Fin Tube Performance

E.C. Smith and A.Y. Gunter
Hudson Products Corporation
Houston, Texas

And

S.P. Victory, Jr.
Rice University
Houston, Texas

Single-tube heat transfer and mechanical strain gauge test data are presented for bimetal extruded and footed tension would interference fit fin tubes. These tubes were also tested on a semiplant-scale apparatus at steady-state and cycling conditions with air and water.

Reprinted from *Chemical Engineering Progress*

July 1966

The overall performance of air-cooled heat exchangers depends primarily on the effectiveness of two basic elements, the fin tube and the air moving equipment. This article is concerned with only one of these elements, namely the fin tube. The majority of commercial fin tubes used in modern process plants have helical, smooth surface fins varying from 8 to 11 fins/in. and 1/2 to 5/8 in. in height, Figure 1. The fin tube liners are normally 1-in. O.D. tubes of various metallic composition. Airside film coefficients for the various fin tubes are essentially identical; therefore, for the various tubes there is a variation only in the fin thickness, fin-to-tube thermal conductance, and resistance to thermal shock or atmospheric corrosion. The authors' company manufactures or uses all types of interference fit fin tubes presently available and has exerted considerable effort in research over the past three years, directed toward the determination of the optimum temperature limits and thermal derating required in using the various types of fin tubing.

Increased Interest

A review of the literature in the heat transfer field reveals an increasing interest, since 1947, in the initial contact pressures and thermal conductance, at the bimetal interface of plates and duplex tubes (1-22). The effect of thermal shock and cycling on contact pressure and thermal conductance has been studied to a lesser degree (7, 9, 11) and this information for extruded fin tubes is available for a maximum tube length of 5 ft. The authors are not aware of any data available for the effects of thermal shock and cycling on footed tension wound fin tubes.

An indication of the contact conductance between aluminum and carbon steel plates at various pressures and temperatures is given by Wheeler (9). Wheeler's results are for plate surfaces tested in a vacuum, having 70 to 100 it in. roughness with a variation of interface contact pressure from 0 to 1,000 lb./sq. in. The authors' extrapolated Wheeler's curve to 3,000 lb./sq. in., since this is well within the elastic limits of the two materials. It then shows that a reduction in contact pressure of 3,000 lb./sq. in. increases the contact thermal resistance 0.00035 hr.-sq. ft.-°F/Btu.

Gardner and Carnavos (12) and Gardner (20) present a theoretical approach to the stress problem for interference fit bimetal fin tubes which they used in analyzing the data obtained from multi-tube heat transfer tests. In their theoretical approach to this stress problem the following assumptions were made:

1. The dimension, b, representing the fin base contact length, is equal to the fin pitch.

2. Originally Gardner and Carnavos (12) assumed that a yield stress of 5,000 to 6,000 lb./sq. in. for dead soft aluminum should be allowable and P_{co} could be as high as 3,500 lb./sq. in. Higher stress values for as fabricated, due to work hardening, were recommended by Gardner (20) to be adjusted according to his Figure 3.

3. The slender fin is assumed not to deform laterally under these stresses.

The recommendations made by these authors and Young and Briggs (21) for the isothermal tube wall temperature at which the contact pressure, P_{co}, is exhausted, is shown in Table 1.

Experimental program

The objectives of the present investigation were to:

1. Determine joint contact pressures and T* as manufactured by both mechanical strain gauge and heat transfer tests.
2. Check fin column stability visually, photographically, and by strain gauges.
3. Check effects of variation in intensity of thermal shock and cycling.

The experimental program was then outlined as follows:

1. Select random samples of extruded and tension wound fin tubes (24-ft. long as manufactured). Samples of other manufacturers' tension wound fin tubes were already available. Aluminum, type 6063-0 for extruded and type 1100-0 for footed fins, were used on all samples. The tube liners used were type A-179 carbon steel with 14 BWG average wall, type 304-LC stainless steel with 16 BWG average wall, and type 6063H6 aluminum alloy with 16 BWG average wall.

2. Single-tube tests were run first on all samples at 220°F tube wall temperature for over-all heat transfer rate, U_o.

3. Then 11-1/4-in. samples were cut 2 ft. from the end of the tube for strain gauge tests. Extreme care was taken on tension wound fin tubes to preserve fin tension.

4. The remainder of the sample tube (20 ft.) was then retested on the single-tube tester and either run through the semiplant-scale cycling apparatus, or used for single-tube temperature range tests.

5. After cycling, single-tube tests were repeated. The inside of the tubes were sand blasted and cleaned with chlorothene prior to retesting.

The accuracy of testing is listed in Table 2.

Single-Tube Tester

The single-tube tester is shown in Figure 2. This simple rugged test unit was developed some 14 years ago for manufacturing control purposes and is used in this investigation only as a comparator based on over-all heat transfer rates, U_o.

Briefly, dry saturated steam at 15 lb./sq. in. is measured through a 2-in. by 1-in. venturi steam meter [1] thence through a 3-in. by 1-in. water cooled desuperheater [2] (this is used only for steam pressures under 15 lb./sq. in.). Steam outlet valve [7] is used to control steam quantity. Steam temperatures and pressures are measured at the inlet by a P.I. [4], a mercury manometer [5], and a stem-type thermometer [15], and at the tube outlet by a manometer [6] and a thermometer [14]. Steam pressures below atmospheric, when required, are maintained by an air jet ejector [8] capable of ejecting 500 lb./hr. of steam and noncondensables. Air is supplied by a blower [9] to a movable air duct [10]. Air is measured by a Taylor Biram-type vane anemometer [11]. The 4-in. by 12-in. air duct encloses a 12-in. test section of the fin tube. The duct is split and flanged at [17] to permit insertion of the test fin tube. For any series of comparative tests the air duct was located at the same distance from the tube steam inlet, usually 10 ft.

A test consists of inserting a test fin tube into the headers and the air duct, sealing the air duct joints around the flanges, and then turning on the air and the steam. Adjust the steam flow and the pressure to the comparative test conditions required. Allow 20-min. to reach equilibrium. Take 4 sets of readings, 2-min. apart, and calculate the average conditions, MTD and duty by air side, then, the over-all rate, U_o, based on outside bare tube surface.

For the tests before and after thermal cycling the tests were made with dry saturated steam at 15 lb./sq. in. at three points on the test tube, namely 5, 10, and 15 ft. from the tube inlet and then averaged.

The single-tube temperature range tests were run at 7 ft. from tube inlet at steam pressures from 1.5 to 29.7 lb./sq. in. abs. Steam mass velocities were adjusted to give the same over-all rate, U_o, by using an all aluminum extruded fin and tube liner. Then when bimetal fin tubes were tested, the lowest wall temperature (140°F) was considered the base $1/U_o$. This base $1/U_o$, would be subtracted from the $1/U_o$'s of succeedingly higher tube wall temperature tests on the same test tube and is shown as increased bond resistance.

Strain Gauge Test Apparatus

A technique for mounting strain gauges on the I.D. of the fin tubes was developed utilizing an air-actuated mandrel (patent application under consideration). This device (Figure 3) is similar in purpose to the well-known balloon technique that has been used in the past. It is felt, however, that the device mentioned herein is easier to use and yields more consistent results than the balloon technique. The mounting device is essentially a split rod that has an air connection on one end and allows air passage to a piston, which is perpendicular to the axis of the rod at the other end. A rubber-faced mandrel is attached to the piston. The rod itself is semicircular having a radius the same as the inside radius of the tube; the mandrel is a circular segment having a radius equal to the inside radius of the tube minus the composite gauge thickness.

Bonded electric resistance strain gauges manufactured by MicroMeasurements, Inc., were used throughout this investigation. Essentially two different gauge configurations were employed: (1) single element gauges (MA-XX-090-DG-120) and (2) 90°-rosettes (MA-XX-125-TA-120). All gauges were of the self-temperature-compensated type so that the thermal expansion coefficient of the particular material on which the gauge was to be mounted would be matched. For instance, if gauges were procured for mounting on a steel liner, then a compensation code number of 6 ppm/°F was chosen so that a minimum value of apparent strain due to temperature would be indicated.

Since the gauges were to be mounted in areas which prohibited soldering of the leads after mounting, it was necessary to install jumper wires on the gauges prior to mounting (Figure 4). Number 34 solid copper, single conductor wire with soldereze insulation was used for this purpose. All jumper wires were cut to a length of 6 in. and a 1/32-in. tinned area was provided at each end. The tape masking system for precise lead wire attachment to gauges, suggested by Bean (19), was used and the leads were attached with 0.020-in. diameter 300°F rosin core solder. An American Beauty No. B-2000-L soldering iron with the tip temperature controlled by a power-stat was ideal for this purpose. Careful control of soldering iron temperature and soldering technique was necessary in order to maintain consistency in lead wire attachment and to avoid damaging the unmounted gauge. The jumpers were taped together with a small piece of masking tape, providing a convenient means of handling the gauge without causing excessive strain on the foil tabs at the solder joint. The tape mask was removed from the gauges with rosin solvent. Precut pieces of masking tape were secured to the top side (grid side) of the strain gauge to fill the gap between the lead wires. In other words, the gauge thickness was built up to the same thickness as the lead wires over the total gauge surface with masking tape. A final piece, the size of the gauge, was placed over the entire gauge surface. The reason for doing this was twofold: (1) to distribute the pressure, exerted on the gauge in mounting, as evenly as possible over the entire gauge surface, and (2) to increase the stiffness of the gauge, thus facilitating handling.

