for . . .

engineers and contractors controlling the design, specification, installation and maintenance of heating, ventilating and air conditioning.

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EDITORIAL STAFF
Editor
CLIFFORD STROCK
Associate Editor
JOSEPH F. KERN, JR.
Contributing Editor
C. H. B. HOTCHKISS

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REPRESENTATIVES
Eastern Representative
ALGWYNNE COLLINS
148 Lafayette Street
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Western Representative
GEORGE G. TURNER
228 North La Salle Street
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Front Cover—The radiant heated room on this month's cover is presented through the courtesy of Wolff & Munier, Inc., New York.

MEMBER
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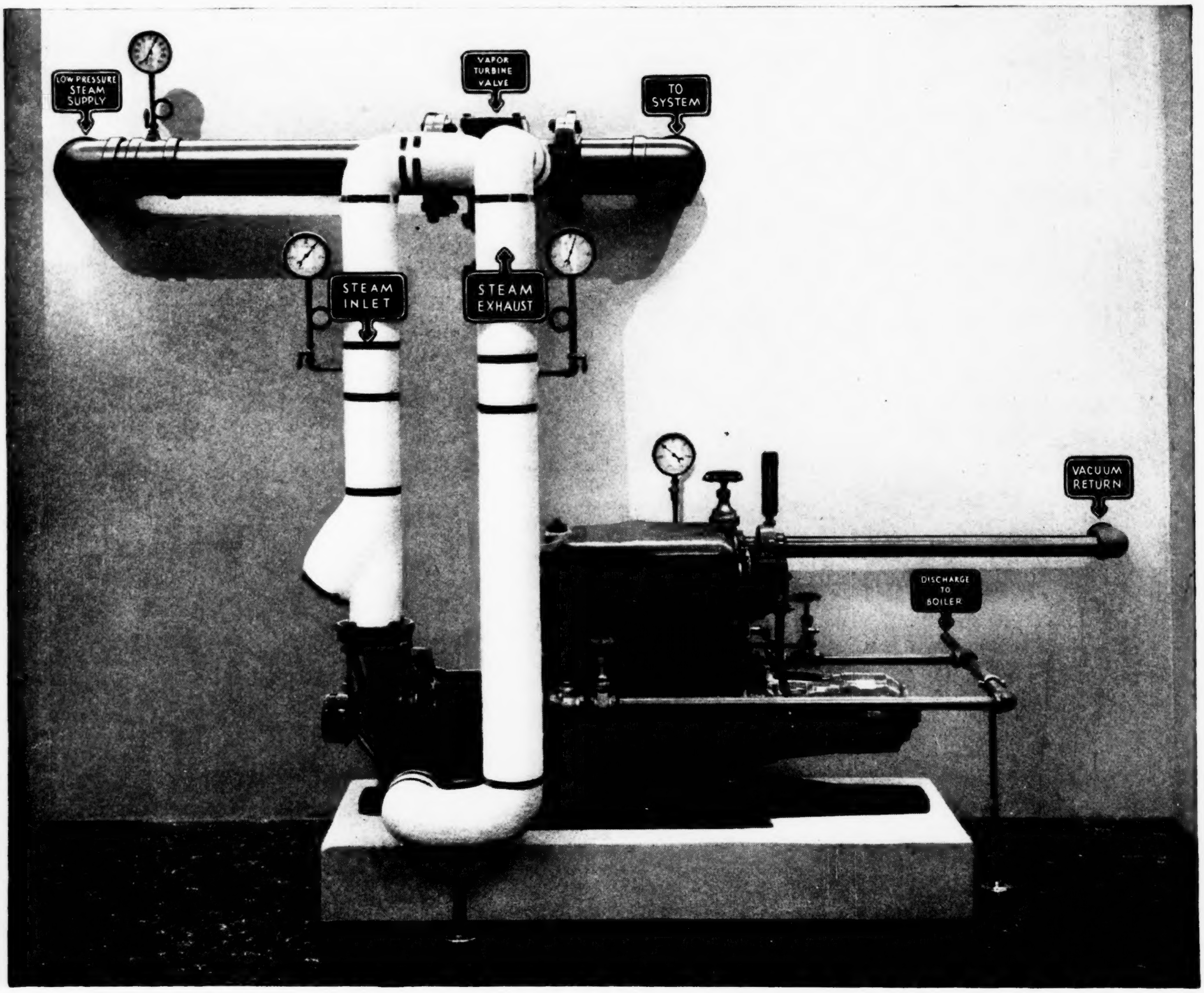
Where Heat Must Not Fail— Install this SAFE Heating Pump!



In Hospitals, Greenhouses, Schools, Public Buildings, Theatres, wherever heating systems must not fail, install the Nash Vapor Turbine, for it is independent of electric current failure, and continues to operate as long as there is steam in the system.

This is because the prime motive power of this economical pump is a special steam turbine, controlled by a unique "Vapor Turbine Valve", which automatically bypasses from the heating main a small portion of steam, the exact amount necessary to develop the power needed to remove the condensate and maintain the required vacuum on the system. Even this small amount is passed immediately back to the mains, and goes on to the system with little heat loss. This pump operates on any system, high or low pressure.

The Vapor Turbine is a most economical pump, for the elimination of electric current does away with current cost, the largest single item in the operation of an ordinary return line heating pump. Bulletin on request.

THE NASH ENGINEERING COMPANY

205-A S. WILSON ROAD, SOUTH NORWALK, CONNECTICUT, U. S. A.

How to Get Business in Ships of the Maritime Commission

RECENT Congressional authorization of the construction of 200 emergency cargo vessels at an estimated cost of $300,000,000 raises to $850,000,000 the Maritime Commission's ship building program under which suppliers of heating, ventilating and air conditioning equipment will share amounts variously estimated at between $24,000,000 and $30,000,000 during the next two years.

While news of the bulk order for 200 emergency ships holds the headlines in shipping circles, indications are that the Commission's 50-ships-per-year program, started in 1938 and scheduled to continue for 10 years, will be of greatest interest to equipment suppliers. The emergency ships, representatives of the Maritime Commission explain, will be just what the name implies—emergency cargo carriers with a maximum of carrying space and a minimum of frills or refinements.

While natural ventilation and radiator heating of the simplest type will be the rule in the emergency vessels, those included in the Commission's primary building program embody heating, ventilating and air conditioning facilities of the latest types.

Under the Commission's primary program contracts have been let for 198 vessels, of which 66 have been delivered and 95 have been launched. Contracts for between 30 and 40 additional ships will be let during 1941, it is expected, for the building program is already ahead of the projected one-a-week schedule and wartime demands for bottoms exceeds that contemplated when Congress authorized the program in 1936.

Since the Maritime Commission's construction program is handled entirely under contract with shipbuilding companies, either for the Commission's own account or in conjunction with private operators, the starting point for suppliers desiring to participate in the program is the shipyard.

The Commission's files in the U. S. Department of Commerce Building at Washington contain copies of all drawings and specifications, most of which are open to inspection by representatives of supply firms, but the Commission does not maintain lists of equipment firms for the purpose of mailing out bid proposals.

Manufacturers' representatives desiring to discuss new or specialized equipment with a view to proposing its adoption by the Commission are interviewed at the office of the Director of Technical Divisions, Maritime Commission, but all transactions are finalized through the shipyard engaged in actual construction work.

With this procedure in mind, accompanying lists of ships under construction, holders of sub-contracts and locations of yards may prove of interest to suppliers of heating, ventilating and air conditioning equipment. Actual contracts for the emergency fleet have not been awarded, although construction of shipways has been authorized as shown at top, next page.

As previously stated, ships to be constructed at the

Company	Location	No. of Ways	Cost of Facilities	No. of Ships
Oregon Shipbuilding Co.	Portland, Ore.	8	$4,787,000	31
California Shipbuilding Co.	Los Angeles, Cal.	8	4,766,000	31
Houston Shipbuilding Co.	Houston, Texas	6	4,680,000	25
Alabama Dry Dock & Shipbuilding Co.	Mobile, Ala.	4	1,322,000	13
North Carolina Shipbuilding Co.	Wilmington, N. C.	6	5,140,000	25
Bethlehem-Fairfield Shipbuilding Co.	Baltimore, Md.	13	7,838,000	50
Louisiana Shipbuilding Co.	New Orleans, La.	6	4,841,000	25
		51	$33,374,500	200

yards listed above do not represent a particularly rich market for suppliers of heating and ventilating equipment. Ships in the tabulation below, however, are the finest and speediest of their class, and a wide variety of modern equipment has been purchased for those completed and will be required for those just launched or still on the ways.

That one third of the vessels authorized in the following schedule are completed should not be interpreted as a discouraging factor to suppliers, for additional contracts are to be let as rapidly as ways are vacated or new ship building facilities are made available. As of February 25, the Maritime Commission's primary program is tabulated at the bottom of this page.

The importance attached to proper heating and ventilating of the new ships being built under direction of the Maritime Commission is indicated when it is recalled that the initial purpose of the program was to provide an auxiliary merchant marine to serve the United States Naval fleet. As explained by Captain Howard L. Vickery, U. S. N., member of the United States Maritime Commission:

"First the Navy and Congress knew that we needed merchant ships for auxiliaries. Second, a program for building these ships was started with Congressional authorization and appropriations. Third, the Navy decided what types of vessels it would need. Fourth, the agency created by Congress to build these ships, the Maritime Commission, designed the vessels with auxiliary needs in mind.

"As a result, 25 of the new merchant ships are already engaged in Naval service as follows: twelve 19-knot tankers as fuel ships, four 17½-knot C-3 cargo ships as submarine and seaplane tenders; eight 15-knot C-2 cargo ships as store ships, store issue ships, store refrigeration ships, ammunition carriers and transport; one 14-knot C-1 ship as a submarine tender."

Designed to keep pace with the fleet in speed, dependability and smartness of appearance, equipment throughout, including heating and ventilating, is of a type unknown to earlier merchant vessels not designed for such dual use.

"In the last seven years," continued Captain Vickery, "improved housing for low-income groups of the population has been the subject of great agitation and considerable accomplishment. Much has been accomplished at the same time in the analogous field of making shipboard living spaces more attractive.

"Accommodations which have been designed and installed on the 65 new ships in operation and on the 133 others still building are superior to the accommodations which most seamen find available when they go ashore. And I think they should be. The efficiency of a ship and her crew is enhanced by contentment with rooms and food. Clean, spacious quarters, adequate showers and toilet facilities, good ventilation and heating all contribute to the operation of a ship.

"Not too long ago seamen slept in crowded rooms filled with three deck bunks and illuminated by a few small bulbs dangling from a cord. The sides of the rooms were hull plates and the air was usually as cold in winter and hotter in summer than the outdoors.

"In C-2 and C-3 vessels, the crew have comfortable quarters with wash basins with hot and cold running water in each room. The quarters on the new ships are in one house except that provision is made for gunners' quarters on the poop at their guns. Ventilation is of the same type on both, using airports and mushroom or torpedo type ventilators."

Heating Systems

With the exception of the radiator-and-natural-ventilation systems prescribed for the emergency fleet, heating and ventilating of Maritime Commission vessels is accomplished by Split Systems, Central Systems, Dual Duct Systems and Unit Systems.

Type of Vessel	No.	Heating	Keels Laid	Launched	Delivered
Passenger (America)	1	Dual duct heating, fully air conditioned	1	1	1
C-1 cargo, 8 pass.	38	Central heating system, with exception of a few having zoning reheaters	34	22	7
C-2 cargo, 12 pass.	41		26	20	20
C-3 cargo, 12 pass.	22		18	18	13
C-3 cargo, 96 pass.	18	Dual-duct on passenger area; central, crew	15	7	2
Cargo and 67 pass.	6	Dual-duct on passenger area; central, crew	3	3	3
Cargo	12	Split system, radiator heating	10	8	7
Cargo	6	Central system	6	3	0
Cargo	8	Central system	0	0	0
Cargo	3	Central system	0	0	0
Tankers	23	Split system	14	12	12
Transports	4	Not disclosed	0	0	0
Newly authorized	16	Types and heating systems not disclosed	0	0	0
	198		127	94	65

The Split system consists of heating with radiators and supplying either unheated or heated ventilation by means of fans. Tankers use unheated forced air, but in other types, the air is heated to 70 or 80F. This system is confined chiefly to cargo ships or crews' quarters, third class passenger quarters and certain office and working space on passenger ships.

The Central system is taking the place of the split system on many of the new cargo ships. The advantage being that little equipment or space is required beyond that required for ventilation. Air is taken into a fan room on the motor deck, forward of the stack, then filtered, pre-heated and later re-heated to the final temperature necessary to heat to 70F. It is usually necessary to raise the final air temperature to 90 to 110F, depending upon the volume of air supplied and the outside temperature.

