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Impacts of Fouling and Cleaning on the Performance of Plate Fin and Spine Fin Heat Exchangers

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An experimental study was conducted to investigate the effects of air-side fouling and cleaning on the performances of various condenser coils used in unitary air-conditioning systems. A total of six condenser coils with different fin geometry and row number were tested. Performance tests were performed at three different conditions: clean-as-received, after fouling, and after cleaning. In all cases, it was observed that the fouling was mostly confined to the frontal face of the heat exchanger as reported in the previous investigations. The amount of deposited dust was more dependent on fin geometry for the single-row heat exchangers than for the double-row heat exchangers. The predominant effect of fouling was to cause a more significant increase in air-side pressure drop than a degradation in heat transfer performance. For the single-row heat exchangers, the pressure drop increased by 28 to 31%, while the heat transfer performance decreased by 7 to 12% at the standard air face velocity of 1.53 m/s depending on fin shape. For the double-row heat exchangers, the pressure drop increased by 22 to 37%, and heat transfer performance decreased by only 4-5% at the same air face velocity. Once the contaminated coils were cleaned according to the given cleaning procedure the original performance of the heat exchangers could almost be recovered completely. The pressure drop could be restored within 1 to 7% and the heat transfer performance could be recovered to within 1 to 5% of the originally clean heat exchangers. Therefore, it is concluded that a periodic application of the specified cleaning technique will be effective in maintaining the thermal performance of the condenser coils.

Key Words : Air-side Fouling, Condenser, Cleaning, Pressure Drop, Heat Transfer Performance

Nomenclature

Symbol

A : Total surface area [m^2]
C_p : Constant-pressure specific heat [J/kg·K]
h : Enthalpy [J/kg] or convective heat transfer coefficient [W/m^2·K]
m : Mass flow rate [kg/s]
ΔP : Pressure drop [Pa]
Q : Heat transfer rate [W]
R_f : Air-side fouling factor [m^{2}K/W]
R_m : Metal thermal resistance of external fins and tubes [K/W]
ΔT_m : Log mean temperature difference
T : Temperature [℃]
U : Overall heat transfer coefficient [W/m^2·K]
V : Air flow rate [m^3/s]
W_f : Weight of dust captured in the filter [kg]
W_t : Total amount of dust sprayed from the dust injector [kg]
Φ : Humidity ratio

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ξ_f : Efficiency of filter [%]
η : Fin efficiency
v : Specific volume of wet air [m^3/kg]

Subscript

a : Air-side
avg : Average
f : Fin or filter
i : Inlet
o : Outlet
n : Nozzle
w : Water-side

1. Introduction

Fouling may be defined as the formation of deposits on heat transfer surfaces. It is well known that fouling impedes heat transfer and increases pressure drop for a given flow rate to the flow of fluids over the surface (Taborek et al., 1972; Suitor et al., 1977; Bott, 1981). Engineers working in the heat transfer area have a particular interest in fouling since it significantly impacts the performance and lifetime of heat transfer equipment. If a deposition occurs on a solid surface in the presence of a dirty gas stream especially, the process is called gas-side fouling. It may be encountered in all of the energy intensive industries such as the food, textile, pulp and paper, chemical petroleum, primary metals, cement, and glass industries. Contamination of a condenser in refrigeration and air-conditioning systems by airborne dust and debris is also a good example of air-side fouling. It causes a decrease in the heat exchanger capacity, and eventually a decrease in system efficiency. More recently, gas-side fouling has received considerable attention owing to increased interest in heat recovery from exhaust gas streams, which was stimulated in large part by increasing fuel costs in the early 1970s.

Marner (1990) conducted an extensive review of studies that involved various aspects of gas-side fouling during the time period from 1970 to 1990. The mechanisms that led to gas-side fouling were explained in detail, and analytical and experimental studies were listed. The paper presented an excellent overview of the fouling studies but focused on gas-side fouling in boilers and gas turbines at very high temperatures. In contrast, the gas-side fouling of extended surface heat exchangers, as typically used in refrigeration and air conditioning applications and which is the topic of the current study, has received very little attention so far. Only a few studies are found in the literature. These studies are summarized here.