Gauge Mounting

The actual gauge mounting procedure was as follows:

1. The mandrel was placed on the piston of the mounting device.

2. The pre-wired gauge was attached to the mandrel with double-sided pressure sensitive tape in the desired orientation.

3. The gauge back was cleaned with neutralizer and allowed to air dry.

4. A thin coat of Bean's BAP-1 cement was applied to the gauge backing and to the gauge area of the specimen. Both were allowed to air dry for 30 min. (A thin coat of BAP-1 adhesive had been applied previously to the gauge area of the specimen and allowed to air dry).

5. The complete gauge mounting assembly was slipped inside the tube and the gauge was lined up in the desired location.

6. A modified C-clamp was placed on the protruding end of the mounting device clamping the device to the finned tube; thus securing the mounting device to the finned tube allowing no chance of slippage of the assembly.

7. Air pressure was applied to the piston providing the required clamping pressure (20 lb./sq. in.) for the gauge. The assembly was placed in the oven at 275°F ± °5F and cured for 1 hr.

8. After curing of the adhesive was completed, the device was removed from the oven, the piston contracted by applying a vacuum to the air line, and the tape debris removed from the gauge area with rosin solvent.

A soldering terminal was attached on the inside of the test tubes near the end of the tube using BAP-1 adhesive. The purpose of the terminal was to allow a transition from the jumper wires to the lead wires and to assume that forces applied to the lead wires would not be transmitted to the gauge. Number 26 stranded copper wire with Hyrad insulation was used for the lead wires. All lead wires were cut to 6 ft. in length and stripped and tinned for 1/8 in. on the terminal end and 1 in. on the indicator end. The leads were soldered to the terminals and the entire connection was coated with Bean's Gagekote No. 3 and air dried. The gauge was also coated with Bean's Gagekote No. 3 at this time.

In addition to the various test specimens which were prepared for this investigation, two compensator tubes (a liner without fins with a gauge mounted on the inside in a similar fashion as on a finned tube) were prepared for each liner material and each gauge type used. As part of the test procedure a strain temperature curve was prepared, using one compensator tube as an active gauge and the other as a dummy gauge, over the entire range of temperature (75°F to 300°F). This was done periodically as a check on the quality of temperature compensation and on the compensator tubes.

Test Procedure

The actual test procedure consisted of placing the tubes to be tested along with their respective compensator tubes in the test oven. Precaution was taken to ensure that all lead wires were in the oven for the same length and the leads were connected to a Baldwin Type "N" strain indicator. Zero readings were taken at ambient temperature (75°F) and the oven temperature increased in increments to 300°F. Strain readings were taken at each temperature increment after thermal equilibrium was attained which took about 30 min. Readings were always taken with one compensator and then the other to provide a check. Also, each sample was cycled several times in order to provide a check on the reproducibility of the strain readings.

After testing, each tube was split (Figure 5) and the gauge orientation, and fin dimensions were checked.

Using this procedure strain gauges could be applied up to 8 in. inside the tube (Figure 5) with an accuracy of ±3° in orientation, and an over-all accuracy in the strain reading of ±10 μ in./in. The measured surface strains were then related to the contact pressure at the bimetal interface by using the elastic relationships for deformation of thick walled cylinders (see Equation 177 Appendix).

Gauges were also mounted on the fin surface. The procedure used was essentially the same as described above.

Thermal Cycling Apparatus

The thermal cycling apparatus is shown in Figure 6. Review of Young and Katz (11) article and of plant operations indicates that longer tube length and lower wall temperatures (200°F to 400°F) should be used. The intensity of thermal shock could be varied by using air and water as cycling agents. The metallurgy was selected to cover maximum and minimum thermal coefficients of expansion for the bimetal fin tubes. This would be aluminum fins with carbon steel, 316-LC stainless steel and 6063H6 aluminum tube liners.

The test equipment consists of a circuit including a 2-ft. by 4-ft. shell still [2] heated by gas; a 22 gal./min. rotary pump [3] ; a relief valve [4] ; two 20-ft. test fin tubes rolled into free headers and mounted on a rigid frame enclosed in a box with water spray and an air flow apparatus [5]; piping connecting the still pump and the fin tubes. The fin tubes are enclosed in a box so air circulation is less than 25 ft./min. to face area when water or air is not used. Thermometer and thermocouple wells [17, 18, and 19] are provided in the inlet and the outlet of the fin tubes and the still to determine temperature in the circuit.

A Minneapolis-Honeywell Electronik 17 [7] temperature recorder-controller was used on the fin tube outlet thermocouple. This controlled the maximum and the minimum tube fluid outlet temperatures by opening and closing two Asco 1/2-in. solenoid valves on the water line to 24 1/8-in. Schutte-Koerting water spray nozzles set on 10-in. centers above each fin tube or to the power for the air fan for air cycling. A steady-state temperature test consisted of setting up oil flow to maximum rate (22 gal./min.) and adjusting still firing rate to hold temperature at fin tube outlet constant at desired test temperature. This test temperature would be maintained for 48 hr. The test fin tubes would be observed for crawling of aluminum along the tube liner by the register dots on the tube liner [16] and for cracks in the aluminum sheath.

The test fin tubes were tested at the quarter and center points on the Hudson single-tube tester before and after each test to determine the change in bond resistance. The inside of the tube was sand blasted, and cleaned with chlorothene before initial and retests.

Temperatures of oil, air and water, and quantity of oil by Pottermeter Model 546P and periodic weight tank measurements were recorded each hour.

For cycling tests the same procedure was followed except when the oil inlet temperature from fin tube reached desired test level the temperature controller opens solenoid water valve to water sprays. The lower temperature on the controller is adjustable with still firing rate to give water spray time of 45 sec. and a total cycle time of 5 to 6 min. A minimum of 500 cycles/test was used. This would take 48 hr.

Air cycling was accomplished by the temperature controller turning on the fan motor [20] when desired temperature was reached. Air flow was continued for 1-1/2 min. and total cycle was approximately 5 min.

Discussion of Results

In general, results from the three different test methods show reasonable agreement. There is, however, considerable variation in contact pressure and cycling results for footed tension wound fin tubes of various manufacture. The authors have used the highest contact pressures and the lowest cycling resistances in all discussions for the tension wound and average values for the extruded fin tubes.

As mentioned in the introduction, this article is limited to extruded and tension wound fin tubes. Other manufacturers' fin tubes were tested but not reported; namely, B, E, and F covering embedded, knife edge, and plate fins, respectively.

Test Results for P_{co} and T*

P_{co} and T* are the original manufactured contact pressure and isothermal temperature at which this contact pressure is exhausted. These two values are the basis for all air gap, r_g, calculations.

Table 3 lists and compares data reported here with prior test work or recommendations. It will be noted for extruded fin tubes that the authors' data (T* == 198 to 215°F) check closely with Young and Briggs (21) (T* = 200°F) and is lower than that of Gardner (20) (T* = 280°F). For footed tension wound fin tubes, the data reported here T* = 145 to 150°F) are again much lower than the Gardner (20) recommendation of 260°F. The present data check T* and P_{co} by both heat transfer single-tube tests, Figure 7, and by isothermal mechanical strain gauge tests, Figure 8. The close agreement between the two test procedures (10%) and with Young and Briggs (21) is significant of accuracy.

Dimensions *b* and t_f

The isothermal mechanical strain gauge tests reported in Figure 8 permit approximate evaluation of the dimension *b* (that portion of the fin foot in direct metallic contact with the tube liner) when P_{co} is present.

Comparison of extruded and footed tension wound fin tubes on dimension *b* and variations of μ with t_f (Equation 20 Appendix) is shown in Table 4. It will be noted that, when P_{co} is present, *b* is 0.033 in. for extruded and 0.013 in. for footed tension wound fin tubes. This is determined by proportion. Figure 8 for extruded fin tubes shows that the average contact pressure, P_{co} at 80°F, is 1,100 lb./sq. in. at the bimetal interface for P = 0.125 in. P_{co}, at 80°F, using Equation 20 (Appendix) (with t_f = 0.025 in., μ = 0.213) is 4,310 lb./sq. in. Then $b = 1100 \div 4310$ X 0.125 in. = 0.033 in. Obviously, from Figure 1, this fin tapers from 0.025 in. at point B to 0.05 in. at the sheath. This would increase μ 18%, if t_f were 0.05 in. instead of 0.025 in. (see Table 4.) In turn, P_{co} would be lowered 18% and *b* increased to 0.039 in.