The Dual Duct system is used only on the Maritime Commission's passenger ships, either for passenger quarters or for the entire crew. The system includes two systems of ducts, one carrying air heated to a maximum of 130F and the other carry air cooled to 40 or 60F. The two ducts are joined by a mixing damper in the areas to be heated, permitting individual control to suit the requirements of occupants.

Unit Systems are self-contained heating and cooling units designed to air-condition spaces as small as staterooms and as large as lounges or public rooms. They include a circulating water heating unit providing temperatures to 150F, and a cold water unit providing temperatures to 45F.

Supplementary heaters — steam, unit type and in some instances electric—are frequently required in remote portions of the ship. Electric heating is often favored in radio rooms.

Cargo Hold Ventilation

No ship of the Commission's regular program has been or will be built without some of the holds being mechanically ventilated and a high percentage of refrigerator capacity is provided for foods.

Sixty of the cargo ships built or contracted for are designed to control completely the humidity in general cargo holds, protecting shipments from damage due to moisture or sweating. The system adopted employs a new system of humidity control, in addition to providing an average of three changes per hour.

As a guide to probable heating and ventilating requirements, it is estimated that an average of from 3% to 4% of the cost of the ship is required for purchase and installation of such equipment. Following are brief descriptions of the types of ships authorized:

Cargo C-1: Average cost, $2,000,000: The C-1's are the smallest of the ships being constructed by the Commission. Provision for five holds and eight passengers.

Cargo C-2: Average cost, $2,500,000: These are 15½ knot vessels of shelter deck type, with five holds, usually one or more refrigerated.

Cargo C-3: Average cost, $2,750,000: Designed to satisfy the need for a vessel of greater deadweight cargo capacity and greater speed than the C-2 design. Turbine or steam power drives the ship at 16½ knots; five cargo holds and space for 12 passengers are provided.

American Export Lines: Average cost, $2,265,000: Somewhat similar to the C-3 type, these are steam turbine vessels of the shelter deck type. They have seven holds and no passenger quarters.

Seas Shipping Co.: Average cost, $2,400,000: Of the shelter deck type, the ships have a raked stem and a cruiser stern and are steam propelled. Five cargo holds are provided, accommodations for 12 passengers and the crew are located amidships.

C-3 Cargo and Passenger: Average cost, $4,000,000: As designed for Round-the-World service of the American President lines, the hull is essentially the same as C-3 cargo ships, with suitable alterations for 96 passengers and 124 officers and crew. These quarters are designed and fitted to assure a high degree of comfort.

Mississippi Shipping Co.: Average cost, $3,150,000: Somewhat similar to C-3 passenger and cargo vessels, these vessels provide for 67 passengers.

Status of Program

With the exception of the 16 new vessels authorized late in February, the incompleted portion of the Maritime Commission's primary program is as follows:

SHIPS UNDER CONSTRUCTION FOR MARITIME COMMISSION

TYPE	NAME	OPERATOR	BUILDER	% COMPLETED
C-3 pass. and cargo	*President Hayes*	American President Lines	Newport News Shipbldg. & Drydock Co.	94.3
C-3 pass. and cargo	*President Garfield*	American President Lines	Newport News Shipbldg. & Drydock Co.	89.2
C-3 pass. and cargo	*President Adams*	American President Lines	Newport News Shipbldg. & Drydock Co.	76.0
C-3 pass. and cargo	*President Van Buren*	American President Lines	Newport News Shipbldg. & Drydock Co.	48.5
C-3 pass. and cargo	*Rio Hudson*	Moore-McCormack Lines, Inc.	Sun Shipbldg. & Drydock Co.	61.5
C-3 pass. and cargo	*Rio Parana*	Moore-McCormack Lines, Inc.	Sun Shipbldg. & Drydock Co.	54.5
C-3 pass. and cargo	*Rio LaPlata*	Moore-McCormack Lines, Inc.	Sun Shipbldg. & Drydock Co.	48.6
C-3 pass. and cargo	*Rio DeJaneiro*	Moore-McCormack Lines, Inc.	Sun Shipbldg. & Drydock Co.	44.2
C-3 cargo	*Mormactide*	Moore-McCormack Lines, Inc.	Ingalls Shipbldg. Corp., Pascagoula, Miss.	95.5
C-3 cargo	*Hull 256**	Moore-McCormack Lines, Inc.	Ingalls Shipbldg. Corp., Pascagoula, Miss.	86.1
Cargo	*Robin Locksley*	Seas Shipping Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Point, Md.	96.0
Cargo	*Robin Doncaster*	Seas Shipping Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Point, Md.	81.9
Cargo	*Robin Kettering*	Seas Shipping Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Point, Md.	66.1
C1B cargo	*Cape Mendocino*	N. Y. & Cuba Mail SS Co., Inc.	Consolidated Steel Corporation, Ltd.	71.4
C1B cargo	*Hull 57*	N. Y. & Cuba Mail SS Co., Inc.	Consolidated Steel Corporation, Ltd.	57.1
C1B cargo	*Hull 158*	N. Y. & Cuba Mail SS Co., Inc.	Consolidated Steel Corporation, Ltd.	38.6
C1B cargo	*Hull 159*	N. Y. & Cuba Mail SS Co., Inc.	Consolidated Steel Corporation, Ltd.	23.4
C1B cargo	*Solon Turman*	Lykes Bros. SS Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Point, Md.	73.0
C1B cargo	*Thompson Lykes*	Lykes Bros. SS Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Point, Md.	83.1
C1B cargo	*James McKay*	Lykes Bros. SS Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Point, Md.	40.8
C1B cargo	*Cape San Martin*	Grace Lines, Inc.	Bethlehem Steel Co., Shipbldg. Div., San Francisco, Calif.	99.8
C1B cargo	*Alcoa Pioneer*	Alcoa SS Co., Inc.	Bethlehem Steel Co., Shipbldg. Div., San Francisco, Calif.	95.4
C1B cargo	*Alcoa Pilgrim*	Alcoa SS Co., Inc.	Bethlehem Steel Co., Shipbldg. Div., San Francisco, Calif.	81.1
C1B cargo	*Alcoa Patriot*	Alcoa SS Co., Inc.	Bethlehem Steel Co., Shipbldg. Div., San Francisco, Calif.	64.6
C1B cargo	*Hull 5364*	Alcoa SS Co., Inc.	Bethlehem Steel Co., Shipbldg. Div., San Francisco, Calif.	48.8

(Continued on next page)

SHIPS UNDER CONSTRUCTION FOR MARITIME COMMISSION *(Continued from Preceding Page)*

Type	Name	Operator	Builder	% Completed
C1B cargo	*Cape Ann*	Alcoa SS Co., Inc.	Bethlehem Steel Co., Shipbldg. Div., Staten Island, N. Y.	95.4
C1B cargo	*Cape Nedick*	Alcoa SS Co., Inc.	Bethlehem Steel Co., Shipbldg. Div., Staten Island, N. Y.	86.4
C1B cargo	*Cape Cod*	Alcoa SS Co., Inc.	Bethlehem Steel Co., Shipbldg. Div., Staten Island, N. Y.	65.4
C1B cargo	*Hull 8019*	Alcoa SS Co., Inc.	Bethlehem Steel Co., Shipbldg. Div., Staten Island, N. Y.	54.2
C1B cargo	*Amer. Manufacturer*	U. S. Lines	Western Pipe & Steel Co.	97.0
C1B cargo	*American Leader*	U. S. Lines	Western Pipe & Steel Co.	83.8
C1B cargo	*American Builder*	U. S. Lines	Western Pipe & Steel Co.	69.0
C1B cargo	*American Press*	U. S. Lines	Western Pipe & Steel Co.	54.1
C1B cargo	*American Packer*	U. S. Lines	Western Pipe & Steel Co.	38.5
C2S cargo	*Robin Sherwood*	Seas Shipping Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Pt., Md.	43.1
C2S cargo	*Robin Tuxford*	Seas Shipping Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Pt., Md.	32.3
C2S cargo	*Robin Wentley*	Seas Shipping Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Pt., Md.	26.3
C3E cargo	*Examiner*	American Export Lines	Bethlehem Steel Co., Shipbldg. Div., Fore River, Quincy, Mass.	80.0
C3P pass. and cargo	*Hull 265*	American South African Line	Ingalls Shipbldg. Corp., Pascagoula, Miss.	39.3
C3P pass. and cargo	*Hull 266*	American South African Line	Ingalls Shipbldg. Corp., Pascagoula, Miss.	36.1
C3P pass. and cargo	*Hull 267*	American South African Line	Ingalls Shipbldg. Corp., Pascagoula, Miss.	30.5
C3IN pass. and cargo	*Hull 268*	U. S. Lines	Ingalls Shipbldg. Corp., Pascagoula, Miss.	20.6
C3A pass. and cargo	*President Polk*	American President Lines	Newport News Shipbldg. & Drydock Co.	42.0
C2SU cargo	*Japan Mail*	American Mail Line	Sun Shipbldg. & Drydock Co.	8.5
C2SU cargo	*China Mail*	American Mail Line	Sun Shipbldg. & Drydock Co.	7.8
C2SU cargo	*Island Mail*	American Mail Line	Sun Shipbldg. & Drydock Co.	5.5
C2SU cargo	*Hull 202*	U. S. Lines	Sun Shipbldg. & Drydock Co.	5.5
C2SU cargo	*Hull 203*	U. S. Lines	Sun Shipbldg. & Drydock Co.	1.8
C2SU cargo	*Hull 204*	U. S. Lines	Sun Shipbldg. & Drydock Co.	1.6
C2SU cargo	*Hull 205*	U. S. Lines	Sun Shipbldg. & Drydock Co.	1.6
C2SU cargo	*Hull 206*	U. S. Lines	Sun Shipbldg. & Drydock Co.	1.6
C1B cargo	*Cape Alava*	American Mail Line	Seattle-Tacoma Shipbldg. Corp.	93.6
C1B cargo	*Cape Flattery*	American Mail Line	Seattle-Tacoma Shipbldg. Corp.	86.3
C1B cargo	*Cape Clear*	American Mail Line	Seattle-Tacoma Shipbldg. Corp.	73.6
C1B cargo	*Cape Fairweather*	American Mail Line	Seattle-Tacoma Shipbldg. Corp.	64.2
C1B cargo	*Hull 40*	American Mail Line	Seattle-Tacoma Shipbldg. Corp.	46.1
C2T cargo	*Rainbow*	U. S. Lines	Tampa Shipbldg. Co., Inc.	52.5
C2T cargo	*Hull 38*	U. S. Lines	Tampa Shipbldg. Co., Inc.	44.5
C2T cargo	*Hull 39*	U. S. Lines	Tampa Shipbldg. Co., Inc.	27.0
C2T cargo	*Hull 40*	U. S. Lines	Tampa Shipbldg. Co., Inc.	26.8
C2G cargo	*Hull 179*	Grace Line	Federal Shipbldg. & Drydock Co.	61.6
C2F cargo	*Hull 180*	Lykes Bros.	Federal Shipbldg. & Drydock Co.	39.7
C2F cargo	*Hull 181*	Lykes Bros.	Federal Shipbldg. & Drydock Co.	38.0
C2F cargo	*Hull 182*	Lykes Bros.	Federal Shipbldg. & Drydock Co.	36.5
C2F cargo	*Hull 183*	Lykes Bros.	Federal Shipbldg. & Drydock Co.	27.7
C2F cargo	*Seattle Mail*	American Mail Line	Federal Shipbldg. & Drydock Co.	25.0
C2F cargo	*Tacoma Mail*	American Mail Line	Federal Shipbldg. & Drydock Co.	23.4
C2F cargo	*Portland Mail*	American Mail Line	Federal Shipbldg. & Drydock Co.	21.6
C3M cargo	*Mormacsea*	Moore-McCormack Lines, Inc.	Moore Drydock Co.	97.5
C3M cargo	*Mormacsun*	Moore-McCormack Lines, Inc.	Moore Drydock Co.	87.7
C1A cargo	*Cape Lookout*	Moore-McCormack Lines, Inc.	Penn Shipyards, Inc.	58.3
C1A cargo	*Hull 228*	Moore-McCormack Lines, Inc.	Penn Shipyards, Inc.	21.7
C1A cargo	*Marina*	Bull Line	Pusey & Jones Co.	61.5
C1A cargo	*Hull 1076*	Bull Line	Pusey & Jones Co.	36.5
Tanker	*Corsicana*	Socony-Vacuum Oil Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Pt., Md.	51.0
Tanker	*Caddo*	Socony-Vacuum Oil Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Pt., Md.	43.5
Tanker	*Calusa*	Socony-Vacuum Oil Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Pt., Md.	11.5
Tanker	*Catawba*	Socony-Vacuum Oil Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Pt., Md.	9.5
Tanker	*Colina*	Socony-Vacuum Oil Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Pt., Md.	4.7
Tanker	*Conastoga*	Socony-Vacuum Oil Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Pt., Md.	4.7
Tanker	*Hull 221*	Keystone Tankship Corp.	Sun Shipbldg. & Drydock Co.	0.0
Tanker	*Hull 222*	Keystone Tankship Corp.	Sun Shipbldg. & Drydock Co.	0.0
Passenger and cargo	*Hull 4362*	Miss. Shipbuilding Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Pt., Md.	0.0
Passenger and cargo	*Hull 4363*	Miss. Shipbuilding Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Pt., Md.	0.0
Passenger and cargo	*Hull 4364*	Miss. Shipbuilding Co.	Bethlehem Steel Co., Shipbldg. Div., Sparrows Pt., Md.	0.0
C2SAL cargo	*Exceller*	American Exp. Lines	Bath Iron Works, Bath, Me.	36.4
C2SAL cargo	*Extavia*	American Exp. Lines	Bath Iron Works, Bath, Me.	33.5
C2SAL cargo	*Exanthia*	American Exp. Lines	Bath Iron Works, Bath, Me.	16.4
C2SAL cargo	*Exiria*	American Exp. Lines	Bath Iron Works, Bath, Me.	16.4
Tanker	*Hull 226*	Kaymar Tankship Corp.	Sun Shipbldg. & Drydock Co.	0.0
Tanker	*Hull 227*	Seamar Tankship Corp.	Sun Shipbldg. & Drydock Co.	0.0
Tanker	*Hull 228*	Seamar Tankship Corp.	Sun Shipbldg. & Drydock Co.	0.0
Design Z1 Schoolship	*American Sailor*	U. S. Coast Guard	Bethlehem Shipbldg. Div., Key Highway Plant, Baltimore	65.0
C3SA2 cargo	*Hull 293*	—	Ingalls Shipbldg. Corp., Pascagoula, Miss.	4.9
C3SA2 cargo	*Hull 294*	—	Ingalls Shipbldg. Corp., Pascagoula, Miss.	4.9
C3SA2 cargo	*Hull 295*	—	Ingalls Shipbldg. Corp., Pascagoula, Miss.	2.0
C3SA2 cargo	*Hull 296*	—	Ingalls Shipbldg. Corp., Pascagoula, Miss.	2.0
C3IN pass. and cargo	*Hull 297*	U. S. Lines	Ingalls Shipbldg. Corp., Pascagoula, Miss.	2.8
C3IN pass. and cargo	*Hull 298*	U. S. Lines	Ingalls Shipbldg. Corp., Pascagoula, Miss.	2.8
C3IN pass. and cargo	*Hull 299*	U. S. Lines	Ingalls Shipbldg. Corp., Pascagoula, Miss.	2.8
Transport	*Hull 6*	U. S. Army	Seattle-Tacoma Shipbldg. Corp.	0.0
Transport	*Hull 7*	U. S. Army	Seattle-Tacoma Shipbldg. Corp.	0.0
C3SA1	*Hull 8*	Isthmian SS Co.	Seattle-Tacoma Shipbldg. Corp.	0.0
C3SA1	*Hull 9*	Isthmian SS Co.	Seattle-Tacoma Shipbldg. Corp.	0.0
C3SA2 cargo	*Hull 62*	Isthmian SS Co.	Western Pipe & Steel Co.	0.0
C3SA2 cargo	*Hull 63*	Isthmian SS Co.	Western Pipe & Steel Co.	0.0
C3SA2 cargo	*Hull 64*	Isthmian SS Co.	Western Pipe & Steel Co.	0.0
C3SA2 cargo	*Hull 65*	Isthmian SS Co.	Western Pipe & Steel Co.	0.0
C2-S1-A1 cargo	*Hull 201*	Alcoa SS Co., Inc.	Moore Drydock Co.	0.0
C2-S1-A1 cargo	*Hull 202*	Alcoa SS Co., Inc.	Moore Drydock Co.	0.0
C2-S1-A1 cargo	*Hull 203*	Alcoa SS Co., Inc.	Moore Drydock Co.	0.0
C3-S-A1 cargo	*Hull 10*	Isthmian SS Co.	Seattle-Tacoma Shipbldg. Corp.	0.0
C3-S-A1 cargo	*Hull 11*	Isthmian SS Co.	Seattle-Tacoma Shipbldg. Corp.	0.0
C2G cargo	*Hull 228*	Grace Line, Inc.	Federal Shipbldg. & Drydock Co.	15.6
Transport	*Hull 206*	U. S. Navy	Consolidated Steel Corp.	0.0
Transport	*Hull 207*	U. S. Navy	Consolidated Steel Corp.	0.0