Cowell and Cross (1981) investigated the effects of gas-side fouling on pressure drop and heat transfer characteristics in twenty-two automobile and industrial engine radiators. The authors concluded that increase in pressure drop was substantially greater than the reduction in heat transfer and that the fouling by dust was confined almost totally to the front face of the heat exchanger core. The authors also state that the hydraulic diameter of the basic orifice in the front face of the radiator determines the effect of fouling and high performance louvered fins are the best for resistance to fouling by dust.

Bott and Bemrose (1983) carried out a systematic study of air-side fouling in finned tube bundles using fin densities of 354-433 fins/m, fin heights of 12.7−15.9 mm, a fin array of 1, 2, 3, and 4 rows of a staggered, equilateral layout, and air velocities of 1.85-5.99 m/s. The authors concluded that the air-side fouling had a pronounced effect on the air-side pressure drop (f_{foul}=1.4-2.5 f_{clean}) and that heat transfer performance, as measured in the form of the j-factor, decreased only slightly with time. They also reported that the first and last rows of the heat exchanger coils fouled more heavily than the middle rows.

Zhang et al. (1990) tested particle fouling of a diesel air charge cooler. The test parameters included foulant concentration, particle size, and temperature gradient. The authors found that there were two parts that contributed to fouling; a delay in which fouling was not apparent, and a rapid exponential fouling process. High velocities and small particles were found to accelerate fouling. The fouling layer was generally soft and easy to remove.

There are a few studies that addressed the effects of condenser fouling on overall air conditioning system performance. Bultman et al. (1993)

reported a 7.6% decrease in system COP for a 40% reduction in condenser air flow for an air conditioner. Breuker and Braun (1998) showed a 5% loss in capacity and a 8% loss in COP when about 25% of the condenser coil was blocked due to fouling. Despite the longstanding problems of gas-side fouling of extended surface heat exchangers, there have been remarkably little systematic studies of the phenomenon. In summary, quite apart from considerations of improved design techniques, more research should be done in determining the effects of gas-side fouling on hydrodynamic and thermal performances of heat exchangers such as condensers and evaporators having complicated extended surfaces.

The current study aims to investigate the effect of particle contamination and subsequent cleaning on the performance of various condensers used in unitary air-conditioning systems. The importance of this research lies in its ability to promote further understanding of the fouling process, the effects of fouling on different coil types, and to provide better information for heat exchanger field service procedures.

2. Experimental Set-up and Data Reduction

2.1 Test facility and procedure

Figure 1 shows a schematic diagram of the test facility which consists of an open and rectangular air duct, a closed water loop, a dust injecting system, and the heating coils to be tested. The cross-sectional dimensions of the air duct are 48.82 cm tall by 90.17 cm wide. A variable speed blower draws room air into the preheating section of the air duct which was connected to the test section inlet. Four 5 kW electric heaters were located directly downstream of the blower. Currently, the air duct is capable of delivering the coil face velocity from 0.2 to 2.0 m/s approximately and the dry bulb temperature extending from the room temperature to 55℃. In order to measure the static pressure drop across the test coil, a total of eight pressure tap holes with the diameter of 0.159 mm were mounted circumferentially (four holes upstream and four holes downstream of the test coil, respectively).

Temperature grids with nine K-type thermocouples were mounted upstream and downstream of the test coil, respectively. Air stream mixers were installed in front of each temperature grid to get uniform air temperature distributions. Immediately after the test section a final filter was inserted into the air duct to collect the dust passed through the test coil. Two relative humidity meters were also installed inside the air duct upstream and downstream of the test coil in order to measure the relative humidity of the air stream.

The nozzle apparatus constructed according to ASHRAE Standard 33-1978, was located downstream of the test coil and allowed the measurement of the volumetric air flowrate. The discharge of the nozzle apparatus was connected to the outdoors to vent the air. This test loop was sealed with duct tape so that air leakages at places that would influence the capacity measurements did not exceed 1.0% of the test air flowrate.