Test data was taken with strain gauges on both the fins and the I.D. of the tube liner to check contact pressure. These were biaxial gauges mounted opposite each other at point B, on the fin 0.14 in. above tube O.D., and the readings were averaged to eliminate any indications of bending and to reflect only strains due to radial loading. These data show at 80°F-5,860 lb./sq. in. on 0.025 in. t_f at point B-0.14 in. from the liner. This checks about 20% high, but in view of the difficulty in the orientation of strain gauges on the fins and uncertainty in dimensions *b* and t_f, it is considered satisfactory.

The preceding discussion covers dimension *b,* when fin contact pressure P_{co} exists. Obviously when P_{co} is exhausted and an air gap, r_g, is present *b* must be equal to, or approximate, the fin pitch, P, as pointed out by Gardner and Carnavos (12). There is some uncertainty due to possible eccentricity of the air gap and to lateral movement of the fin tips.

Fin Column Stability

In the theoretical approach to this problem (12, 20) the fin column is assumed to remain vertical, thus stable. It is obvious that as long as the fin remains vertical and stresses remain within the elastic limits, the fin is capable of exerting more pressure on its base than if its tip moved laterally.

Observation of the isothermal strain gauge tests on 11-1/4-in. samples from 80 to 300°F showed considerable movement of the fin tips for footed tension wound fin tubes and a slight amount for extruded. Fixed position photographs were taken, from which Figure 9 was prepared. This showed lateral movement of 0.6 fin pitch, P_1 on the fin tip for footed tension wound fin tubes, which is five times the fin column average thickness. Extruded fin tips indicate no apparent movement.

Visual observation down to O°F shows additional fin tip movement for footed tension wound fin tips and a start of lateral fin tip movement on extruded fins. Therefore, the theoretical assumption for no lateral fin tip movement is invalid. This lateral movement results in lower manufactured contact pressure.

The tip deformation described here can possibly be explained qualitatively by observing the difference, in fin cross-sections (Figure 1), between extruded and footed tension wound fins. The tension wound fin is attached to a thin cantilever base through a knuckle radius, resulting in inherent instability as compared to the centrally located tapered extruded fin with fixed base.

One additional point should be considered on the footed tension wound fin where data shows b = 0.013 in. when P_{co} is present. The cantilever fin foot is not constrained, except by shear just adjacent to the knuckle radius. It is possible for thermal differential expansion on this foot to begin before P_{co} is exhausted, thus affecting the r_g calculations at the lower values. Wheeler (9) states that there is the equivalent of 500 μ in. air gap when P_{co} is present. Since strain gauge tests indicate b is only 10 to 20% of the fin pitch, then the actual metal to metal point contact is much less than this.

Summarizing, fin column stability has a marked effect on manufactured contact pressure. Obviously, the geometry (thus instability) of the footed tension wound fin is the major factor in its low manuactured contact pressure.

Thermal Cycling Tests

Figure 10 gives test data on extruded and footed tension wound fin tubes under steady-state and cycling conditions with air and water. It should be pointed out that steady-state tests have a variation in tube wall temperatures of 10 to 15°F due to a 20 to 25°F drop in ambient air between daylight and nighttime operations.

Operation of air coolers in process plants can be cyclic in nature due to several factors, among which are:

- Cyclic process
- Forced draft with rainstorms
- Water spray on tubes for additional cooling in summer months
- Steam condensers
- Two speed fan motors
- Automatic shutters
- Auto-Variable fans

In general, the intensity of thermal shock of air cycling (76 to 116°F ΔT) was about 50% of water cycling, (116 to 240°F ΔT). The increase in bond resistance due to cycling, Figure 10, followed this percentage for both extruded and tension wound fin tubes.

The steady-state tests increased bond resistance only after original manufactured contact pressure, P_{co}, was exhausted. The increased bond resistance for steady state, r_s, approximated the air cycling results for extruded and was about 50% of air cycling for footed tension wound.

Figure 11 shows results of cycling on an extruded aluminum fin aluminum tube liner, and an aluminum fin-304-LC stainless steel tube liner. This confirms, that when coefficients of thermal expansion are equal, or nearly so, there is very little effect due to cycling. Table 5 lists coefficients of thermal expansion for various metals.

Young and Katz (11) data covered 5-ft. lengths of stripped and unstripped extruded fin tubes at isothermal tube wall temperatures of 350°F and 650°F up to 12,000 cycles. Cycle time was 8 min. at 350°F with 280°F ΔT and 2 hr. at 600°F with 450°F ΔT. This compares with 500 cycles at 5 to 6 cycle min. time and 240°F ΔT at 360°F wall temperature for the authors' tests with 20 ft. lengths stripped.

There is no real comparison between the two sets of tests. Young and Katz tests had no tube-side fluid flowing when cooling down. The data reported here have tube fluid heating during the cooling cycle so it would be comparable to most plant operations. However, Young and Katz reported 0.0009 and 0.0012 increased bond resistance on two carbon steel liners with 28 cycles at 600°F isothermal wall temperature and 450°F ΔT. The authors' data showed 0.00095 with 500 cycles at 360°F and 240°F ΔT.

Young and Katz also reported longitudinal expansion of the aluminum sheath as well as some tangential cracks. Longitudinal expansion was noted in the tests reported here. However, no tangential cracks were encountered below 360°F on air and water cycling.

Mechanism During Cycling

The mechanism that causes increased bond resistance, when cycling exists, appears to be due to alternately releasing the manufactured contact pressure, then with longitudinal movement, reclamping, thus reducing the manufactured contact pressure, P_{co} and metal-to-metal point contact.

Figure 11 is shown to give some indication of the intensity and rapidity of temperature drop and resultant compressive stress increase in an extruded bimetal fin tube when water spray is turned on. This is an approximate curve based on the mass of aluminum fin and carbon steel tube metal, with heat transfer rates present during the period, on a heat balance basis. It will be noted the ΔT, between the fin centroid and tube wall, of approximately 140°F develops in 1 sec., which could amount to 6,000 lb./sq. in. contact pressure at the fln base. Obviously, there are too many unknowns in the calculations to put in real numerical values, but it does indicate severity of the test.

Another objective of cycling with air and water was to compare extruded and footed tension wound fin tubes on protection of tube liners from water scale and water or air corrosion. Figures 12 and 13 are photographs of conditions of typical tube liners after cycling. Figure 13 represents tubes tested in this investigation. Figure 12 is a typical tube taken from four bundles retubed for a petrochemical plant in the Gulf Coast area. These were Manufacturer C footed tension wound fin tubes-ten 1/2-in. high aluminum fins per inch on 1-in. 14 BWG carbon steel tubes, which had been sprayed with water in the summer months for two years. Tube wall temperature was approximately 225°F.

Obviously, the footed tension wound fin tube does not protect the tube liner while the extruded fin tube gives good protection. At a nominal cost, the bare part of the extruded fin tube between fins and tube sheet can be covered with an aluminum sheath, or other impervious coating to provide complete protection of the tube.

The white material showing up in the two photographs is lime scale from the water. The dark material is iron oxide. Note the helical dark rings on the tube liners of the footed tension wound fin tubes. First thought would be that these rings represent the space between fin feet where water or air could attack. Actually this is partially true but the majority of this represents vertical fin pressure at this point and movement of the point under cycling conditions when P_{co} is varied.

Air Gap Resistance

For any given bimetal fin tube wall temperature there are a large number of possible air gap resistances, r_g, due to variable air and tube fluid film coefficients and temperatures. Both Gardner and Carnavos (12) and Young and Briggs (21) give equations or graphs to calculate r_g at any given condition. However, these equations do not directly compare extruded and footed tension wound fin tubes without a large number of calculations for the average engineer. Figure 14 was prepared to compare these two fin tubes at 125°F average air temperature and service rate U_o of 75 which covers a majority of the process services encountered. In addition, to give an overall comparison, increased resistance due to steady state and cycling, r_s, has been added to r_g, under curves A, B and C. The right-hand side of this figure gives percent derating curves at various service U_o's for the two fin tubes. These derating curves are accurate for U_o = 75, but read low for rates more than 75 and high for less than 75. This figure shows a dotted line where the fin tube is not recommended for the service.

The comparison at 200°F wall temperature steady state shows r_b is 0.001 for footed tension wound fin tubes and zero for extruded. The percent effective surface, at U_o = 75, is 92 and 100% respectively. This difference varies with cycling usually being higher.

It should be pointed out here that these are maximum differences since in making up these curves the tube fluid outlet wall temperature is high enough that r_g is present. The majority of services will have tube fluid outlet temperatures below the point at which P_{co} is exhausted and will represent some percentage of this maximum difference. However, for services having average tube wall temperatures, T_w 200°F and above, the percentage of maximum difference will be over 50% in all cases.

Conclusions

This work has resulted in several conclusions:

1. Isothermal mechanical strain gauge tests are an accurate means of determining T* and the average contact pressure P_c.