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*Hull numbers are builder's numbers.

Ventilation of a Trichlorethylene Degreaser

By WILLIAM N. WITHERIDGE[†] and HERBERT T. WALWORTH[*]

High concentrations of trichlorethylene may exist in degreasing room atmospheres if there is no mechanical ventilation. Where exhaust ventilation is applied locally the problem arises as to the most effective way of designing the ventilating system. In this article, abstracted from "The Journal of Industrial Hygiene and Toxicology." the authors present the results of a series of experiments with four types of ventilating slots around the degreaser.

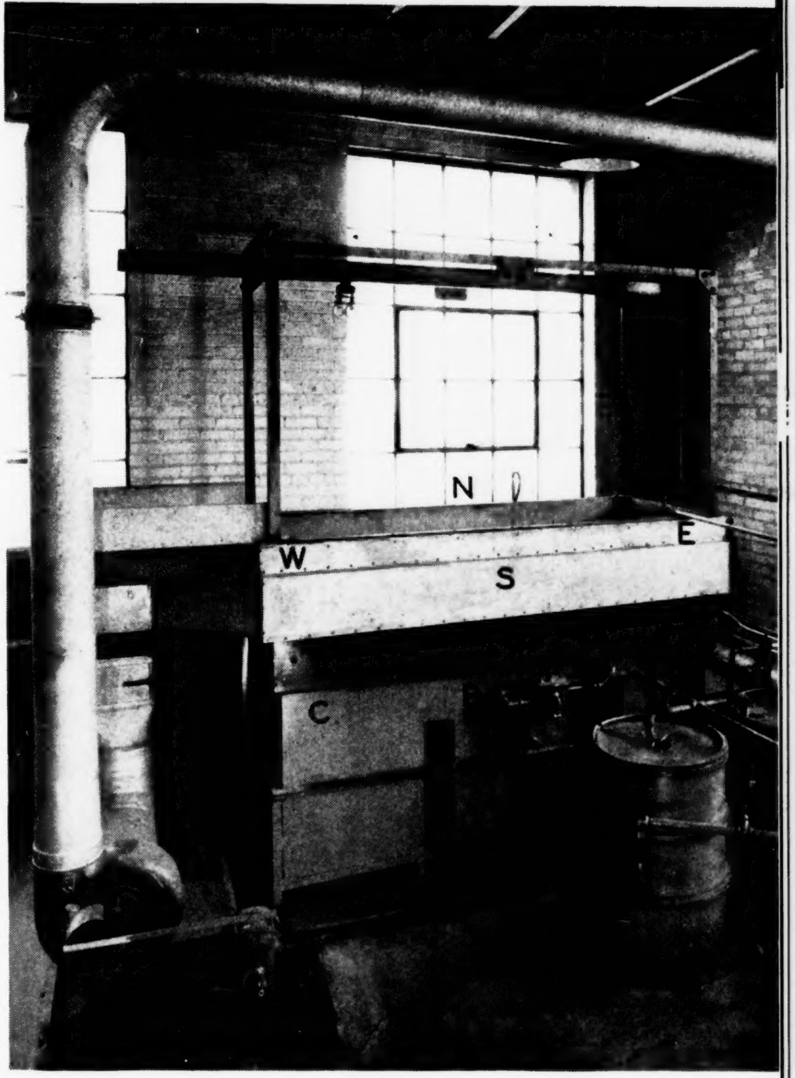

Fig. 1. Side view of a degreaser with platform removed. A, boiling chamber; B, rinsing chamber; C, vapor chamber; N, north side; E, east end; S, south Side; W, west end.

BECAUSE of limited information on the design of local ventilation for open tank degreasers, a study was conducted with four types of local exhaust systems on a degreasing tank 20 sq. ft. in area. The study was undertaken to determine for these exhaust systems the effect of ventilation rate, air velocity at the face of exhaust openings and location of exhaust openings on trichlorethylene exposure and loss.

Figs. 1 and 2 show the arrangement of degreaser and ventilating apparatus in the experimental room. The overall room dimensions were 18 ft. 9 in. wide, 38 ft. 7 in. long and 16 ft. 4 in. high, making a volume of approximately 12,000 cu. ft. The room was in the northeast corner of the building with its long dimension on the east side. The degreaser was located in the northeast corner of the room with its boiling chamber A toward the east and the vapor chamber C toward the west (Fig. 1). Hereafter the sides and ends of the tank will be known as "north, east, south and west."

The degreaser used in this study was an open type, three-dip or three-stage machine 72 in. high, 88 in. long and 33 in. wide, surrounded by a 28 in. movable platform. The normal distance from the top edge of the tank to the vapor level was 19 inches. Under operating conditions the solvent capacity of the boiling and rinsing chambers was 90 gallons each or a total of 180 gallons.

The four types of local exhaust ventilation applied to this tank are illustrated in Figs. 1 to 5. Figs. 1 and 2 show the "vertical slot" type, Fig. 3, the "horizontal slot" type and Fig. 4 the "elongated hole" type. The "round hole" type is indicated diagrammatically in Fig. 5, which shows the arrangement of the four systems in relation to the side of the degreasing tank.

In spite of instructions to operators concerning correct degreasing technic, cleaned parts often entrain hot solvent upon removal, which contributes to the solvent exposure. For this reason, the ventilating system incorporated an exhausted table or grille at the "take-out" end of the tank where cleaned parts could be placed for evaporation and removal of entrained solvent (Figs. 1, 2 and 3).

The centrifugal fan used on this study had a 12 in. diameter impeller connected by a V-belt drive to a 1 hp. motor on an adjustable base. The motor pulley was an adjustable type giving a wide range of fan speeds. The

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[†] Bureau of Industrial Hygiene of the Detroit Department of Health, Detroit, Mich.

[*] Division of Industrial Hygiene, Montana State Board of Health, Helena, Montana. Formerly with the Detroit Dept. of Health.

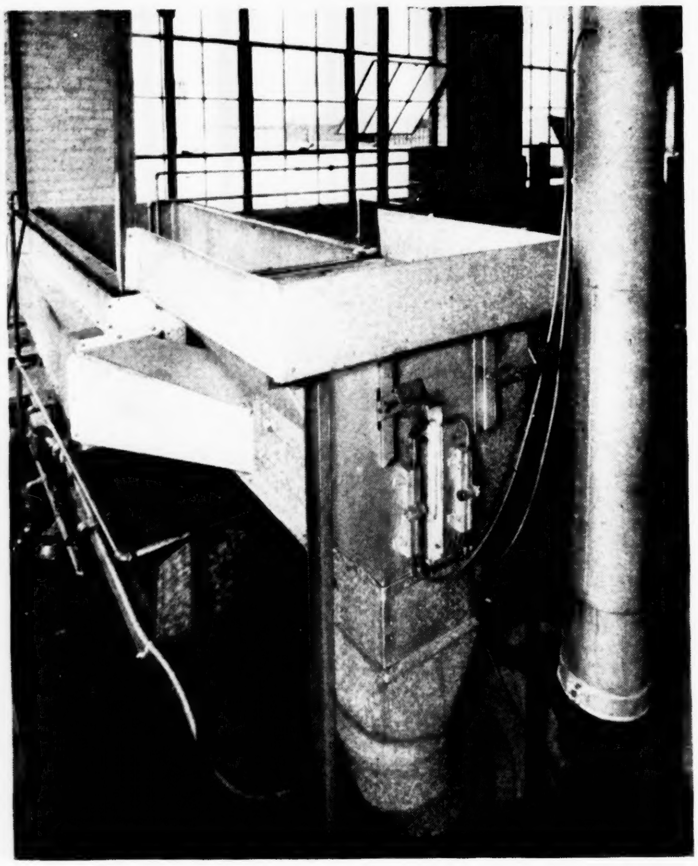


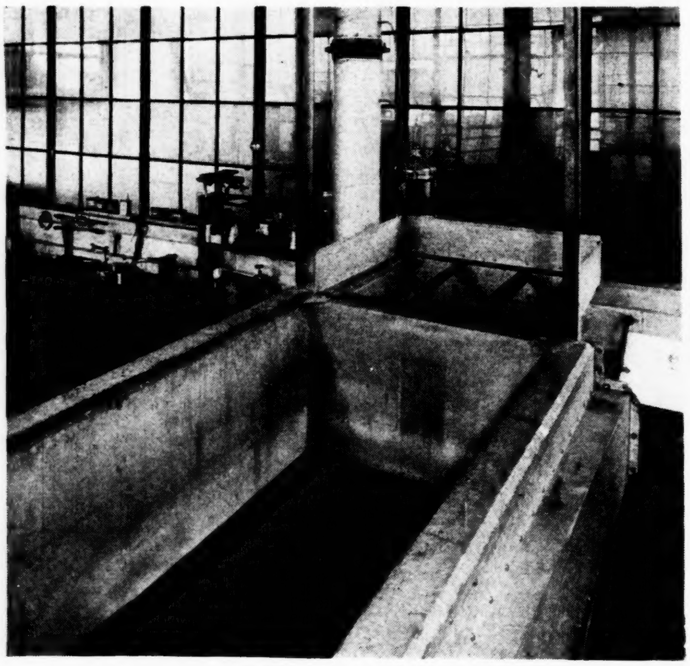
Fig. 2. West end of degreaser showing vertical slot exhaust and fan connections.

air discharge was through a 9 in. round exhaust duct terminating outside the building. A thin plate orifice was placed in this duct 6 feet from the fan and was provided with connections to a manometer for measuring total airflow through the system.