The test coil itself was directly connected to the closed water flow loop. The main components of the water loop include a flexible impeller pump, a mass flow meter, a surge tank and an electric

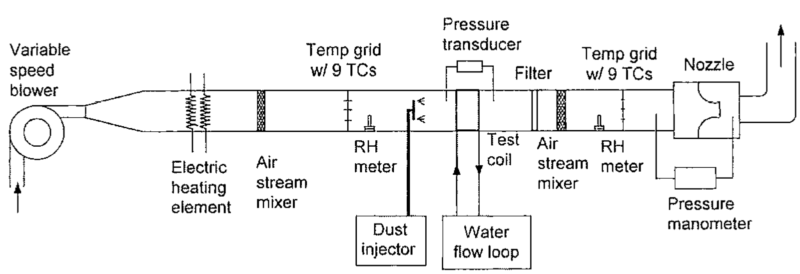


Fig. 1 Schematic diagram of the experimental facilities

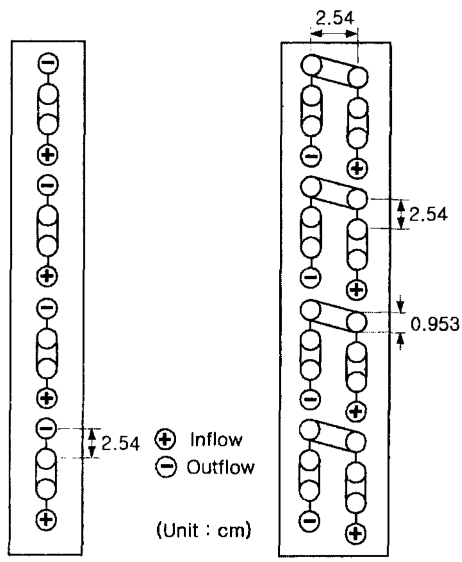
Table 1 Surface description of the test coils

Coil No.	Fin type	No. of row	Coil surface description
HX01	plate	1	22 fins/in, plain fins
HX02	plate	1	22 fins/in, louvered fins
HX03	spine	1	spine fins (standard)
HX04	plate	2	22 fins/in, one-by-one, louvered fins
HX05	plate	2	22 fins/in, continuous, louvered fins
HX06	spine	2	(woven) spine fins

heating system. The variable speed controller connected to a motor was used to adjust the flow rate. The surge tank was also inserted between the pump and the flow meter to minimize possible pressure fluctuations caused by the pump. Galvanized steel tubes of 1.27 cm diameter were used to connect the components of the water loop. Hot water flowed inside the test coil tubing to supply heat to the air flowing in the wind tunnel. Two thermistors, calibrated to an accuracy of 0.07℃ were used to measure the inlet and outlet temperatures of the water. The mass flow rate of water was measured using a mass flow meter with an accuracy of $\pm 0.044\%$ of the indicated value.

A dust injecting system was constructed to spray the dust into the wind tunnel and onto the frontal surface of test coils. It consists of a dust injector and spray nozzles mounted inside the air duct. The commercially available dust injector (*LMS Technologies, INC., U.S.A.*) was used in the experiments. Since the current air duct is approximately twice as wide as it is tall, symmetri cally bifurcated spray nozzles and two perforated circular mixing plates, having a 40% open area, were used to ensure good mixing in accordance with ASHRAE Standard 52.1-1992. Commercially available synthetic test dust prepared to ASHRAE specifications (72% standardized air cleaner test dust, 25% powered carbon and 5% No. 7 cotton linters) was used as foulant.

A total of six different condenser coils as shown in Table 1 were used in the experiments to investigate the air-side fouling effect of the heat exchangers. The first three heat exchangers are single row condensers with 16 tubes (i.d.=0.953cm) in a single row, while the second three heat exchangers are double row condensers with 16 tubes (i.d.=0.953 cm) in each row. Both single row heat exchangers and double row ones were circuited for water operation according to the suggestion by the sponsor.



(a) Single row coil (b) Double row coil

Fig. 2 Tube circuits of the heat exchangers

The exchangers are circuited for both a single row and double row heat exchangers as shown in. Fig. 2. All the tubes are connected to either the inlet or outlet manifold. The inlet and outlet manifolds are made by directly soldering the 3/8 in. tubes from the heat exchangers to a 3/4 in. copper tube. A pair of flanges at both sides for each test coil was also added to mount them easily to the air duct. Figure 3 shows a photograph of HX01 (single row with plain plate fins) including the inlet and outlet manifolds and the mounting frame.

The tests for all the heat exchangers were

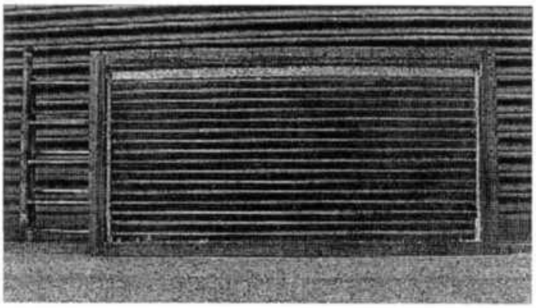


Fig. 3 Photograph of heat exchanger (HX 01) after circuiting tubes and installing frame

performed at three different conditions: 1) at clean condition, 2) at fouled condition after dust loading for 3 hours with a dust rate of 100 grams per hour, and 3) at cleaned condition after cleaning the coil with a chemical cleaner.