2. Fin column stability is one of the important limitations to the maximum values obtainable for manufactured T* and P_c.

3. The average contact pressure, P_c, as obtained directly from strain gauge data, was 1,100 lb./sq. in. for the extruded fin tubes and 250 lb./sq. in. for the footed tension wound fin tubes at 80°F manufactured temperature.

4. For extruded fin tubes, the data reported here (T* = 198 to 215°F; P_{co} = 4,310 lb./sq. in.) checks closely with Young and Briggs (21) (T* = 200°F; P_{co} = 3,700 lb./sq. in.) and is lower than Gardner (20) (T* = 280°F).

5. For footed tension wound fin tubes, the data reported here (T* = 145 to 150°F; P_{co} = 2,340 lb./sq. in.) is again much lower than Gardner's (20) recommendation (T* = 260°F).

6. The Gardner and Carnavos (12) Equation 22 (Appendix) for r_g, is satisfactory for design purposes if T* and PCO are known by test data.

7. Increased bond resistance due to thermal cycling is appreciable.

8. Tube liner protection is important.

9. Fin tip movement requires additional theoretical analysis of the stress problem with further strain gauge testing from the fin side to check the theory.

ACKNOWLEDGMENT

The authors appreciate the permission of Hudson Products Corp. to publish the data and techniques presented in this paper, Further acknowledgment is due to all the personnel of Hudson Products Corp. who worked on this project.

Notation

A_o outside bare surface of fin tube liner, sq. ft.

B base contact width of fin, in. d - outside diameter of fin tube liner, in.

D outside diameter of fin, in.

E_f modulus of elasticity of fin material, lb./sq. in.

g radial gap between fin and liner in.

k_f conductivity of fin material, Btu hr.-sq. ft.-°F-ft.

k_e conductivity of entrapped fluid, Btu/hr.-sq. ft.-°F-ft.

P fin pitch, in.

P_c contact pressure between fin base and liner, lb./sq. in.

P_{co} contact pressure as fabricated between fin base and liner, lb./sq. in.

r_1 inside radius of liner, in.

r_2 outside radius of liner, in.

r_b equal to $r_g + r_s$, in.

r_s increased resistance due to cycling, hr.-sq. ft.-°F/Btu.

r_f fin metal resistance, hr.-sq. ft.-°F/Btu.

r_g gap resistance, hr.-sq. ft.-°F/Btu.

r_{go} initial gap resistance, hr.-sq. ft.°F/Btu.

r_i inside fluid film, plus dirt, plus liner wall resistance, hr.-sq. ft.°F/Btu.

r_i' inside fluid film plus dirt resistance, hr-sq. ft.-°F/Btu.

r_o outside fluid film plus dirt resistance, br.-sq. ft.-°F/Btu.

r_w liner wall metal resistance, hr.sq. ft.-°F/Btu.

R over-all heat transfer resistance, hr.-sq. ft.-°F/Btu.

$R*$ over-all heat transfer resistance in absence of any gap resistance, hr.-sq. ft.-°F/Btu.

t liner wall thickness, in.

t_f fin base thickness, in.

ΔT maximum fin metal temperature differential between start and end of cycle, °F.

$\mathrm{T_o}$ fin and liner temperature as fabricated, °F.

T_a ambient fluid bulk temperature, °F.

T_h heating medium temperature, °F.

T_w liner wall temperature °F.

$T*$ isothermal tube liner temperature where $\mathrm{P_{co}} = 0$, °F.

$\mathrm{U_o}$ over-all heat transfer coefficient based on liner bare surface, Btu/hr.-sq. ft.-°F.

Greek Letters:

ε_t tangential unit strain, μ in./in., 1 x 10^{-6} in./in.

ε_a axial unit strain, μ in./in., 1 x 10^{-6} in./in.

μ constant defined in Equation 20, sq. in./lb.

a_f thermal expansion coefficient of fin material, $(°F)^{-1}$.

a_t thermal expansion coefficient of liner material, $(°F)^{-1}$.

v_f Poisson ratio of fin material.

v_t Poisson ratio of liner material.

ρ resistance parameter defined in Equation 23, hr.-sq. ft.-°F/Btu.

σ_t tangential stress at inside surface of liner, lb./sq. in.

LITERATURE CITED

1. Brunet, A, W., and F. F. Buckland, "Thermal Contact Resistance of Laminated and Machined Joints," *Trans. Am. Soc. Mech. Engrs.*, 71, p. 253-257 (1949).

2. Weills, N, D., and E. A. Ryder, "Thermal Resistance Measurements of Joints Formed Between Stationary Metal Surfaces *Trans Am. Soc. Mech. Engrs., 71, p.* 259-267 (1949).

3. Kouwenhoven, W. B., and J. H. Potter, "Thermal Resistance of Metal Contacts," *Welding J. (N. Y,) Research Suppl., 27,* p. 515-s–520-s (1948).

4. Cetinkale. T. N., and M, Fishenden, "Thermal Conductance of Metal Surfaces in Contact," *Proc. Gen. Disc. Heat Transfer, Inst. Mech. Engrs. (London) and Am. Soc. Mech. Engrs.*, p. 271-275 (1951).

5. Barzelay, M. E., K. N. Tong, and C. F. Holloway, "Thermal Conductance of Contacts in Aircraft Joints," NACA TN 3167 (1964); "Effect of Pressure on Thermal Conductance of Contact Joints," NACA TN 3295 (1955).

6. Boni, F., "The Design of a Liquid Metal Heated Bayonet Tube Steam Generator," *J. Am. Soc. Naval Engrs., 70*, p. 231-243 (1958).

7. Mehringer, F. J., "Design, Fabrication, and Performance of the Duplex Heat Transfer Tube," U, S. Atomic Energy Comm., Research Report KAPL-1797 (1957).

8. King, E. C., and R. C. Andrews, "The Generation of Steam From Liquid Metal at High Heat Fluxes," Am. Inst. Chem. Engrs. Heat Transfer Symposium, Preprint No. 5, Annual Meeting, St. Louis, Mo. (1953).

9. Wheeler, R. G., "Thermal Contact Conductance of Fuel Element Materials," paper presented at the Fifth Nuclear Congr., Cleveland, Ohio (April 7, 1959).

10. Young, E. H., *et al.*, "Development of an Apparatus for the Measurement of Low l3ond Resistances in Finned and Bare Duplex Tubes," Engineering Research Institute, Univ. of Michigan, Report No. 48, Project 1592 (1957).

11. Young, E. H., and M. L. Katz, "The Effect of Thermal Cycling to 350°F and 600°F on the Heat Transfer Performance of Integral Finned Tubes," *Chem. Eng. Progr. SymPosium Ser., 24* (1959); also Am. Document. Inst. No. ADI 5859.

12. Gardner, K. A., and T. C. Carnavos, "Thermal Contact Resistance in Finned Tubing," *Trans. Am. Soc. Mech. Engrs., Heat Transfer, 82,* No. 2, p. 279-293 (1960).

13. Timoshenko, S., "Strength of Materials," Part II, 3rd ed., p. 205-210, D. Van Nostrand, Inc., Princeton, N. J. (1966).

14. Fenech, H., and W. M. Rohsenow, "Prediction of Thermal Conductance of Metallic Surfaces in Contact," paper presented at Fifth Natl. Heat Transfer Conf. and Exhibit, Houston, Tex., Paper No. 62-Ht-32, (August 5-8, 1962).

15. Moon, J. S., and R. N. Keeler, "A Theoretical Consideration of Directional Effects in Heat Flow at the Interface of Dissimilar Metals," paper presented at the Fifth Natl. Heat Transfer Conf. and Exhibit, Houston, Tex.. A.I.Ch.E. Preprint 16 (August 5-8, 1962).

16. Henry, J. J., and H. Fenech, "The Use of Analog Computers for Determining Surface Parameters Required for Prediction Of Thermal Contact Conductance," paper presented at Am. Sec. Mech. Engrs. Winter Annual Meeting, Philadelphia, Penn., Paper No. 63-WA-150 (November 17-22,1963).

17. Stubstad, W. R., "Measurements of Thermal Conductance in Vacuum," paper presented at Am. Soc. Mech, Engrs. Winter Annual Meeting, Philadelphia, Penn., Paper No. 63-WA-150 (November 17-22, 1963).

18. Petri, F. J., "An Experimental Investigation of Thermal-Contact Resistance. in a Vacuum," paper presented at . So, Mech. Engrs. Winter Annual Meeting, Philadelphia, Penn., Paper No. 63-WA-156 (November 17-22, 1963).

19. Bean, W. T., "Strain Gage Accessories and Technique," Detroit, Mich. (1963).

20. Gardner, K. A., "Rational Design Temperature Limits for Interference Pit Finned Tubing," Fifth Natl. Heat Transfer Conf., Cleveland, Ohio (August 10, 1964).

21. Young, E. H., and D. E. Briggs, "The Bond Resistance of Integral Finned Tubes with Steel Liners," 55th Natl. Meeting, Am. Inst. Chem. Engrs., Houston, Tex. (February 11, 1965).