Procedure

The cycle of operation for this type of degreasing equipment is as follows: Immerse parts in boiling solvent in chamber A for 15 to 25 seconds or until free of oil or grease; transfer work slowly to rinsing chamber B for 5 to 10 seconds; transfer work slowly to chamber C for immersion in solvent vapor for about 15 seconds for final rinse by condensation on the cooled metal surfaces; remove work from this chamber slowly (manufacturer recommends a rate not exceeding 11 f.p.m.) so as to cause as little disturbance as possible to the solvent vapor level. By leaving parts in the vapor chamber for a few seconds, they become heated to such a temperature that most of the solvent will be evaporated upon their slow removal from the tank, thus preventing excessive "carry-out."

Fig. 3. View toward west or "take-out" end of degreaser showing horizontal slot exhaust and grille.



Trichlorethylene air samples were collected at a rate of 1 l.p.m. by a specially designed combustion apparatus modified from the original described by Tebbens. A total of 243 samples were collected and analyzed.

Air samples were collected at the mid-point of each side 13 inches above the edge and 6 inches toward the center of the degreaser. This point was predetermined to be the approximate operator's breathing zone for this tank with a 28 inch platform. Exhaust stack samples were collected from a hole in the exhaust duct about 36 inches from the fan.

This procedure was followed at tank edge ventilation rates of 615, 980 and 1400 c.f.m. for the vertical slot, 615 and 980 c.f.m. for the horizontal slot and 615 c.f.m. for the round and elongated hole systems. The volume of air through the grille was held constant at 330 c.f.m. in all cases except when the vertical slot was operated at 1400 c.f.m. in which case the grille was completely closed. The air velocity across the grille opening varied from 80 to 160 f.p.m. at the level where cleaned parts ordinarily would rest.

TABLE 1.—SUMMARY COMPARISON OF FOUR TYPES OF DEGREASER EXHAUST SYSTEMS

	No Ventilation	Vertical Slot	Horizontal Slot	Round Holes	Elongated Holes
Total Ventilation (c.f.m.)..	0	945	945	945	945
Airflow through Slots or Holes	0	615	615	615	615
Airflow through Grille	0	330	330	330	330
Air Velocities at Face of Slots or Holes—					
East End	0	220-240	275-300	1540-1630	510-550
West End	0	500-600	600-700	1580-1620	510-530
North Side	0	230-500	300-575	1550-2250	620-2000
South Side	0	250-475	300-575	1550-2250	620-2000
Trichlorethylene Concentrations (p.p.m.)—					
East End	229	193	47	80	161
West End	342	238	55	58	58
North Side	237	112	88	62	76
South Side	259	163	83	82	71
Averages	267	177	68	71	91
Static Pressure at Fan Intake (inches of water)..		0.15	0.15	0.91	0.77
Power Required without Orifice Meter in Exhaust Duct (air horsepower) ..		0.14	0.14	0.26	0.24

Results

Table 1 and Fig. 6 present a comparison of the performance of the four exhaust systems. With the horizontal slot and round hole systems, a total air flow of 615 c.f.m. through the 20 ft. perimeter of the tank proved adequate to reduce trichlorethylene exposures from a maximum of 400 p.p.m. to below 100 p.p.m. The elongated hole system was equally as effective as the horizontal slot and round hole types except at the east end, which was probably the result of poor air exhaust distribution around the tank. The vertical slot system was least effective in controlling solvent exposures from the standpoint of ventilation rate as it required 1400 c.f.m. to reduce solvent exposures to 100 p.p.m. The round and elongated hole systems required about twice as much power as the horizontal slot system at the same ventilation rate to reduce trichlorethylene concentrations to 100 p.p.m. or less.

The inside and outside types of manifold ductwork constructed for this study are not necessarily recommended as the best types for permanent installation since their design was influenced to a certain extent by convenience of interchangeability from one type of exhaust opening to another.

Table 2 shows solvent losses under various operating conditions as well as the amounts of solvent exhausted to the outside by the different ventilating systems at the lowest ventilation rate. The figures for solvent exhausted were computed from trichlorethylene concentrations in samples of air drawn from the exhaust duct. In general, the amounts of solvent exhausted were higher when cleaning parts than when no cleaning was done in the tank. In 2 cases of a total of 49 stack samples the amounts were higher when no parts were being cleaned.

TABLE 2.—SOLVENT REPLACEMENTS AND QUANTITIES OF SOLVENT EXHAUSTED WITH EACH TYPE OF EXHAUST SYSTEM

Amount of Ventilation C.F.M.			Type of Exhaust System	Solvent Exhausted*		Solvent Replacements†
Total	Grille	Tank Edge		Not Cleaning Parts	Cleaning Parts 100% of Time	
0	0	0	None	0	0	7.5
945	330	615	Vertical Slot	7.3	7.7	8.6
			Horizontal Slot	5.5	7.8	10.9
			Round Holes	9.9	18.2	11.4
			Elongated Holes	8.0	15.0	—
1310	330	980	Vertical Slot	—	—	9.0
			Horizontal Slot	—	—	10.9
1400	0	1400	Vertical Slot	—	—	8.6

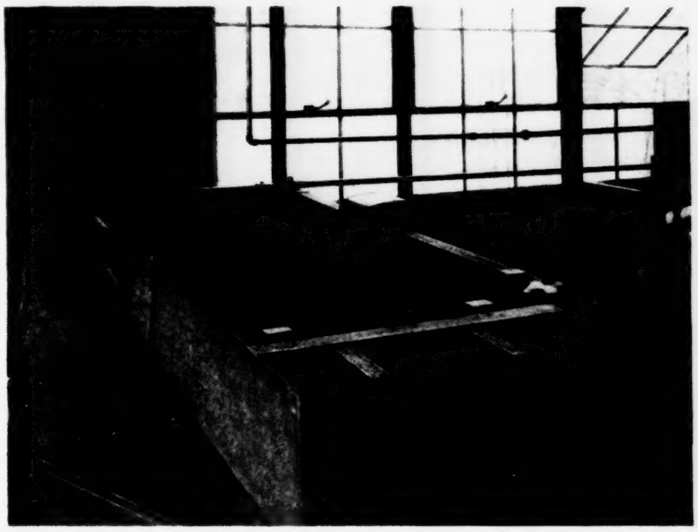
*In gallons per five 9-hr. days.

†In gallons per five 9-hr. days cleaning parts approximately 20% of time.

The actual solvent losses as determined by solvent replacements are given in the last column of Table 2. The vertical slot system did not remove as much solvent from the tank as the other exhaust types, but, as indicated in Table 1 and Fig. 6, it did not fulfill efficiently, from the standpoint of airflow required, the prime purpose of local exhaust ventilation, namely, the removal of solvent vapors from the worker's breathing zone. The round hole and elongated hole types, in addition to requiring more power, appeared also to exhaust more solvent than the vertical and horizontal slot types. The higher solvent losses observed with the round and elongated hole systems may have been due to the location of the holes 2 inches below the inside edge of the tank and thus 2 inches closer to the vapor level than the horizontal slot.

Solvent losses did not vary significantly with the ventilation rate, although it is possible that data collected over a longer period of time might have established such a relationship. The application of local exhaust ventilation to the degreaser used in this study increased solvent replacements an average of 0.5 gallon per 9-hour day when cleaning parts approximately 20% of the time. This is equivalent to an increase of 0.3 gallon per square foot of tank opening per 100 hours of operation. At full time operation, ventilation increased the solvent loss on this tank a maximum of 2 gallons

Fig. 4. View toward east end of degreaser showing elongated hole exhaust openings.



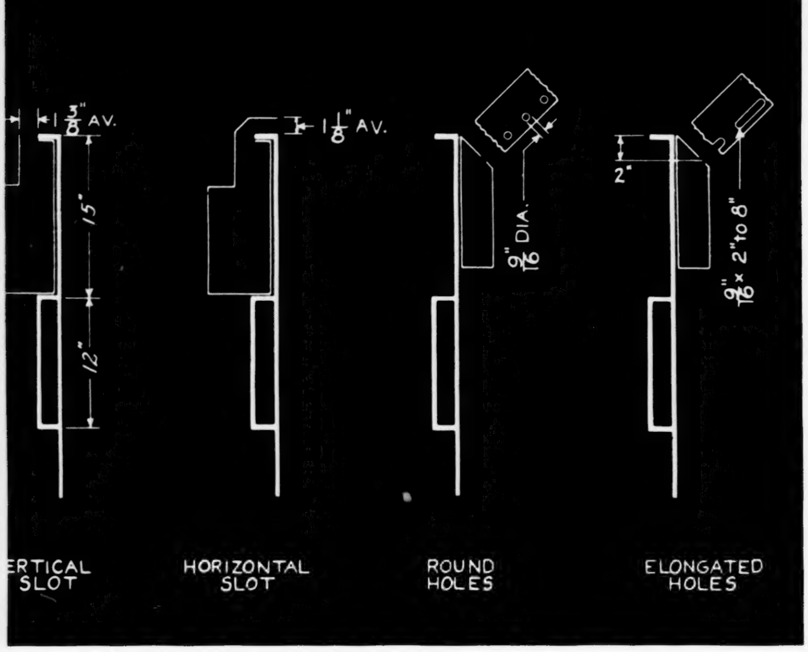


Fig. 5. Diagram of four types of exhaust openings at top edge of degreasing tank. Vertical and horizontal slot types attached outside of tank; other types inside.

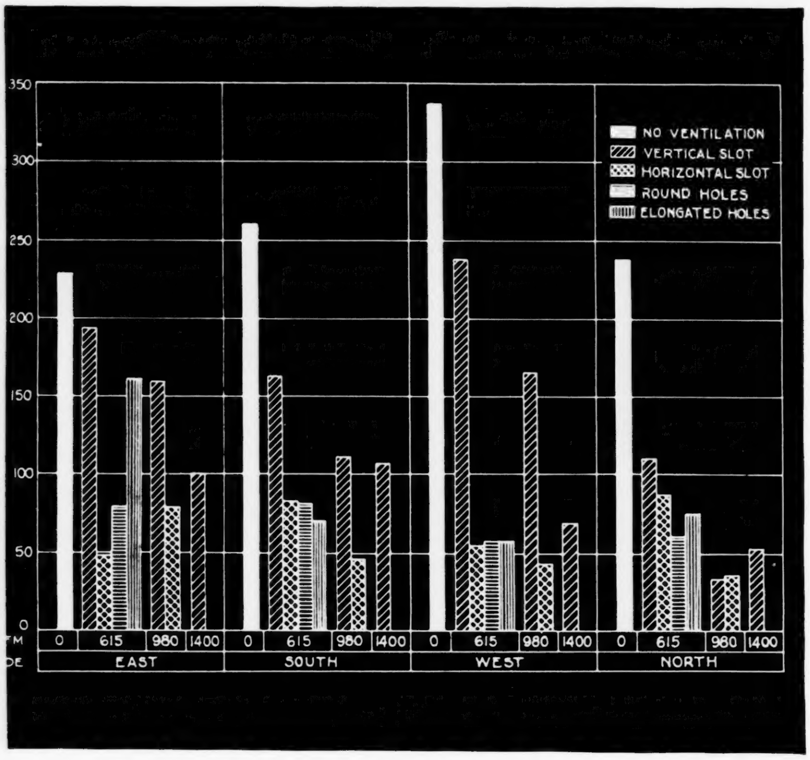
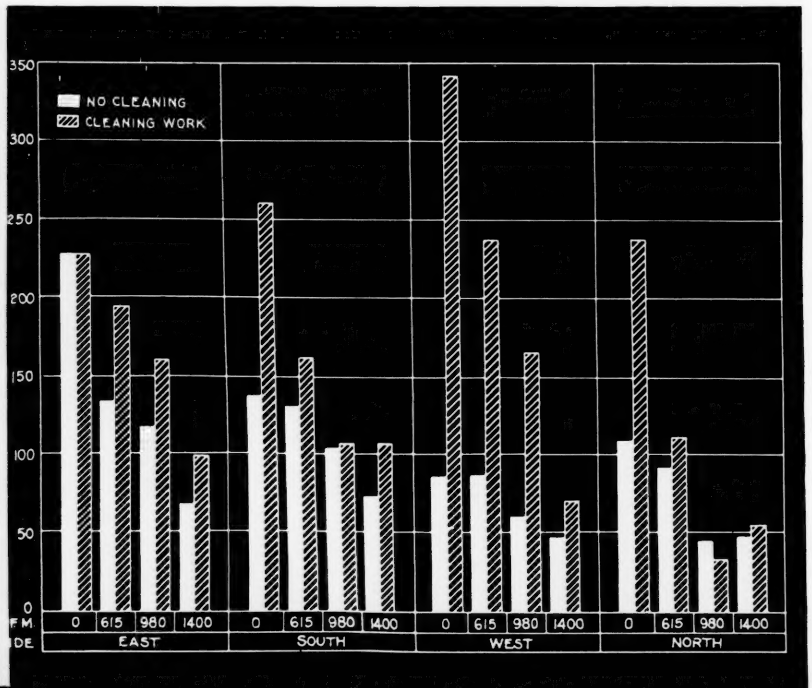


Fig. 6 (Above). Trichlorethylene exposures in operator's breathing zone with each type of exhaust opening while cleaning parts in degreaser.