A first test coil was conducted at clean condition to begin with, which was followed by the dust loading, and then was tested at fouled conditions. At this point the coil was taken out of the ters, pressure transducers, and mass flow meter were collected by a HP data acquisition system and then sent the signals to a personal computer using the HP VEE data acquisition software. In addition, the net amount of the deposited dust onto the condenser coil was measured by a mass balance.

As aforementioned, a total of 300g ASHRAE standard dust was sprayed into the air stream during a three-hour interval. Some amount of the dust was deposited onto the heat coil surface while the remaining portion of the passes through the test coil and was collected by the final (downstream) filter. Therefore, the relative amount of dust deposited on the condensers in percentage during the loading can determined by the following equation:

$$W_r = 100^* [W_t - W_f/\xi_f] / W_t (\%) \qquad (1)$$

During the tests, the air-side pressure drops of each heat exchanger were measured directly by differential pressure transducer. The air-side preswhere ΔT_m indicates the log mean temperature difference.

The term $1/(UA)_{ava}$ is the averaged overall thermal resistance of a heat exchanger and is expressed by

$$1/(UA)_{avg}=1/(\eta hA)_a+(R_f/A)_a+R_m+1/(hA)_w \quad (6)$$

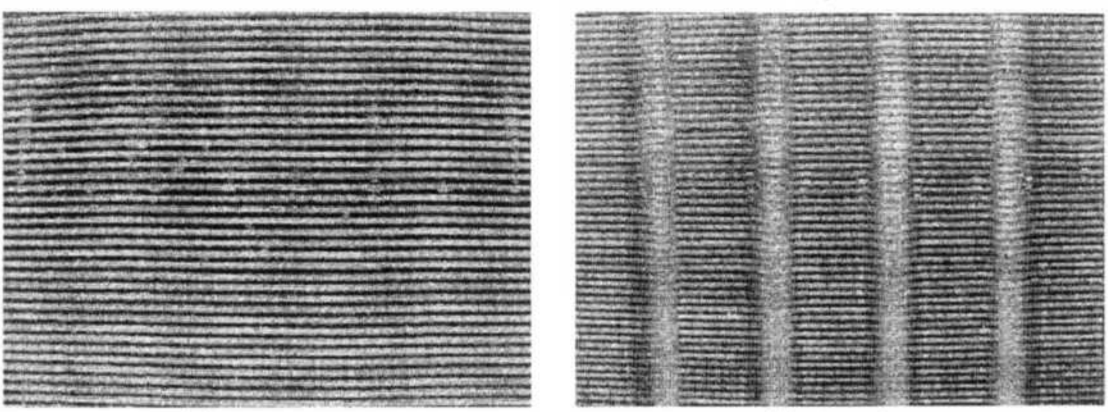
Each term in right-hand side in the equation (6) indicates the air-side convective resistance, the fouling resistance, the conduction resistance and the water-side convective resistance, respectively. The last two terms, i.e., R_m and $1/(hA)_w$, in the right-hand side of equation (6), can be assumed as constant since water flow rate is kept constant during experiments. However, the remaining two terms in the right-hand side cannot be calculated separately unless the thickness of dust deposit on the heat exchanger is uniform. The combined effects of these two terms due to fouling and cleaning will be expressed as the variation of $1/(UA)_{avg}$ value eventually. In this study, the thermal performance of a heat exchanger due to fouling and cleaning will be expressed in terms of the variation of $1/(UA)_{ava}$ value.

3. Results and Discussion

3.1 Fouling behavior and dust of the heat exchangers

After completion of the dust-loading period for each heat exchanger, photographs were taken toobserve the build up of dust on the heat exchanger surface. The uniformity of the deposited dust on the heat exchanger surface was also checked qualitatively by visual inspection. The result shows that the sprayed dust was accumulated almost uniformly throughout the frontal surface of the heat exchanger, which implies that the dust injecting system worked well.