22. Shlykov, Yu.P., and Ye.A. Ganin, "Thermal Resistance of Metallic Contacts," *Intl. J. Heat Mass Transfer, 7,* p. 921-929 (1964).

Smith Gunter Victory

Ennis C. Smith attended Rice Institute. He has worked with Imperial Sugar Co., City of Houston Water Div., and Dow Chemical Co. Since 1948 he has been with Hudson Products Corp., where he is vice president and general manager.

Addison Y. Gunter received his B.S.C.E. degree from Texas A&M University. He is the author of three papers and twenty-one patents on heat transfer. He is assistant to the vice president, Hudson Products Corp., directing applied research and development on cooling equipment.

Sydney P. Victory received his B.S.C.E. from the University of Houston. He was instructor of civil engineering, Rice University, 1963-65, and is presently project engineer with Hudson Engineering Corp.

APPENDIX

Timoshenko's Equation 177 (*13*) for determining average contact pressure:

$$P_c = -\tfrac{1}{2}\,\sigma_t(r_2^2 - r_1^2)/r_2^2$$

where $\sigma_t = (\epsilon_t + \nu_t\epsilon_a)\,[E_t/(1 - \nu_t^2)]$

Gardner and Carnavos Equation 22 (*12*) for determining gap resistance:

$$r_g = \rho\left\{(\alpha_f - \alpha_t)\,(T_h - T_o) - \mu P_{co} - \left[\alpha_f\left(1 - \frac{r_o}{R^* + r_g}\right) - \alpha_t\left(\frac{r_i}{R^* + r_g}\right)\right](T_h - T_a)\right\}$$

Gardner and Carnavos Equation 22a (*12*) linearized version of Equation 22:

$$r_g = R^*\frac{\left\{[(\alpha_f - \alpha_t)\,(T_h - T_o) - \mu P_{co}] - \left[\alpha_f\left(1 - \frac{r_o}{R^*}\right) - \alpha_t\left(\frac{r_i}{R^*}\right)\right](T_h - T_a)\right\}}{\frac{R^*}{\rho} + \left[\alpha_f\left(\frac{r_o}{R^*}\right) + \alpha_t\left(\frac{r_i}{R^*}\right)\right](T_h - T_a)}$$

where (Equation 20) $\mu = \left\{\frac{1}{E_f}\left[\frac{D^2 + d^2}{D^2 - d^2} + \nu_f\right] + \frac{1}{E_t}\frac{t_f}{P}\left[\frac{d^2 + (d - 2t)^2}{d^2 - (d - 2t)^2} - \nu_t\right]\right\}$

and (Equation 23) $\rho = (D^2 - d^2)/48bK_e$

Young and Briggs Equation 20 (*21*) for determining contact pressure:

$$P_{co} = \left[\alpha_f(T_{fin} - 70) - \alpha_t(T_w - 70) - \left(\frac{24k_e r_b}{d}\right)\right]/\mu$$

for isothermal conditions $r_b = 0$, thus,

$$P_{CO} = [\alpha_f(T_{fin} - 70) - \alpha_t(T_w - 70)]/\mu$$

where μ is given in Equation 22a above and T_{fin} is the average temperature of fin metals, °F. #

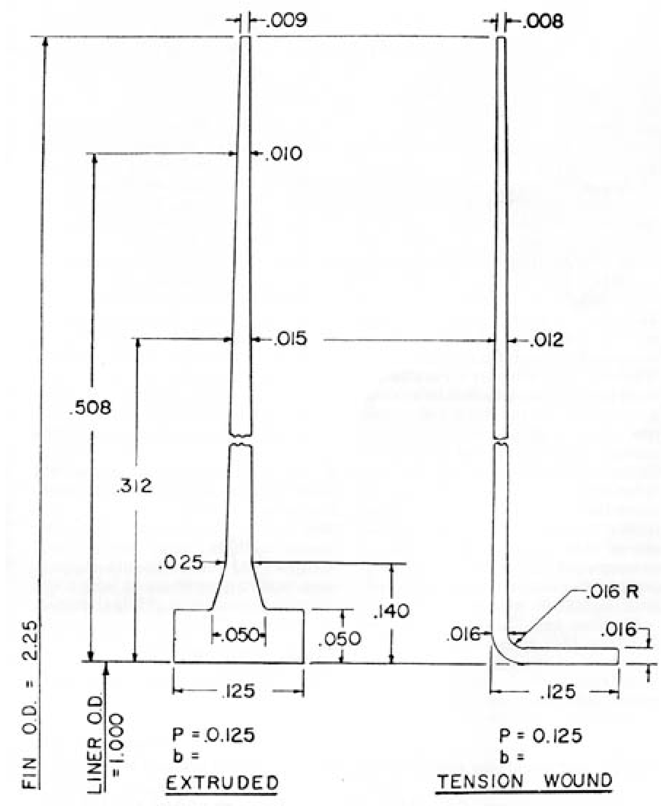

Figure 1. Scale drawing of extruded and footed tension wound interference fit fin tubes. Dimensions in inches.

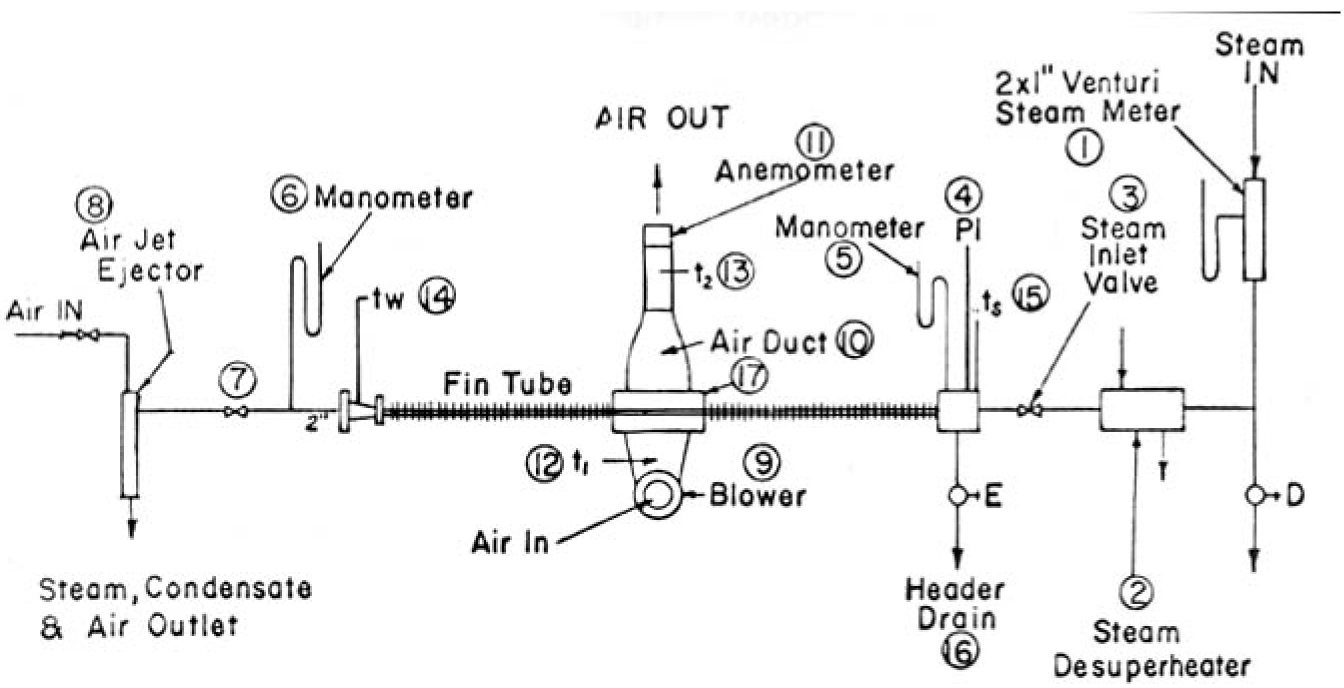

Figure 2. Schematic diagram of Hudson single-tube tester.

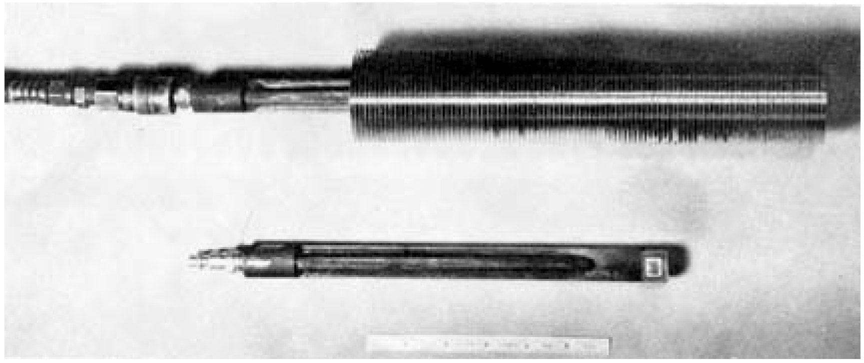

Figure 3. Strain gauge mounting device.