Fig. 7 (Below). Average trichlorethylene concentrations in operator's breathing zone at each side of tank while cleaning and not cleaning, vertical slot ventilation.



per day or 1.2 gallons per square foot per 100 hours. These increases due to ventilation are small when compared with the increased solvent losses resulting from careless or improper operation of degreasers. A recent survey in New York found solvent losses in practice running from 0.3 to 15 gallons per square foot of tank area per 100 hours of operation, with a median value of 3.5 gallons.

The increase of trichlorethylene concentrations in the operator's breathing zone when parts were being degreased as compared with the concentrations when no work was being done is illustrated in Fig. 7 for the vertical type slot. In general, an increase in ventilation caused a corresponding decrease in trichlorethylene concentration in the operator's breathing zone. The highest trichlorethylene concentrations without ventilation were found over the east or boiling end of the degreaser when no parts were being cleaned, probably because of convectional disturbances. However, with ventilation during the cleaning cycle, trichlorethylene concentrations were highest at the west or "take-out" end except at the maximum ventilation rate.

It appears that a certain degree of non-uniformity of air exhaust distribution around the tank is not necessarily a disadvantage when the system is constructed so that the highest velocities occur at the "take-out" end of the degreaser. This is illustrated in Fig. 8 for the vertical slot system. With no ventilation, the solvent exposures during degreasing were highest at the west end. As ventilation was applied the trichlorethylene concentrations over this end were gradually reduced to the range of concentrations over the other three sides until, at the highest ventilation rate, they were actually lower than those at the east end.

The horizontal slot was not operated at the highest ventilation rate, since the low and intermediate rates reduced the exposures to below 100 p.p.m. (Fig. 9).

Fig. 10 shows a set of contours and streamlines for the horizontal slot. The velocity contours were substantially the same for the four types of exhaust openings at 6 inches or more from the face of the openings and at the same ventilation rates (c.f.m. per foot of tank perimeter). Streamlines indicated that the vertical slot type drew the smallest proportion of air from the space directly above the degreaser, probably because of its position at the outer edge of the tank.

When air currents directly over the tank from the level of 6 inches below to 16 inches above the edge were 30 to 35 f.p.m. (the usual range), the upward convection current velocities at the outer edge of the tank caused by heat dissipated by the degreaser were 50 to 75 f.p.m. At no time during the operation of the degreaser either with or without ventilation were air currents observed to "roll over" the edge of the tank and fall toward the floor, so far as could be determined by smoke tests.

Frequently the statement is made that ventilation of trichlorethylene degreasers should be accomplished at the edge of the tank because trichlorethylene vapors are heavier than air and naturally fall to the floor. However, since the momentary trichlorethylene concentrations above the edge of the tank probably do not exceed 1% or 10,000 p.p.m. under the most unfavorable operating conditions, and since this concentration would have little effect on the resultant specific gravity of the air-vapor mixture, aerodynamically the mixture would behave practically the same as pure air at the same temperature. Furthermore, updrafts around the tank work directly against any theoretical tendency of vapors to fall over the edge. Finally, the marked inferiority of the vertical slot type of ventilation in the reduction of solvent exposure as compared with the other three types studied seems to indicate that gravity is of little or no assistance in capturing the warm mixture of air and trichlorethylene vapor. Nevertheless, ventilation should be applied at the perimeter of the tank for another reason. In that position it will not draw vapors across the operator's breathing zone, as would be the case with overhead or "updraft" ventilation.

Summary

1. Four types of local exhaust systems for a large open type degreaser were studied to determine their effectiveness in controlling trichlorethylene exposures. Air samples were collected in the operator's breathing zone and solvent losses were observed at different ventilating rates.

2. Of the four ventilating systems used in this study, the horizontal slot type proved most efficient in reducing solvent exposures, from the standpoint of power requirement.

3. Vertical slot ventilation caused the lowest solvent loss increase but was least efficient in reducing solvent exposures.

4. The round hole system caused the highest solvent loss increase and required approximately twice as much power as the horizontal slot type to reduce solvent exposures to 100 p.p.m.

5. The elongated hole system failed to reduce solvent exposures sufficiently at the boiling chamber end of the tank, probably because of poor air exhaust distribution. Otherwise, its performance was similar to that of the round hole system.

6. The maximum solvent loss increase due to ventilation during continuous use of this degreaser was 2 gallons per 9-hour day or 1.2 gallons per square foot of tank area per 100 hours' operation. No significant relation between ventilation rate and solvent loss was observed.

7. At no time were trichlorethylene vapors found to "roll over" the edge of the degreaser during operation, either with or without ventilation.

8. A ventilation rate of 615 c.f.m. around the 20 ft. perimeter of the tank using the horizontal slot or round hole system reduced the average trichlorethylene concentrations in the operator's breathing zone from a maximum of 400 p.p.m. without ventilation to below 100 p.p.m. The vertical slot system required a ventilation rate of 1400 c.f.m. to accomplish the same result.

Acknowledgment is made to the Detroit Rex Products Company for its assistance and courtesies and for the use of photographs appearing in this report, and to Melvin W. First of the Bureau of Industrial Hygiene for his assistance.

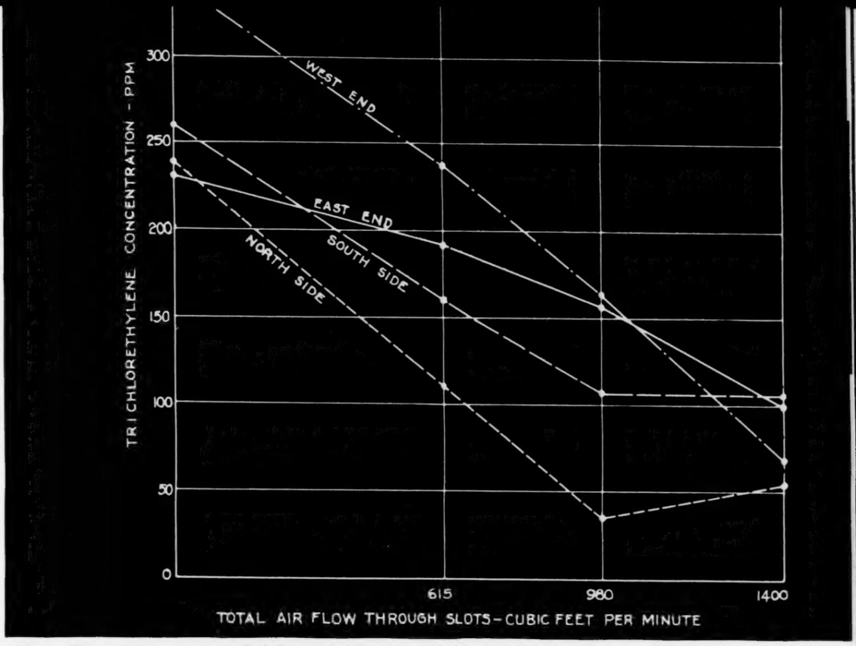


Fig. 8. Average trichlorethylene concentrations in operator's breathing zone with vertical slot ventilation while cleaning parts in degreaser.

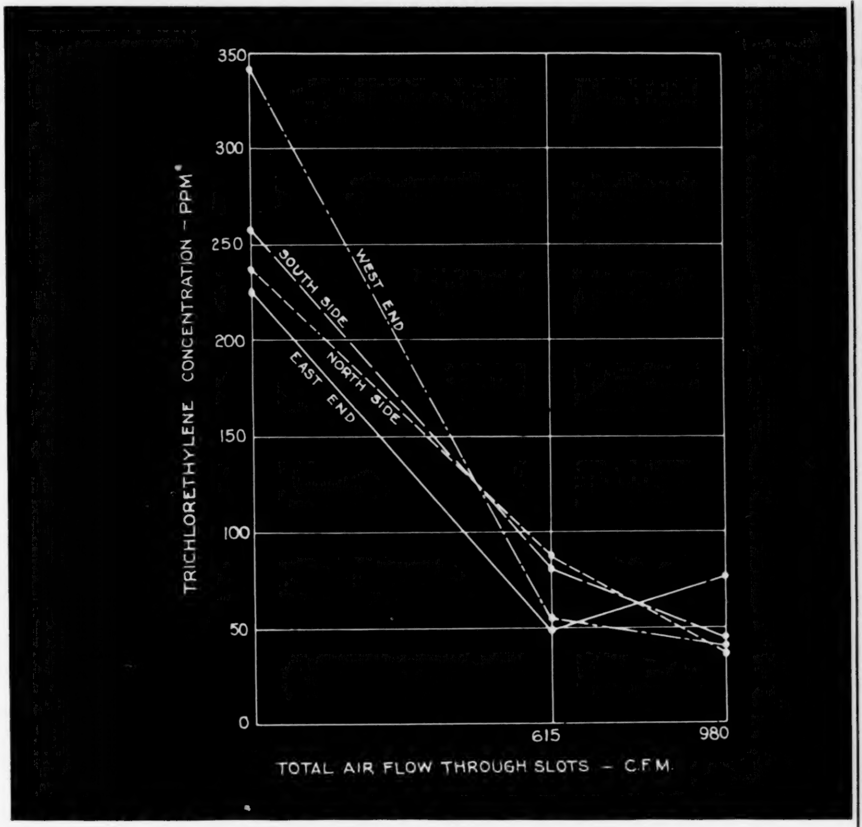
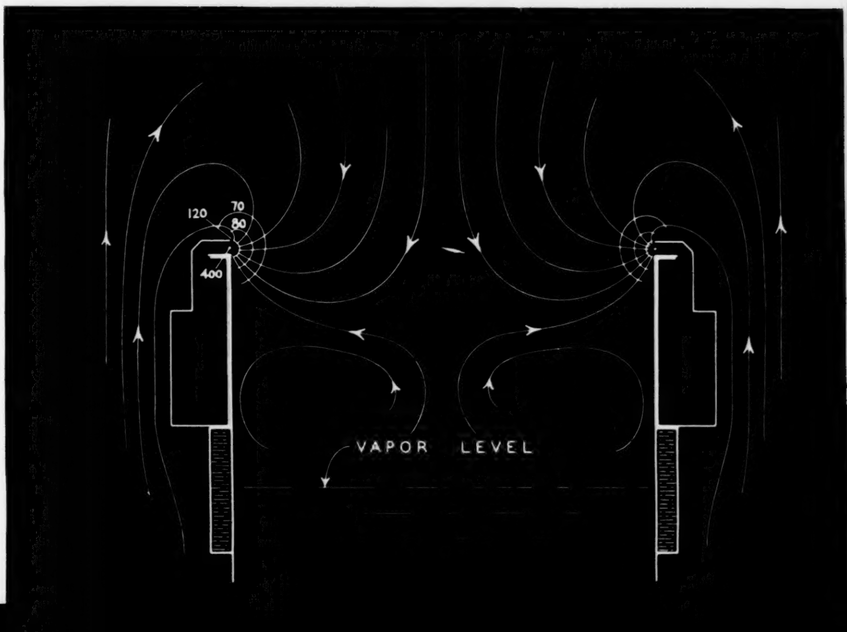


Fig. 9 (Above). Average trichlorethylene concentrations in operator's breathing zone with horizontal slot ventilation while cleaning parts in degreaser.