Enlarged photographic views of the contamiated heat exchanger with plain plate fins (HX01) after dust loading are presented in Figures 4(a) and (b). These figures show partial portions of the frontal view and the rear view of the corresponding heat exchanger. By comparing the frontal views of the heat exchangers with the rear views (even for the other heat exchangers not shown here), it could be seen that the fouling was much more severe on the frontal face are of each heat exchanger. The same or similar trends were also reported in previous investigations in the literature (Cowell and Cross, 1981 ; Bott and Bemrose, 1983). The reason for the restriction of the dust fouling to the frontal face area has not been investigated in great depth. However, it is believed that the basic reason for this behavior is that for the heat exchangers tested, the air flow essentially passes straight through the heat exchanger core. Thus, the frontal surface picks up the fouling particles that impinge directly upon it, and effectively screens the inside of the core. Fin features such



(a) Frontal view (b) Rear view

Fig. 4 Photographs of the dust build-up of the single row condenser coil with plain plate fins (HX01) as dimples, louvers, and spines, which stand normal to the surface, and which therefore, may not initially be screened by the frontal surface, very rapidly become so as the face area begins to foul.

It is interesting to note that for the plate fin condenser coil, the dust was more accumulated at the leading edges of the fins as seen in the frontal view of Fig. 4(a) while this was not observed for the spine fin condenser coils. For the plate fin coil, these bridging shapes of the fouling particles act as turbulators that increase the turbulence level of air entering the condenser coil, which results in an increase in pressure drop as well as an increase in heat transfer.

3.2 Dust weight deposited onto the heat exchangers

Figure 5 presents the relative amount of dust deposited onto the heat exchangers. In this figure, the y-axis indicates the ratio of the net deposited amount of dust to the supplied amount of dust (=300g) in percentage. The results show that the amount of dust deposited onto the condenser coils strongly depends on the number of rows as well as the fin configuration. In particular, the dependence of the deposited amount of dust with fin geometry was more significant for the single-row condenser coils than for the double-row coils. The amount of dust deposited onto the single-row condenser coils increased significantly as the fin geometry gets more complex. The amount of deposited dust also increased with increasing the number of rows as expected. However, the variation of dust deposition with fin geometry was not so significant for the double-row condenser coils when compared to the variation for the single-row condenser coils. As shown in this figure, approximately 30-50% of the dust was accumulated on the single-row condenser coils, while about 60-67% of the dust was accumulated on the double-row condenser coils.

It should be mentioned that the total heat transfer surfaces of HX04 and HX05 are almost twice larger than that of HX02. Intuition may suggest that the amount of dust deposited onto the double-row heat exchangers should also be twice larger than that of the single-row heat exchanger. However, the results show that the deposited amounts of dust onto the double-row condenser coils are only 1.38 to 1.53 times larger than the one deposited on the single-row heat coil. A similar result is obtained by comparing the amount of dust deposited on HX03 (single-row spine fin) and HX06 (double row spine fin).

These results confirm the conclusions that the dust deposition is more significant on the frontal surface/first row fins and tubes than on the second row fins and tubes. Such a different amount of deposited dust depending on the fin geometry and number of rows will cause different effects of fouling on pressure drop and heat transfer characteristics for each condenser coil.

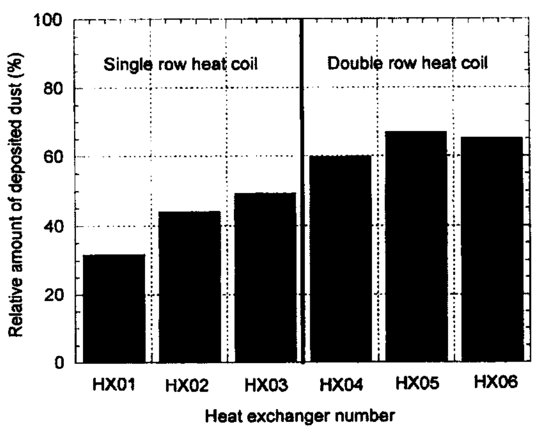


Fig. 5 Relative amount of dust deposited onto the heat exchangers

3.3 Pressure drop and heat transfer characteristics

The air-side pressure drops across the single-row plate fin heat exchanger without enhancement (HX01) at three different conditions as a function of air face velocity are shown in Fig. 6. Experimental data were collected within a range of air face velocities of 0.3 to 2.0 m/s. The open circle and the closed circle symbols indicate pressure drops at clean (as received) and fouled condition, respectively, while the cross symbol presents pressure drops at after-cleaning condition. As shown in the figure, the pressure drop of the heat exchanger for all three conditions increased exponentially with increasing air-side face velocity as expected.