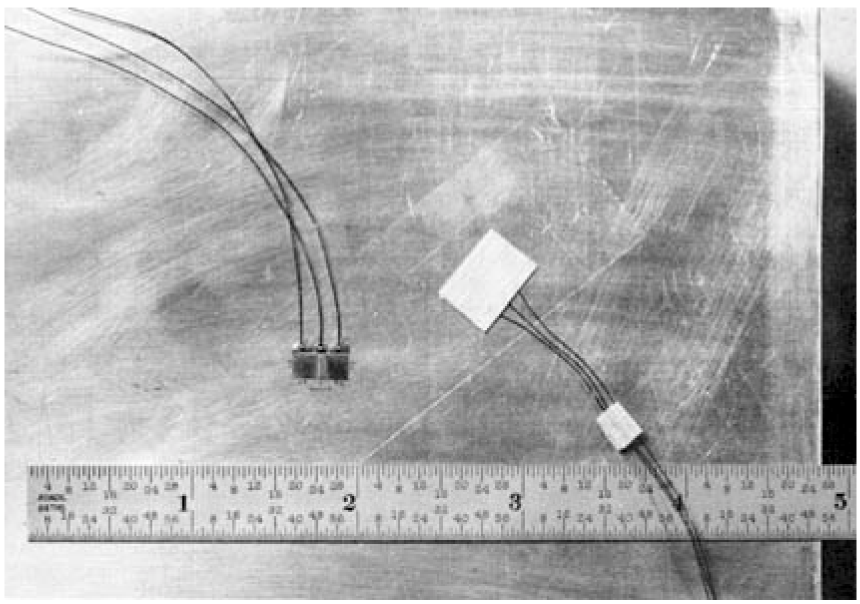

Figure 4. Strain gauges with leads attached just prior to mounting.

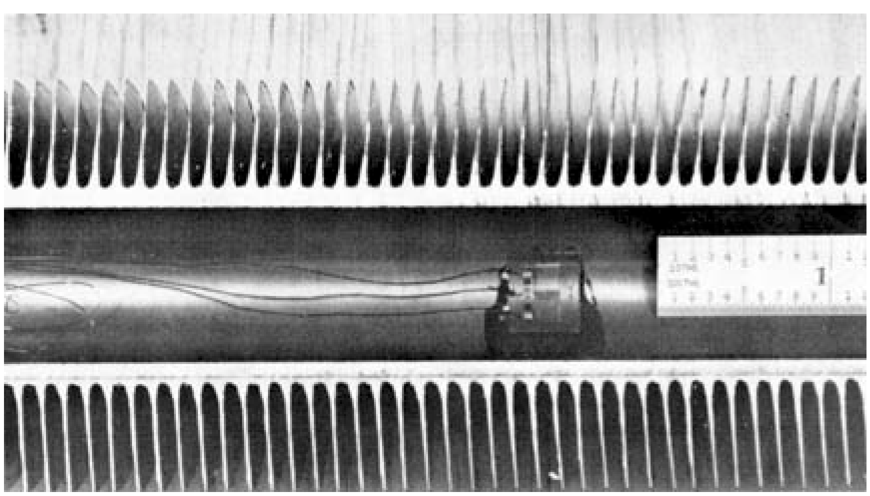

Figure 5. Tube slit after testing showing strain gauge orientation and locations..

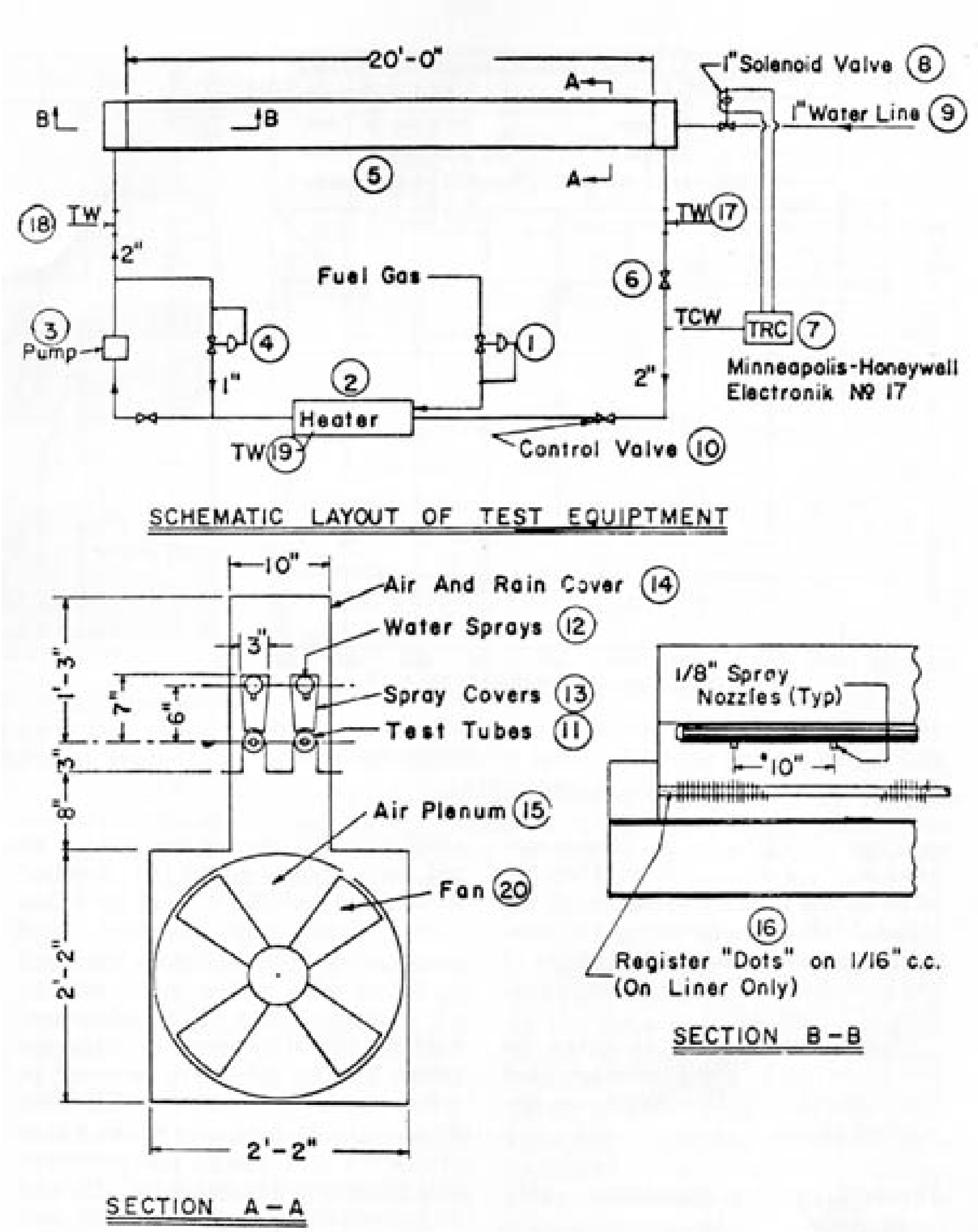

Figure 6. Schematic layout of fin tube cycling equipment.

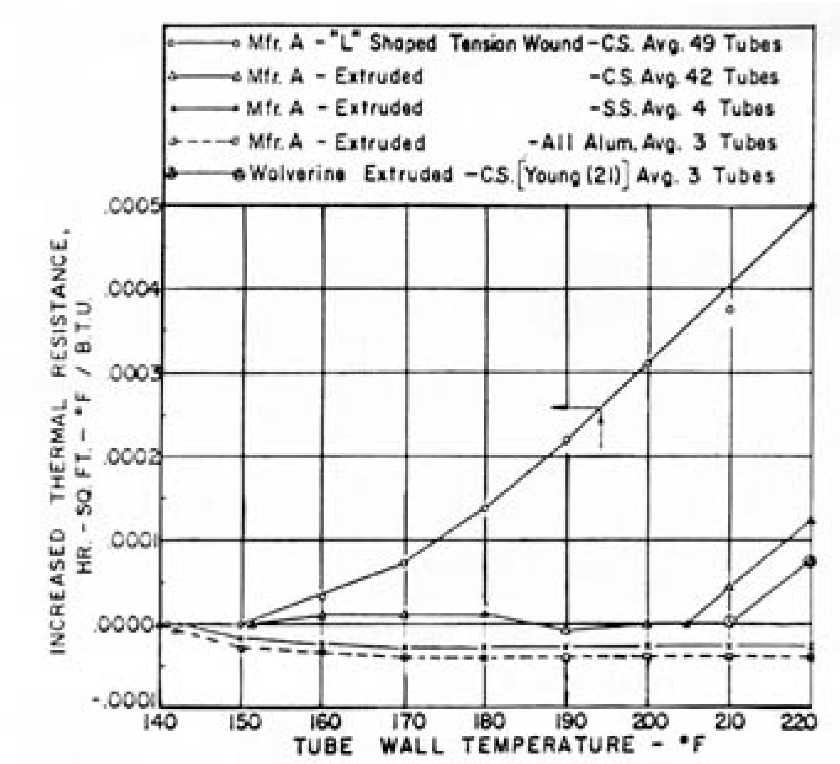

Figure 7. Single-tube temperature range tests on extruded and footed tension wound fin tubes of various metallurgy (Manufacturer A Test point spread ±5%)

