s^{-1}. Moreover, at a low bypass ratio (BR = 0.1–0.3), ΔP is decreased slowly. This can be explained from the present bypass device which is shown in Fig. 2(c). The major reason is caused by the flow pattern being delivered. As shown in Fig. 2(c), bypass air flows into the mixing region from the top of the bypass device. Because the bypass airflow is quite large at $V_{fr} = 4$ ms^{-1}, it acts as an air curtain and inhibits the main air stream. Therefore, the main airflow is distorted by the bypass airflow and results in an uneven decline of pressure drop. This implies that the design of the bypass design is quite essential in the practice of real applications.

The experimental data of Q_R and P_R for the 2-row coil are shown in Fig. 5(a) and (b) under RH = 80%. Compared to the conditions with RH = 50%, the overall performances are better under RH = 80%. It's because the latent heat of the heat exchanger increases with the environmental relative humidity. Meanwhile the pressure drop of the heat exchanger increases as relative humidity increases. As a consequence, the decrease in pressure drop through the concept of partial bypass becomes more pronounced. Hence, it's apparent to see a better improvement under a high relative humidity. As seen in Fig. 5(a) and (b), for $V_{fr} = 0.5$ and 2 ms^{-1}, $Q_R > 1$ as well as $P_R < 1$ can be achieved at a lower bypass ratio. Especially for $V_{fr} = 0.5$ ms^{-1}, the pressure drop can reduce about 15% with the same heat transfer rate at BR = 0.4. The experimental results also show that there are better performances for $V_{fr} = 2$ ms^{-1} at high bypass ratio. Even though Q decreased by 11%, ΔP

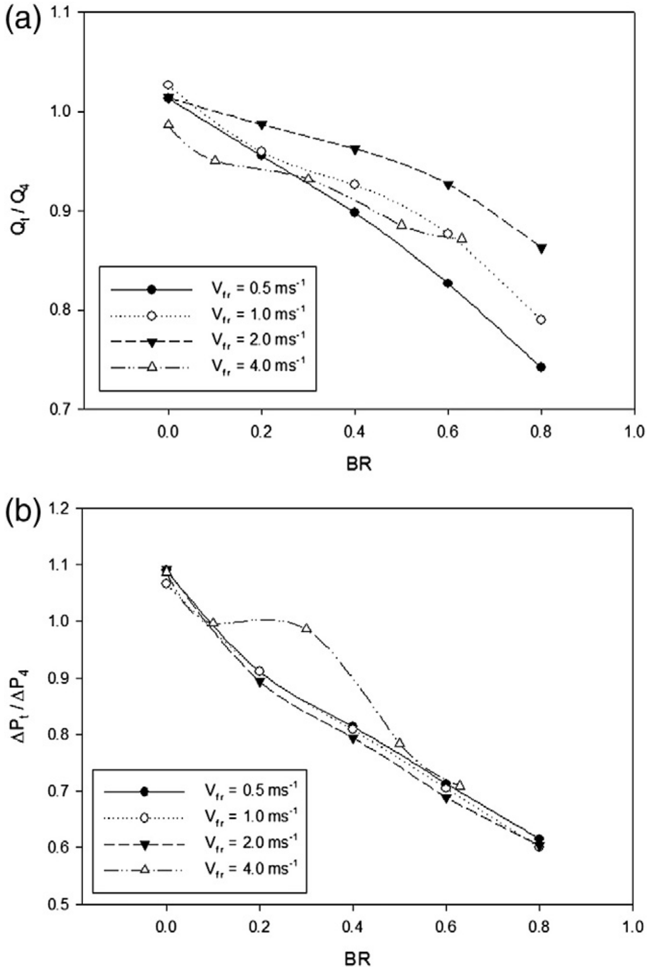


Fig. 7. (a) Experimental results for Q_R vs. bypass ratio for a 4-row coil under RH = 80%; (b) experimental results for P_R vs. bypass ratio for a 4-row coil under RH = 80%.

significantly decreased about 37% at BR = 0.8. On the other hand, the heat transfer rate can be augmented based on an identical pumping power criterion. Nevertheless, for $V_{fr} = 4\ ms^{-1}$, the experimental results also show that at a lower bypass ratio (BR = 0.1–0.3), ΔP remains almost the same. This is the same reason as mentioned from the previous explanation where the directed air flow may distort the main flow.

For a 4-row heat exchanger, the experimental data of Q_R and P_R for the 4-row coil are shown in Fig. 6(a) and (b) under RH = 50%. In general, the performances of a 4-row coil are better than a 2-row coil under RH = 50%. At a lower bypass ratio, basically the best situation is at $V_{fr} = 0.5\ ms^{-1}$. The heat transfer rate just decreased near 5%, but the pressure drop decreased about 17% at BR = 0.2. On the other hand, at a high bypass ratio, the best situation is at $V_{fr} = 4\ ms^{-1}$. For $V_{fr} = 4\ ms^{-1}$, even though Q decreases 16%, ΔP decreases about 42% at BR = 0.8. For a 4-row coil, the percentage of heat transfer rate occurs at first heat exchanger is more than that at second heat exchanger compared to a 2-row coil. According to the figures shown in Fig. 7(a) and (b), the overall performances of a 4-row HX under RH = 80% are similar to a 4-row HX under RH = 50% and is also similar to a 2-row HX under RH = 80%. For all the cases, $V_{fr} = 2\ ms^{-1}$ shows the better results than the other velocities.

4. Conclusions

In this study, the novel "partial bypass" concept had been extended to test its applicability in dehumidifying coils. Tests are performed for a 2-row and 4-row fin-and-tube heat exchangers having plain fin configuration. The inlet dry bulb temperature is fixed at 25 °C while the inlet relative humidities are 50% and 80%, respectively. Based on the foregoing discussion, the following conclusions are made:

(1) For a 2-row coil under RH = 50%, the corresponding Q_R and P_R decrease with the rise of bypass ratio (BR). At a smaller frontal velocity, the regime of appropriate bypass ratio where $Q_R > 1$ and $P_R < 1$ is more apparent. For instance, for $V_{fr} = 0.5\ ms^{-1}$, the value of Q_R is higher or almost equal to 1 and in the meanwhile P_R decreases nearly 6% at BR = 0–0.2.
(2) The effect of partial bypass decreases as the velocity increased. As the velocity gets close to $2\ m\ s^{-1}$, it shows that the performance at a high bypass ratio becomes more efficient. At the same time, when the bypass ratio becomes lager, the tendency of the reduction of ΔP is more drastic than the decreasing of Q.
(3) Nevertheless, not for all the cases that the performances get better as the bypass ratio increases. For $V_{fr} = 4\ m\ s^{-1}$, the experimental results show that the overall effect is less efficient than at $V_{fr} = 2\ m\ s^{-1}$. The major reason is caused by the flow pattern. The rise of bypass airflow is quite large which may act as an air curtain to distort the main airflow, and result in higher pressure drop. This implies that the design of the bypass design is quite imperative in the practice of real applications.
(4) For a 4-row heat exchanger, the experimental data of Q_R and P_R for the 4-row coil performs better than a 2-row coil.

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