After fouling of the heat exchanger, the pressure drop across the heat exchanger increased more significantly with air face velocity than the clean heat exchanger. However, after the dust was removed from the heat exchanger by applying the described cleaning procedure, the pressure drop could almost be restored to the level of the clean heat exchanger. At the standard condition of an air face velocity of 1.53 m/s (300 ft/min), the pressure drop increased by approximately 25% due to fouling compared to the clean (as received) coil. After cleaning, the pressure drop decreased to almost the original value and was only about 4% larger than at clean condition.

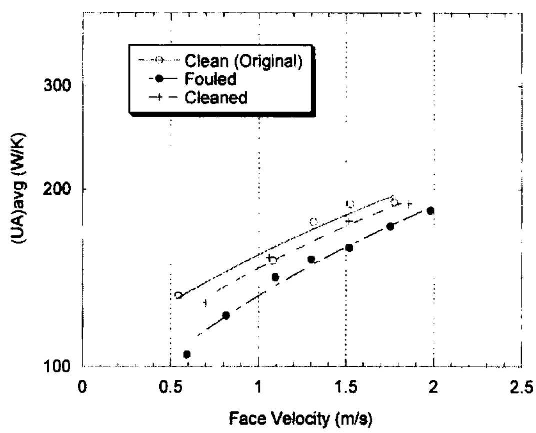


Fig. 6 Pressure drop versus face velocity of HX01 (single-row, plain plate fins)

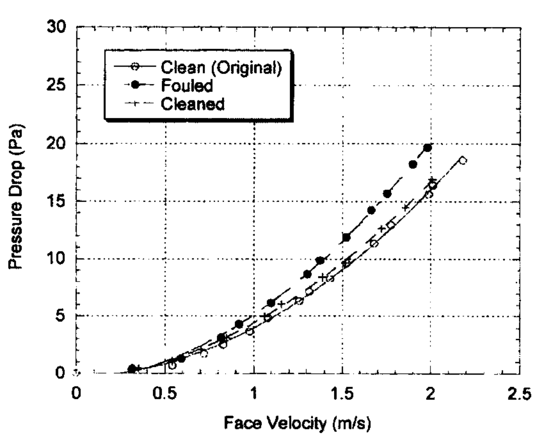


Fig. 7 $(UA)_{avg}$ versus face velocity of HX01 (single-row, plain plate fins)

Figure 7 presents the variation of the thermal performance of HX01 due to fouling and cleaning. The $(UA)_{avg}$ value in (W/K) shown on the y-axis indicates the average overall heat transfer conductance defined by equation (6). The results of $(UA)_{ava}$ were plotted in semi-log form as a function of air face velocity.

The results show that the $(UA)_{avg}$ values of HX01 increase exponentially with increasing air face velocity for all three conditions. After dust loading, the heat transfer capacity of HX01 decreases due to the air-side fouling. However, the heat transfer capacity was recovered after the coil was cleaned. The results also show that the dependence of the $(UA)_{avg}$ values on the air face velocity are almost the same regardless of the condition of the heat exchanger. Similar trend to the HX02 and HX03 (not shown here) was observed as the case of HX01 in that the pressure drop increased and the heat transfer decreased due to fouling, but both, pressure drop and heat transfer capacity, were recovered after cleaning. However, the three single-row heat coils show different degrees of variation in pressure drop and heat transfer characteristics due to fouling and cleaning. The effects of fouling and cleaning on the performances of the single-row heat exchangers are given in more detail in Fig. 8. This figure shows the quantitative comparison of air-side pressure drop and the $(UA)_{avg}$ value between the three coils at the standard air face velocity of 1.53 m/s. Calculations for the variations shown in Fig. 7 were based on curve-fitting the experimental values at each condition.

The results demonstrate that the predominant effect of fouling is to cause a more significant increase in air-side pressure drop as compared to a degradation in heat transfer. The increase in pressure drop due to fouling ranges from 28% to 31%, while the reduction in heat transfer varies from 7% to 12% depending on fin shape of the heat exchanger.