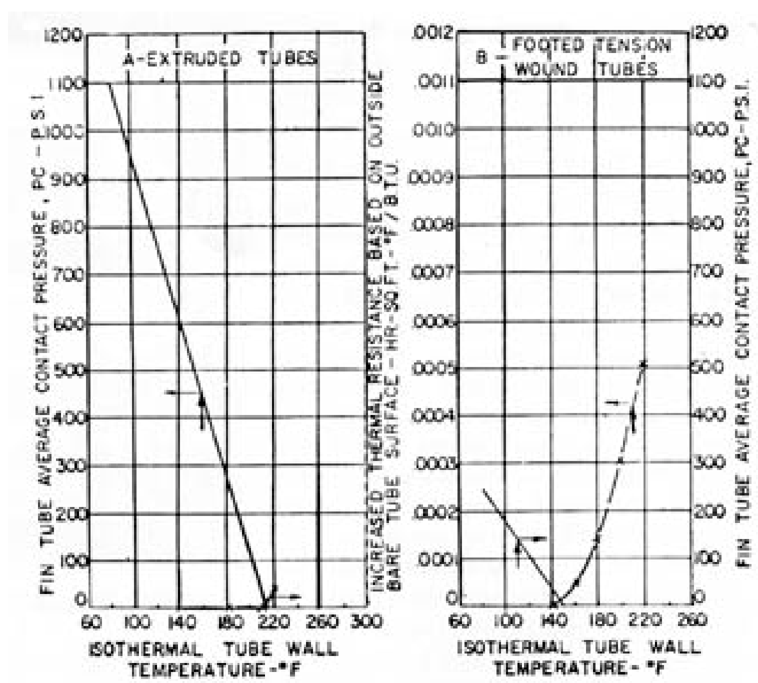

Figure 8. Strain gauge tests (average of 10 and 7 tubes each) comparing average contact pressure with isothermal tube wall temperature (Manufacturer A Test point spread ±5%)

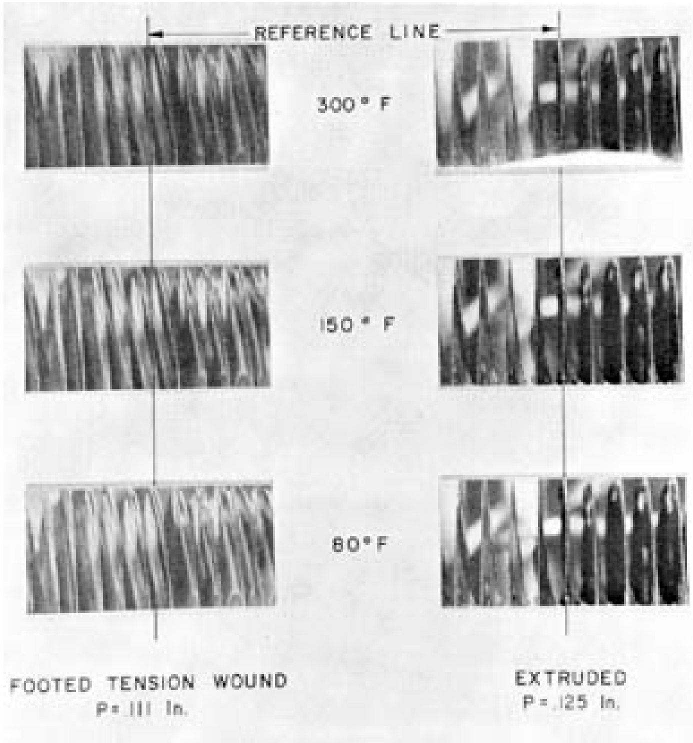

Figure 9. Photograph showing lateral movement of footed tension wound and extruded fin tips with increased isothermal temperature.

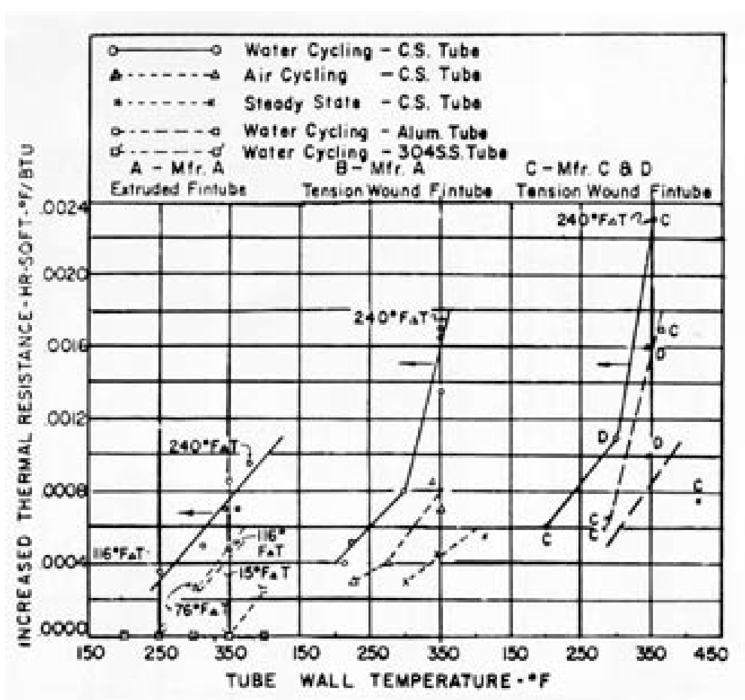

Figure 10. Increased thermal resistance r_s, for extruded and tension wound fin tubes under steady state or air and water cycling, at 500 cycles, and various tube wall temperatures.

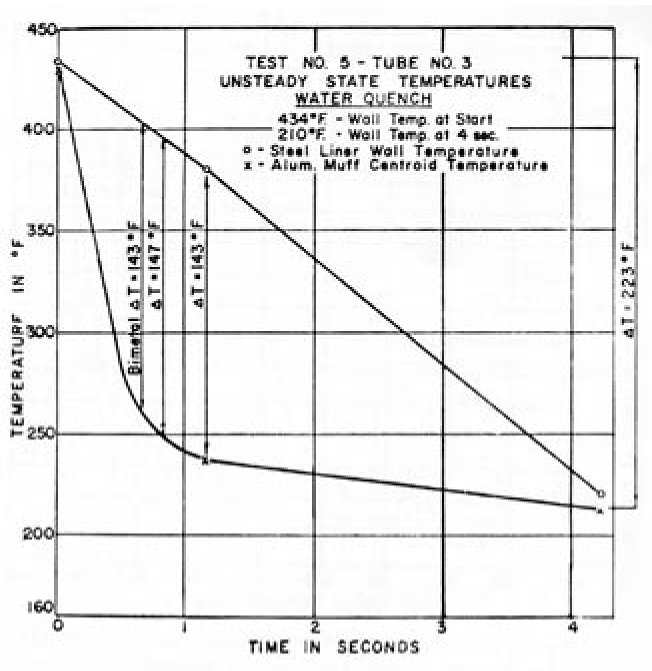

Figure 11. Approximate time-temperature curve for fin and tube liner metal at start of cooling cycle.

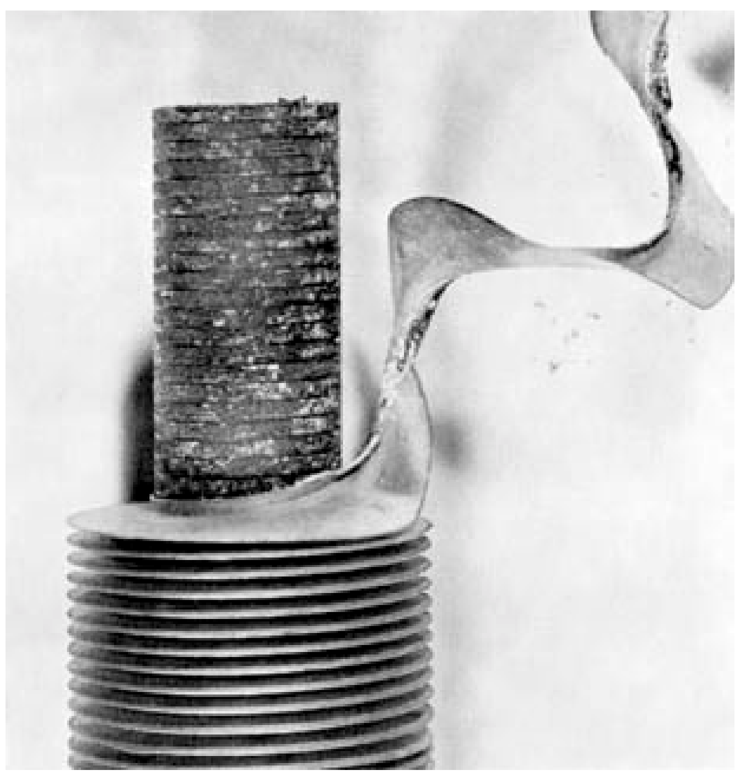

Figure 12. Typical footed tension wound fin tube taken from bundles returned from Gulf Coast petrochemical plant. Tubes had been sprayed with water during summer months of two years.