Fig. 10 (Below). Velocity contours and streamlines with horizontal slot ventilation. Figures represent air velocities in feet per minute at 0, 1, 2 and 3 in. from slot. Measurements for this set of contours were made at midpoint of sides with a total slot ventilation rate cf 615 c f.m.



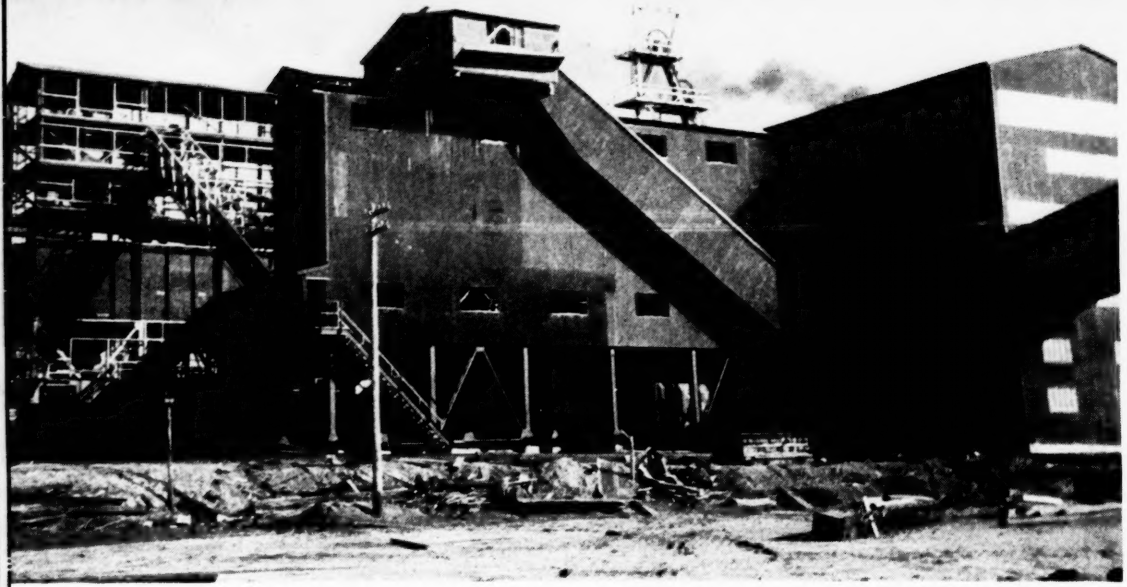


Fig. 1. Mechanical Preparation Plant of the Peabody Coal Company at Westville, Ill., during construction. Plant has capacity of 600 tons of coal per hour.

PEABODY COAL COMPANY

uses industrial unit heaters to solve a difficult heating problem in its coal preparation plant at Westville, Illinois

By D. A. WHITING[†]

HEATING a coal preparation plant is just about as difficult a job as heating a circus tent. These plants are usually constructed of corrugated sheet iron which offer very little resistance to the flow of heat and to make the problem more difficult, there are numerous conveyor openings in the walls. To design a heating system for such a plant requires considerable experience not only in estimating heat losses but also in sizing and placing the heating equipment at proper levels and tempering vast quantities of incoming outside air entering through conveyor openings. Fortunately for the heating engineer large amounts of steam are available in most cases and fortunately for the plant owner fuel cost is secondary.

A good example of a successful heating installation in a building of this type is the system in the Westville, Illinois, Mine No. 24, Mechanical Preparation Plant of the Peabody Coal Company. This plant is located near the town of Catlin in the Danville district of Illinois. The rated capacity is 600 tons of coal per hour and there are five railroad loading tracks as well as provisions for trucks. The plant is one of the most modern of its kind and all equipment is operated from one master control board. The board can be arranged so that all motors can be started automatically in the proper sequence by merely pressing one button, and operator is advised of response of every motor by means of signal light under each control on board.

Heating

Besides meeting the needs of comfort, heating is necessary to prevent the water used in the coal washing operations from freezing. The plant is equipped for washing all of the coal to remove both the impurities and the dust and sizing of coal. The larger sizes of coal—6 in. and over—are hand-picked, but for the smaller sizes, hand-picking is normally inefficient and unprofitable, thus making washing necessary. Present day consumers also insist on dust free coal in the bin both for hand firing and stoker use.

Heat Loss Estimation

A glance at the photograph of the exterior of the plant will give some idea of the difficulties involved in estimating the heating requirements of a building such as this. It will be noticed that the plant is made up of a series of buildings joined together by enclosed conveyor-ways. In many instances the building floors are exposed while in other instances walls are not exposed to out-of-doors, but to unheated inside areas. It will also be noted that both walls and roofs are made of corrugated iron sheeting and that there are a number of steel sash windows. Since heating is needed only where there are workmen or where water is used in the process, not all of the plant is heated; especially the dry coal section. Fig. 3 shows the areas in which heating equipment is installed.

Since no definite information was available on heat loss in this type of construction, it was first necessary to determine the unit heat loss. An examination of the siding on this and similar structures showed that the corrugated siding was 2 ft. wide and the joints were filled with a plastic material. After the plant is in operation for some time, vibration causes the plastic material to loosen and frequently fall out of the joints

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[†] Thomas Conlin Company, Chicago, Heating & Power Piping Contractors.

leaving about one-third of the vertical joint as an open crack. The examination also showed that in most plants of this type the sash area was about 15% of the wall area.

In order to obtain a figure for the heat loss per square foot of outside wall, a typical 100 sq. ft. section was taken. This typical section had 85 sq. ft. of corrugated iron area, 15 sq. ft. of steel sash area, about 24 lineal ft. of 1/16 in. crack around the sash and 80 lineal ft. of open 1/16 in. joint in the siding. The heat loss through this 100 sq. ft. section was:

Corrugated Iron Wall,
85 sq. ft. × 1.50 B.t.u./sq. ft./hr./F = 128
Factory Sash,
15 sq. ft. × 1.10 B.t.u./sq. ft./ hr./F = 17
Factory Sash Infiltration,
24 lin. ft. × 0.9 B.t.u./lin. ft./hr./F = 22
Siding Joint Infiltration,
80 lin. ft. × 0.9 B.t.u./lin. ft./hr./F = 72

B.t.u. per sq. ft. per hr. per deg. F = 239

The heat loss per square foot was:
239 ÷ 100 = 2.39 B.t.u. per sq. ft. per hr. per deg. F. For safety, a factor of 2.5 B.t.u. per sq. ft. per hr. per deg. F was used.

For walls between heated and unheated spaces this figure was decreased to 2.0 B.t.u. per sq. ft. per hr. per deg. F because of negative wind conditions.

Conveyor Openings

In numerous places conveyors pass through the walls of the heated spaces. Since it is impossible to close these openings an estimate had to be made of the heat loss by infiltration through these points. The average conveyor opening is approximately 25 sq. ft. in area and was estimated to have 25% of its area open. In other words, the clear space per conveyor opening is 25% of 25 sq. ft. or 6.25 sq. ft. Assuming a 25-mile per hour wind the infiltration through the opening is
25 mile/hr. = 2200 ft. per min. × 6.25 sq. ft. = 13,750 c.f.m. total infiltration

or the infiltration per square foot is $\frac{13,750}{25}$ = 550 c.f.m. per sq. ft.

Heat loss per square foot of conveyor opening is
550 c.f.m. × 0.02 B.t.u. per cu. ft. = 11 B.t.u./sq. ft./per hr./F.

For interior walls between heated and unheated spaces, a two-mile per hour air velocity is used
2-mile/hr. = 176 ft. per min. × 6.25 sq. ft. = 1100 c.f.m. infiltration.

The heat loss per square foot of conveyor opening is therefore 1100 × .02 ÷ 25 = 0.88 B.t.u./sq. ft./F. Using these heat transmission figures, a heat loss estimate was prepared (see Table 1).

TABLE 1—HEAT LOSS OF COAL PREPARATION PLANT

CONSTRUCTION	SCALE HOUSE B.T.U./-HR./DEG.	WET SCREENS AND PICKING TABLE B.T.U./-HR./DEG.	CONE AND WASHERY B.T.U./-HR./DEG.	DRYER CONVEYOR B.T.U./-HR./DEG.
Roof	384	2881	6000	7750
Wall-Outside	1056	12,828	3715	22,188
Inside	None	None	5820	5600
Floor-Outside	60	498	549	None
Inside	None	None	None	1848
Opening-Outside	231	None	1584	1100
Inside	None	None	239	None
Air Changes (per hour)	5	2	12.5	5
Contents, Cu. Ft.	2552	67,235	25,000	141,900
Exposure, Direction	N-E-S-W	N-S-W	E	E-S-W
Total Heat Loss B.t.u./hr. (60F T.D.)	103,860	1,029,060	1,074,220	2,309,160
Steam, Lb./hr.	108*	1100*	1119*	2405*

*Lb. per hr.

It will be noted that there were four separate areas to be heated. The volumes varied from 2552 cu. ft. to 141,000 cu. ft. The latter figure is for the cone and washery section; the most important portion of plant

Fig. 2. Another construction photograph of the Coal Preparation Plant showing the large washing cone at the right. Heating a plant such as this requires a considerable amount of experience in estimating heat loss and in sizing and placing the heating equipment.

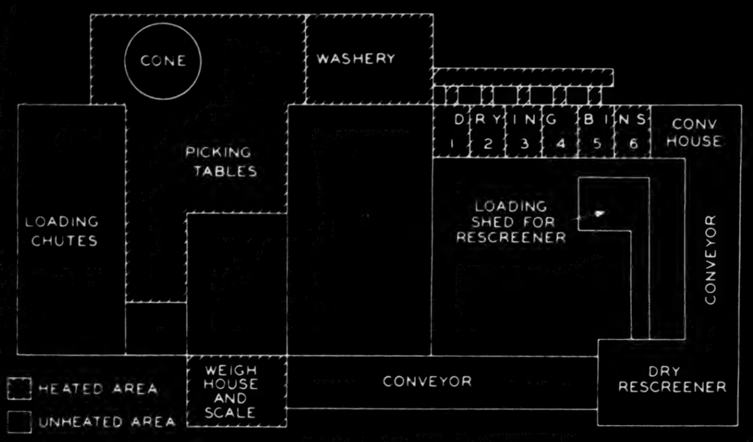


Fig. 3 (Left). Plan of the Coal Preparation Plant showing the location of the heated areas.

with reference to satisfactory and adequate heating. Here, the wash boxes and wet shaker screens are located, also the cone, and enormous cone shaped steel storage tank, which reclaims most of water used in washing by means of settling and blowing off sludge accumulated in the process.

Heating Equipment

In selecting the proper type of heating equipment for a building of this type many points had to be kept in mind. Because the infiltration was necessarily high and because it was important that certain areas be kept warm, it was necessary to select equipment which could deliver large amounts of heat and sufficient air turnovers in a restricted area. The equipment had to be rugged and because of the large amount of coal dust in the air had to be easily cleaned so as to prevent a large falling off of capacity due to poor heat transfer.

One type of equipment which met these conditions was the industrial unit heater. This was the type selected. The units used are of the propeller fan type with horizontal discharge and operate on a pressure of 75 lb. per sq. in. They are placed so as to blow directly into the area to be heated. Coils and finned surfaces are of special design to provide low outlet temperatures with high pressure steam and wide fin spacing for large air quantities and minimizing of coal dust settlement.