By considering the effect of the fin shapes of the different coils on the variation of the pressure drop due to fouling, it can be noted that the pressure drop increased slightly as the fin geometry is getting more complex. This can be partially attributed to the larger amount of dust deposited onto the heat exchanger with more complex fin geometries (see Fig. 5). In contrast, the reduction in heat transfer due to fouling decreases as the fin geometry is getting more complex. As mentioned earlier, the deposited amount of dust and its build-up behavior on the heat exchanger depends heavily on the fin shape. Especially, the build-up behavior of dust on the spine fin heat exchanger is different from that of the plate fin heat exchangers. For the plate fin heat exchangers, the dust accumulates on the leading edges of the fins, while the spine fin heat exchanger collects dust, "all over" the frontal surface. This may possible cause the unexpected results. A more detailed study of this phenomenon may be required to completely understand the reasons for these results.

Figure 8 also shows that the degraded performance of a heat exchanger due to fouling was almost completely recovered when the contaminated coil was cleaned according to the given cleaning procedure. After cleaning, pressure drop was 1 to 7% larger and heat transfer performance was 1 to 5% lower than that of the original heat exchanger. In addition, as the fin geometry is getting more complex, the recoveries in pressure drop and heat transfer reduction are getting better.

The effects of fouling and subsequent cleaning on the performances of the double-row heat exchanger, HX04, are given to Fig. 9 and Fig. 10, representatively. Although the results for HX05 and HX06 are not shown here, similar to the trends observed with the single-row heat exchangers. The pressure drop increased and the heat transfer performance decreased after fouling and using the specified cleaning procedure recovered the original performance after cleaning, in all cases. However, some differences can be noted. The increase in pressure drop after fouling of the double-row heat exchangers is as significant as it was for the single row heat exchangers, but the decrease in heat transfer performance was less

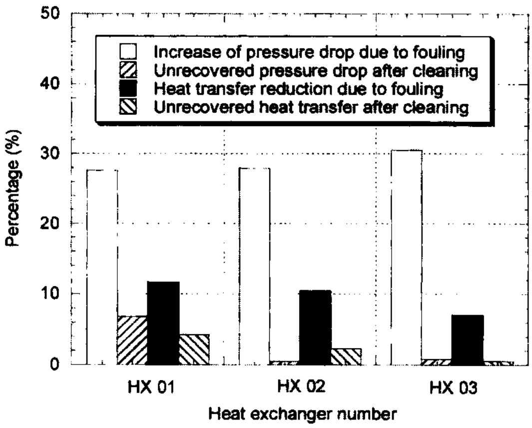


Fig. 9 Pressure drop versus face velocity of HX04 (double-row, one-by-one louvered plate fins)

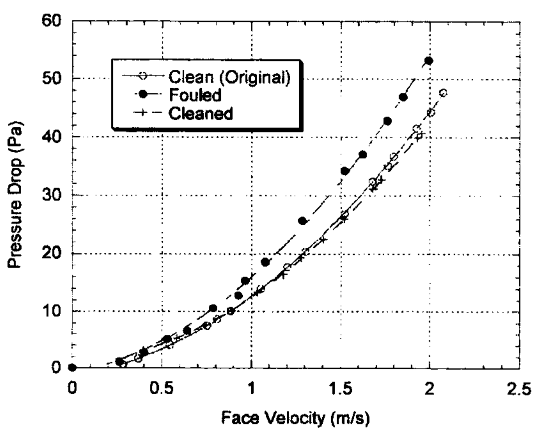


Fig. 8 Variations of pressure drop and heat transfer of single-row heat exchangers due to fouling and cleaning (air face velocity=1.53 m/s)

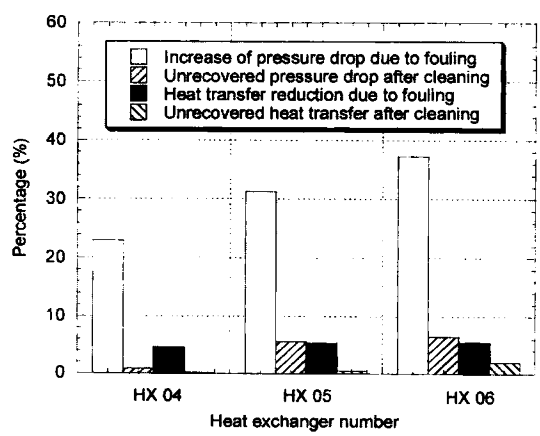


Fig. 10 $(UA)_{avg}$ versus face velocity of HX04 (double-row, one-by-one louvered plate fins)

than what was measured for the single-row heat exchangers.