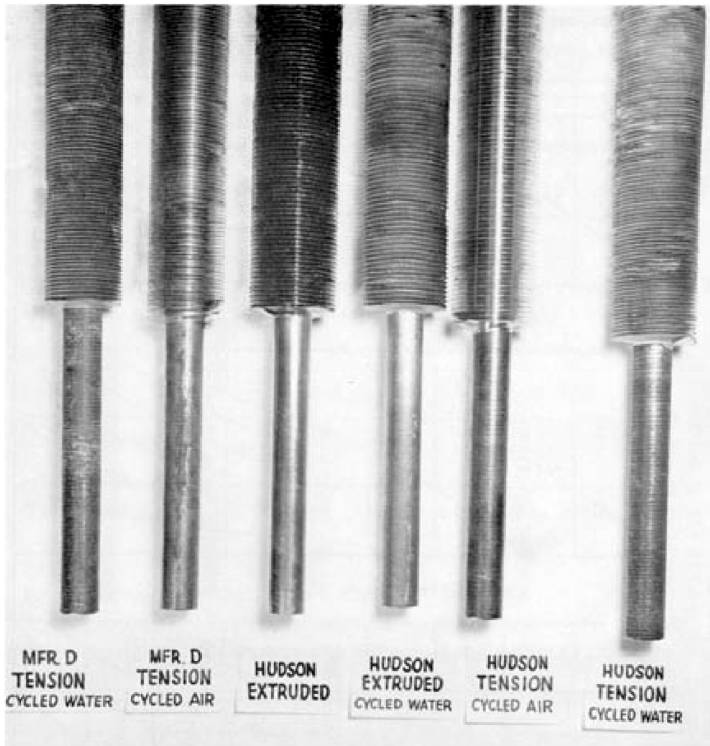

Figure 13. Stripped tube ends of tubes cycled with both air and water (500 cycles, 48-hr. test, 250°F tube wall).

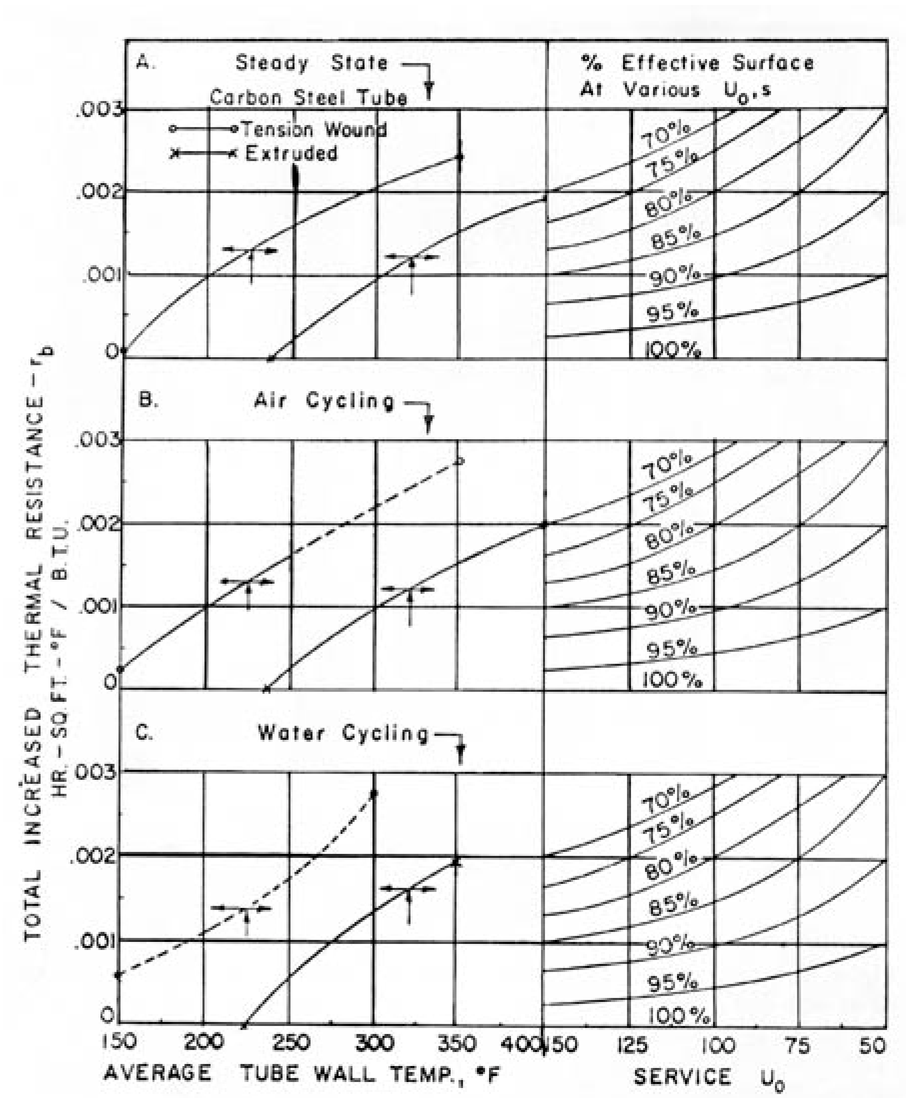

Figure 14. Curves showing total increased bond resistance, r_b, and % derating, for carbon steel extruded and footed tension wound fin tubes under steady state, air and water cycling at various tube wall temperatures. Based on T_a = 125°F, $1/U_o = 0.0133$ based on A_o, $r_o = 0.0061$ and tube wall temperature at unit outlet high enough to exhaust P_{co}. See Table 4 for other data.

Table 1. Isothermal tube wall temperature at which contact pressure is exhausted.

Type of fin	Fin metallurgy	Liner	Gardner (*20*)	Young & Briggs (*21*)
Extruded	Aluminum	Carbon steel	280°F	200°F
Footed tension wound	Aluminum	Carbon steel	260°F	

Table 2. Instrumentation accuracy.

	Single-tube tester	Strain gauge apparatus	Cycling apparatus
Fluid temperature, °F			
Tube	±0.5	—	± 5
Air	±0.2	—	±10
Tube wall temperature, °F	±0.5	±1.0	±10
Increased heat transfer resistance	±0.00015	—	±0.00015
Strain			
μ in./in.		±10	
contact pressure, lb./sq. in.		±50	

Table 3. Comparison of prior data for T* and P_{co} with present data.

	Extruded P_{co}, lb./sq. in.	Extruded P_c, lb./sq. in.	Extruded T^*, °F	Footed tension wound P_{co}, lb./sq. in.	Footed tension wound P_c, lb./sq. in.	Footed tension wound T^*, °F
Gardner (*20*) recommendation	—	—	280	—	—	260
Young & Briggs (*21*); 1-in. tube, nine ½ in. fins/in., manufactured at 70°F.						
Single Tube	3700	—	200	—	—	—
Authors; 1-in. tube, eight ⅝ in. fins/in., manufactured at 80°F.						
Single-tube tests	3760	—	198	2170	—	145
Strain gauge tests	4310	1100	215	2340	250	150

Table 4. Comparison of P_{co}, μ,. and dimension b based on strain gauge tests for eight 5/8-in. fins/in. extruded and footed tension wound fin tubes at 80°F manufacturing temperature.

	P_{co} Liner lb./sq. in.	P_{co} Fin side lb./sq. in	μ^* 0.016-in. t_f $\times 10^{-6}$	μ^* 0.025-in. t_f $\times 10^{-6}$	μ^* 0.05-in. t_f $\times 10^{-6}$	b P_{co} present, in.	b P_{co} exhausted, in.
Extruded	4,310	5,860**	0.203	0.213	0.251	0.032	0.125
Footed tension wound	2,340	—	0.203	—	—	0.013	0.125

Based on Poisson ratio of 0.33 for aluminum and 0.285 steel, and 10 × 10⁶ and 30 × 10⁶ moduli of elasticity, respectively.
**At point B = 0.14-in. above O.D. of tube liner.*

Table 5. Coefficient of thermal expansion for various metals.

Metal	Coefficient, $\times 10^{-6}$
Aluminum	13.5
Admiralty	10.0
Copper	9.6
Carbon steel	6.7
Austenitic stainless steel	10.5