UNIT HEATER SCHEDULE

Location in Plant	Heat Loss B.t.u. Per Hr	Vol. Cu. Ft.	No. of Heaters Installed	Cap. Each B.t.u.	C.f.m. Each
Scale House	103,860	2552	1	131,040	1336
Wet Screens and Picking Tables.	1,074,220	25,000	2	346,207	3620
			1	432,975	4180
Cone and Washery	2,309,160	141,900	7	346,207	3620
Drying Bins and Wet Coal Conveyor	1,029,060	67,235	3	346,207	3620

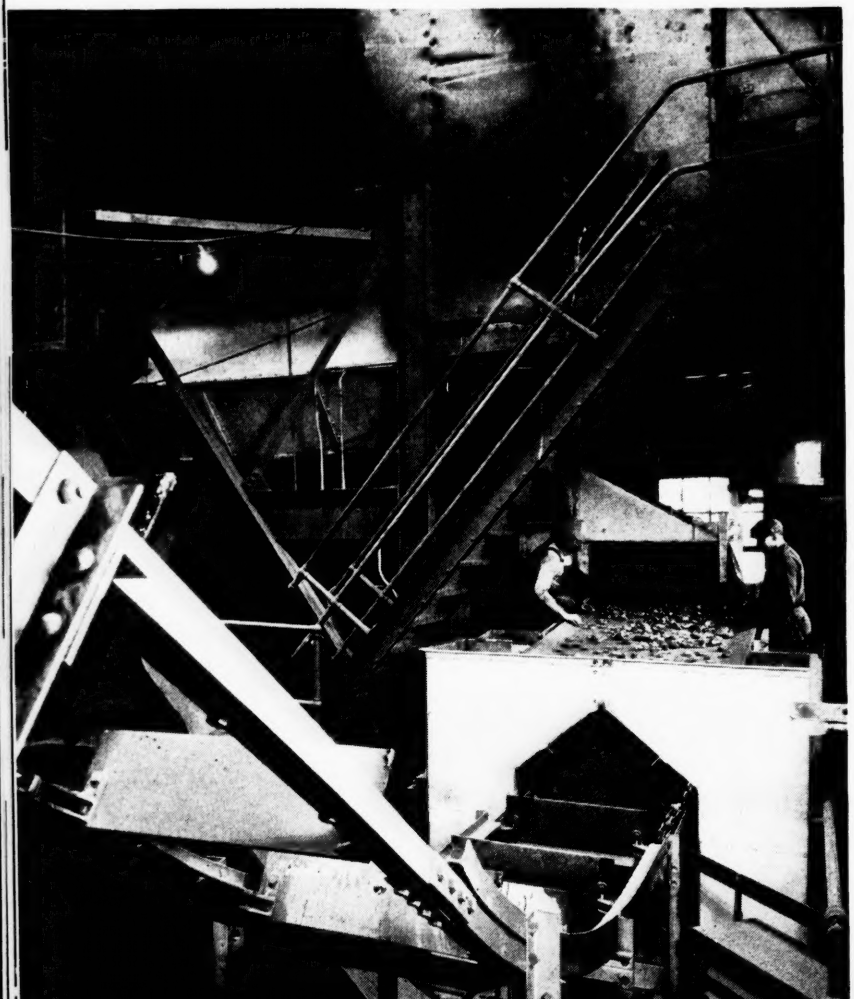
Special provisions had to be made for cleaning the heating units since coal dust builds up very rapidly on the heating elements. A valved horsepower steam hose connection was provided on the main to each unit heater. About every two hours the dust is blown from unit heater coils with high pressure steam from the supply main by means of steam hose with flared metal jet. This simple arrangement makes it possible to maintain a high efficiency of the unit heaters with a minimum amount of trouble.

The portions of plant that are heated are maintained automatically by thermostat, to a temperature of 50F with outside temperature at —10F below zero.

As prepared coal finished its cycle of washing and grading the moisture content is approximately 15%, which is reduced to 3 or 4% when loaded. This is accomplished in the numerous large drying bins by means of specially designed heating coils with cover jackets which allow high temperature air from coils to pass through body of stored coal.

Fig. 4 (Above). Construction photograph showing the railroad carloading section.

Fig. 5 (Below). Coal picking section of a typical coal preparation plant showing (at left) the type of unit heaters used.



Economical Dust Control with Low Resistance Exhaust Systems

By F. F. KRAVATH†

PART 5 — SUMMARY — A SAMPLE SYSTEM

Summary

NOW that all the elements have been separately studied, let us combine them into a sample system and see the overall savings in resistance possible, with the corresponding saving in horsepower. Fig. 10 represents a sample system containing all the elements discussed, and constructed according to the recommendations of this article. Here is shown a hood over a vat or pot giving off non-inflammable vapors and located at the tail end of the system, also served being two rows of grinders. Due to the nature of the material being conveyed, a velocity of 5000 ft. per min. is maintained up to the cyclone which removes the heavier particles. From thence, a velocity of 4000 ft. per min. is maintained up to the filter, excluding cyclone outlet and filter inlet fittings. Since no material is being conveyed, the velocity from filter to fan is kept at 3000 ft. per min., while from fan to system outlet the velocity is gradually reduced, through the evase, to 1500 ft. per min. All elbows in the system have been designed for the minimum resistance loss, the most efficient entry pieces have been used, the cyclone outlet is the efficient Moss nozzle, the filter inlet and outlets are the minimum resistance 7-degree diverging and bell mouth transformers, while the stack is a 7-degree evase. The fan used in this example is a backward curved blade, high efficiency, non-overloading type of exhauster, properly sized and selected.

The result, as shown in Table 2, is an overall resistance loss, including velocity pressure lost at system outlet of only 4.7 inches water gauge. Assuming the system had been designed and constructed as the majority are, without too much attention to the details of efficient airflow, the comparative element resistances might very well be as shown in Table 2 under *Old*. This is not an exaggeration, but merely a typical example of what is very widespread. Thus, it can be seen that if the system handled 15,000 c.f.m., which is not overly large, the comparative horsepower necessary to maintain flow would be 77 with the old layout, and 17 with the design illustrated. What this would mean in dollars and cents can be readily computed assuming plant operation of 250 days a year at 8 hours a day and a normal cost of about 2 cents per kilowatt-hour. Thus

TABLE 2—ELEMENT RESISTANCES IN INCHES WATER GAUGE CORRESPONDING TO SYSTEM SHOWN IN FIG. 10

ITEM	OLD	NEW
Entry Loss	0.71	0.04
Elbows	1.42	1.17
Pipe Junctions	0.25	0.10
Cyclone	1.75	1.05
Cyclone Exit	1.30	0.04
Filter Inlet	1.50	0.35
Filter	2.25	0.75
Filter Exit	0.71	0.04
Discharge Stack	1.60	0.10
Piping	1.74	1.06
TOTAL	13.23	4.70

$$\text{B.H.P.} = \frac{\text{CFM} \times \text{TP}}{\text{Mechanical Efficiency of Fan} \times 6356}$$

$$\underset{\text{old}}{\text{B.H.P.}} = \frac{15000 \times 13.23}{0.42 \times 6356} = 77$$

$$\underset{\text{new}}{\text{B.H.P.}} = \frac{15000 \times 4.7}{0.72 \times 6356} = 17$$

$$\text{Yearly Saving} = 250 \times 8 \times 0.02 \times (77 - 17) \times 0.75 = \$1{,}800$$

While it is quite possible that even a majority of systems in use today might not commit as many sins as the old system we have described, and hence, the savings to be expected might be reduced for this reason, it is very probable that this is more than outweighed by the fact that many plants have systems totalling 200,000 or more c.f.m. and operating for 16 or more hours.

Thus, it is quite important that systems be designed as efficiently as possible, and since systems are only made up of the elements shown and described, it is essential that the designer be acquainted with the flow characteristics of all these elements.

[*In the concluding installment next month Mr. Kravath will take up the matter of plenum chamber systems.*]

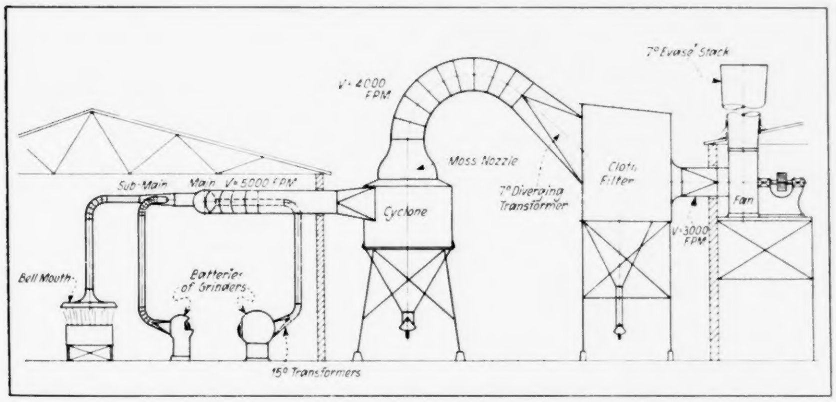


Fig. 10. Simplified drawing showing the general arrangement of a sample system.

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†Office Engineering Division, The Panama Canal.

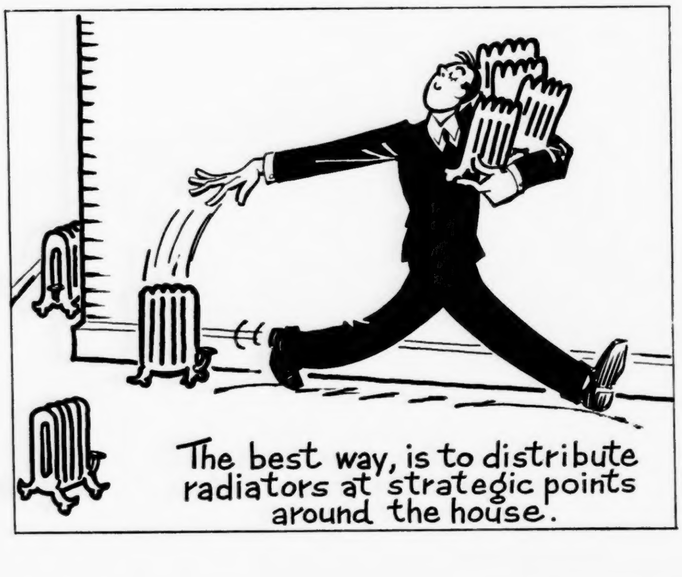
Theory vs. Practice

Exposure Factors

By T. W. REYNOLDS

FOR many years it was customary for engineers to add an arbitrary percentage of the heat losses to the estimated heat losses in order to allow for exposure, and in time the amount of this percentage varied with the user as did also the method of applying the percentage. For example, the writer some years back added 20% on the North, 15% on the West, 10% on the East and 0% on the South; where a room had two exposures with extent of glass and wall in near proportion he would average the two exposure factors, so that a northeast room would have 15% added. If the two walls and their glass were decidedly unlike in extent of wall and glass area he would figure each wall separately, add in its respective allowance for exposure and then proportion the radiators on each wall accordingly.

It has since been pointed out and is now more generally recognized that the coefficient of heat losses through heat losing construction of all types, as well as the infiltration figures for window cracks, are all based on tests as made with a wind velocity of 15 miles per hour. As a consequence, there is no longer any apparent justification for the use of exposure factors except for unusual conditions. Nevertheless their use is still prevalent among many engineers, possibly as a safety factor or as an allowance for the errors of omission and the unforseen factors which are commonly classified under the term of engineering contingencies. In reality the exposure factor is merely an unknown factor to allow for the unknown. But since no one knows what unknowns are being allowed for or what the size of this factor should be, the exposure factor might better be called a factor of ignorance until such time as we know from actual test data just what, if anything, this factor should be. In the meantime, this factor can at least be eliminated from calculations involving boiler or radiator size when checking an existing installation. Where an exposure factor is used as such, it should properly not include any addition to the roof losses of buildings with flat roofs of considerable area for such a roof has no exposure and therefore recognizes no cardinal points of the compass.



The best way, is to distribute radiators at strategic points around the house.

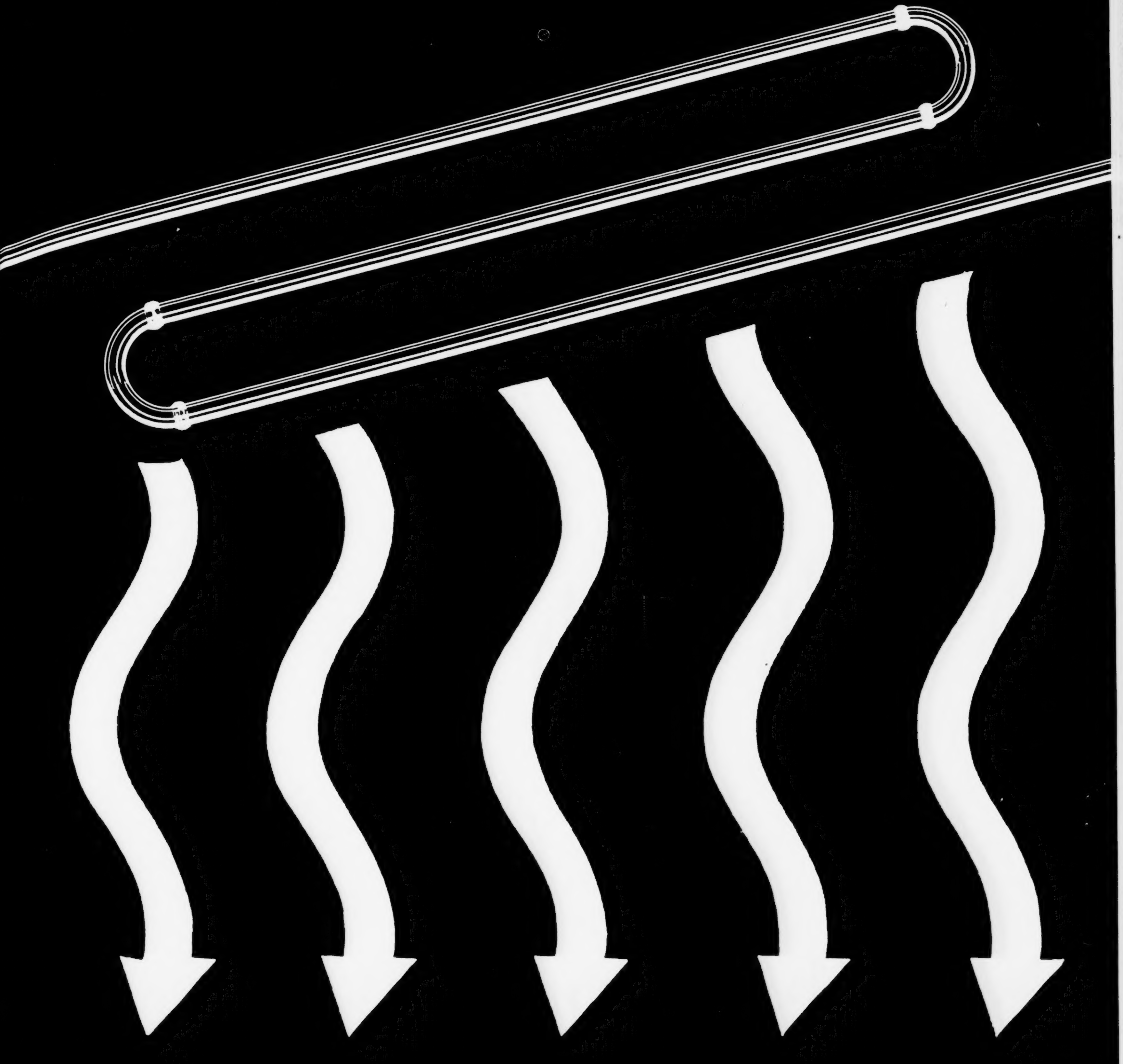
With the greater use of insulation and weather-stripping there seems to be but very little excuse left for exposure factors as the effect of exposure on materials of low heat transmission and cracks of diminishing size is much less noticeable. Similarly, both the heat transmission of radiators and of building materials is now so well known that the variables or unknowns of our heating problems have been reduced to such minimums as to make unnecessary any allowance whatsoever in the way of a safety factor. Gone also are the uncertainties of the time interval on pick-up due to more efficient methods of air removal and the use of light weight radiators and pipes.