This can be explained by the fact that the fouling occurs on the leading edges of the heat exchanger and thus, the, "blockage" of air flow through the coils is similar for single and double-row heat exchangers, which results in similar trends for the pressure drop. However, if the fouling is mostly occurring on the frontal surface then the double-row heating coils provide a greater depth that is still effective for heat transfer as compared to a single row coil, which results in smaller reduction of heat transfer performance for the double-row coils as compared to the single row coils. Once the loaded dust was removed from the heat exchangers, both pressure drop and heat transfer performance could be recovered to within 1% of the clean-as-received performance.

Figure 11 shows a comparison of the effects of fouling and cleaning on the pressure drop and heat transfer performance for the double-row heat exchangers at the standard air face velocity of 1.53 m/s. It can be seen in the figure that the pressure drop increases from 23% to 38% whereas heat transfer reduction ranged from 5% to 6%. Again, it has to be noted that the increase in pressure drop due to fouling is significant and in the same range as it was for the single-row heat exchangers, while the reduction in heat transfer

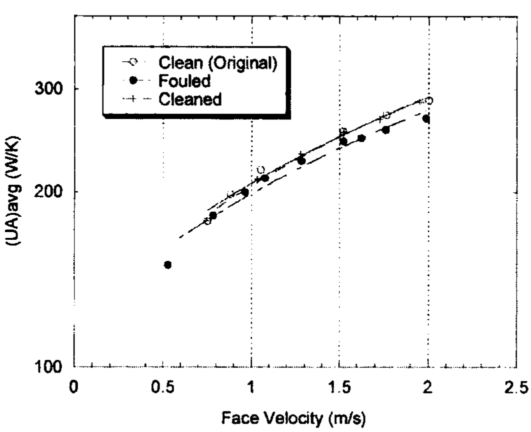


Fig. 11 Variations of pressure drop and heat transfer of double-row heat exchangers due to fouling and cleaning (air face velocity=1.53 m/s)

performance is small and much less than that of the single-row heat exchangers. Therefore, it can be concluded that the double-row heat exchangers are more resistant to heat transfer reduction than single-row heat exchangers.

It can also be seen from Fig. 11 that the pressure drop of the spine fin heat exchanger increased more (38%) than for the other two heat exchangers (23%, 31%) due to fouling. Surprisingly, the increase in pressure drop of the continuous plate fin heat exchanger was slightly larger than that of the one-by-one plate fin heat exchanger after fouling, for reason that are not clear and may warrant additional testing to verify these results.

After applying the specified cleaning procedure, the original heat exchangers performance could be recovered to within 1 to 6% just as it was the case for the single-row heat exchangers.

4. Summary and Conclusions

The purpose of the present study was to investigate the impacts of fouling and cleaning on the performances of condenser coils as used in unitary air-conditioning systems. A total of six condenser coils having different row number and different fin geometries were tested. Each condenser coil was tested at three different conditions : clean-as-received condition, after-fouling condition, and after-cleaning condition.

The main findings of the present study can be summarized as follows. In all cases, it was observed that the fouling was mostly confined to the frontal face of the heat exchanger as reported in the previous investigations. For the single-row heat exchangers, 30 to 50% of the dust fed into the wind tunnel was deposited onto the coils, while for the double-row heat exchangers 60 to 67% of the dust fed into the wind tunnel was accumulated on the test coils. The amount of deposited dust was more dependent on fin geometry for the single-row heat exchangers than for the double-row heat exchangers.

The predominant effect of fouling was to cause a more significant increase in air-side pressure drop than a degradation in heat transfer performance. For the single-row heat exchangers, the pressure drop increased by 28 to 31%, while the heat transfer performance decreased by 7 to 12% depending on fin shape at the standard air face velocity of 1.53 m/s. For the double-row heat exchangers, the pressure drop increased by 22 to 37%, and heat transfer performance decreased by only 4-5% at the same air face velocity.

Once the contaminated coils were cleaned according to the given cleaning procedure the original performance of the heat exchangers could almost be recovered completely. The pressure drop could be restored to within 1 to 7% and the heat transfer performance could be recovered to within 1 to 5% of the originally clean heat exchangers. Therefore, it is concluded that a periodic application of the specified cleaning technique will be effective in maintaining the thermal performance of the condenser coils.

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