In the final analysis, modern heating equipment has incorporated within itself certain safety factors, for example, the commercial capacities of unit heaters seldom fit the calculations precisely so that the next size of heater is selected.

Heating calculations are only for use with spaces whose walls are all or in part exposed to outdoors or to some space at a lower temperature. Thus an inside space requires no radiators in point of theory, as even any air change in this space would be replaced by warm air from the surrounding spaces. Practically, this is not always so, for it has been found that office buildings of large area which are left unheated or partially so over week-ends are cold-air bound in their central core upon the start of business each week, a condition which continues until the cold air is slowly displaced by warm air currents from the radiators on the outer walls. In the meantime, the ability of these radiators on pick-up to quickly heat the outer spaces is handicapped, for they are not usually sized to take the interior warm air load. This, of course, could be arranged for, but the best answer to the problem is to distribute radiators at strategic points throughout the interior.

An inside foyer, with doors opening to a heated entrance hall, is another example of an inside space which sometimes requires heat, due to the likelihood of both outer and inner doors being opened at the same time, and particularly so if the outside door is left open for some time.

RADIANT HEATING



A comprehensive progress report on low-temperature radiant heating for comfort, featuring practical data on design. This is No. 1 of a series of HEATING AND VENTILATING'S Reference Sections.

RADIANT HEATING IN THE U. S.

By JOSEPH F. KERN, Jr.[†]

The purpose of this section is to present practical and easily usable information on the design and construction of radiant heating systems; to summarize current practice in radiant heating by describing the outstanding features of each of all known U.S. installations, and to make available information on the current radiant heating situation in England.

The usual heating system, whether it is of the warm air or radiator type, regulates the heat loss of the human body by controlling the room air temperature. When the body is cold and is consequently losing too much heat the air temperature is raised which, in turn, decreases the convected heat loss of the body. Thus, this type of system controls the heat loss of the body primarily by changing the rate of heat lost by convection.

A radiant heating system, on the other hand, regulates the heat loss of the body by controlling the room surface temperature; that is, the average temperature of the walls, ceiling and floor. When the body is cold, the room surface temperature is raised, which decreases the radiant heat loss of the body. Thus a radiant heating system controls the heat loss of the body primarily by changing the rate of heat lost by radiation.

There are three broad classes of radiant heating systems in use—high, medium and low temperature systems. Some high temperature systems use gas or electric radiant panels which operate at temperatures so high that they are luminous. Others use electric panels which are non-luminous but have temperatures of about 500F. Medium temperature panels with surface temperatures of 140 to 300F use electricity, steam or hot water. Low temperature panels, by far the most common, have surface temperatures below 140F. The panel type use either electricity or hot water, the pipe coil type employ hot water or steam, the air duct type use warm air and an electric wire wallpaper uses electricity. This latter type is made (Gr. Br.) of a type of wallpaper with wire imbedded in it.

The most common type in the United States is the pipe coil panel or warm air surface. Because of this, in the United States a radiant heating system is generally understood to be one in which comparatively large areas of the floor, wall or ceiling are heated by hot water or steam coils or warm air ducts.

The temperatures of the heated surfaces are generally kept quite low, floor temperatures 85F or below, wall temperatures slightly higher and ceiling temperatures 125F or below.

It should be pointed out that the heat radiations from the heated panel are quickly absorbed by the room surfaces. The warmed surface and the heated panel itself heat the air so that the air temperature in radiant heated rooms is usually only a few degrees below the mean wall temperature.

COMFORT AND RADIANT HEATING

The purpose of any heating system is not to supply heat to the occupants but to control the rate of heat loss from the human body so that it just equals the rate of heat production. The human body at all times is generating heat and if the body temperature is to be kept constant at 98.6F, the rate at which the heat is lost must approximately equal the rate at which heat is produced. The problem of regulating the heat loss is complicated by the fact that heat production varies considerably with the degree of activity, the age and physical condition of the body.

It should be noted that the heat loss of the body does not have to be exactly the same as the heat production, since the body has the ability to store considerable amounts of heat without greatly affecting the feeling of comfort, and the body can draw on this reserve during periods when it is losing heat at a rate faster than it is producing it. However, it is generally conceded that the most comfortable conditions are those which make it possible for the body to lose heat at the rate it is being produced.

In recent experiments at the John B. Pierce Laboratory of Hygiene the effects of various air and wall temperatures on the clothed body were studied.

The investigators concluded that the primary factor controlling a person's feeling of comfort was the skin temperature. Skin temperatures of from 90 to 94F were found to be the most comfortable. This seems to be true regardless of the amount of clothing or activity of the person. The primary effect of clothing is that it makes possible a feeling of comfort at lower temperatures than could ordinarily be endured.

As a result of many experiments the investigators at Pierce Laboratory determined the relationship between wall and air temperatures as they affect the skin temperature and have coined a new term called "operative temperature" which takes into account the relative effect of radiation and convection on the heat loss of the human body. Operative temperature does not take into account the effect of humidity since the investigators found that relative humidities up to nearly 80% in the comfort zone had little effect on the heat loss and comfort.

Comfort Chart

Based on this relationship, the author has plotted a chart which graphically indicates the various combinations of air and wall temperatures which give equal feelings of comfort (see Chart 1).

The comfort temperatures on this chart are the same as the Pierce Laboratory's operative temperatures. This comfort temperature or operative temperature differs from the effective temperature in that it takes into account the relative effect of radiation and convection but does not take into account the effect of evaporation. However, since all of the data used in preparing the chart was obtained at relative humidities of 40 to 50% there is a relationship between comfort or operative temperature as shown on the chart and the effective temperature. An effective temperature of 58F is equivalent to an operative or comfort temperature of 60F. Similarly an effective temperature of 66F is equivalent to a comfort temperature of 70F; a 74F effective temperature is equivalent to a comfort temperature of 80F.

Use of Comfort Chart

The Comfort Chart will be found very useful for selecting the design conditions to be maintained in low temperature radiant heated rooms. Knowing the use of the room and the desired air temperature or mean radiant temperature, it is possible to select the corresponding air or mean radiant temperature to give comfortable conditions. For example, if we desire to design a radiant heating system for a children's classroom and want to maintain the air temperature at 65F the table shows that the mean radiant temperature of the walls, floor and ceiling should be approximately 70.5F. This will then give us a comfort temperature of 68F which corresponds to an effective temperature of 65F, which has been recommended as a good condition to maintain in classrooms.

[†] Associate Editor, HEATING & VENTILATING.

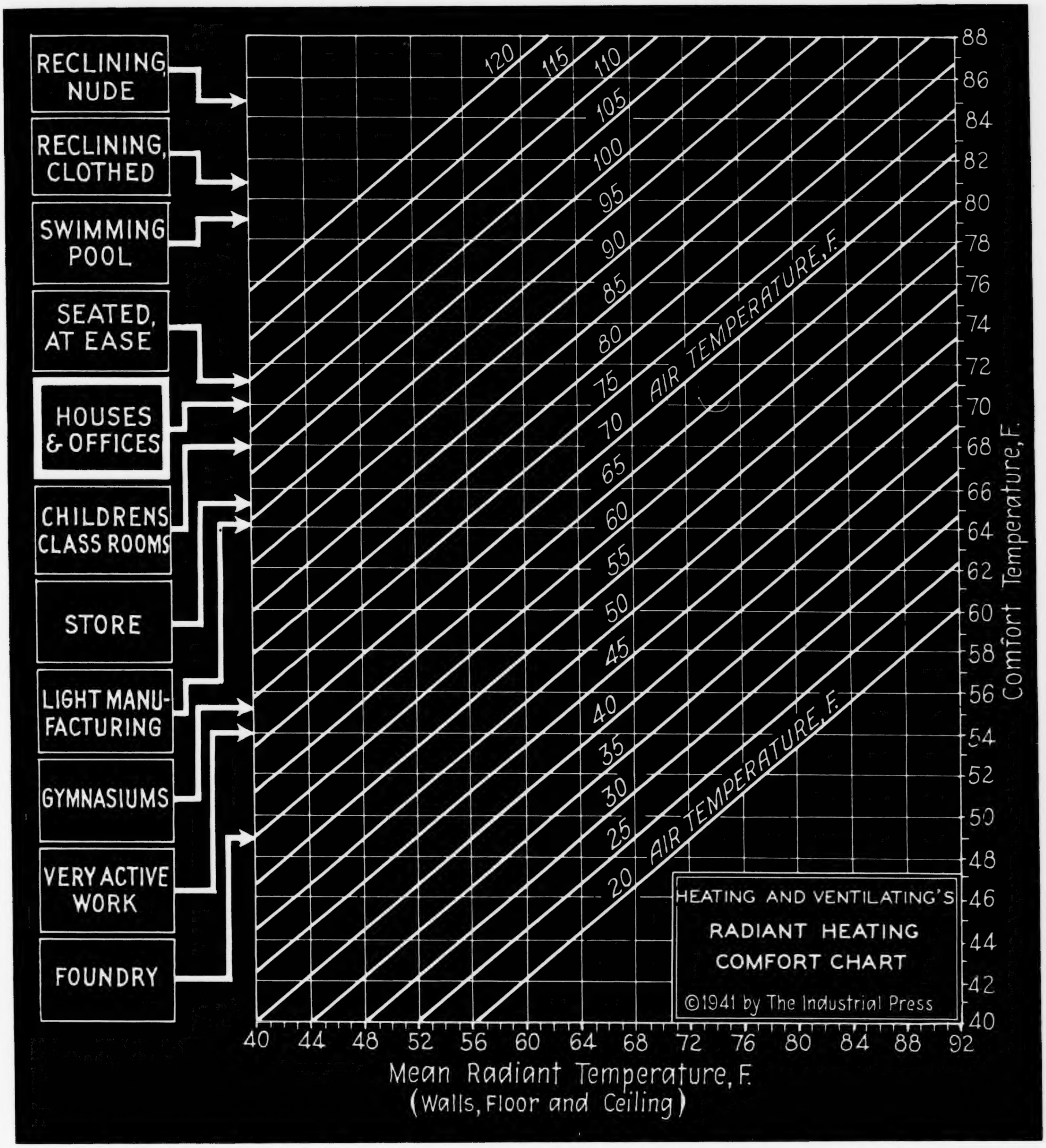


Chart 1, for use in selecting design conditions for radiant-heated buildings in U. S. To use: move right from building type to diagonal line of desired temperature, thence vertically down to required mean radiant temperature.

HOW TO DESIGN RADIANT HEATING

While the design of a radiant heating system appears to be entirely different from the design of the conventional heating system, the heating engineer will find that the problems have much in common. In general, it may be said that the problem in designing any heating system is to determine the quantity of heat to be supplied to maintain the rooms at the desired conditions. Theoretically this would not hold true for a radiant heating system since the wall temperatures would be used to control the heat loss of the occupants of the room and no attention would be paid to the room air temperature. In practice, however, it is impossible to have a true radiant heating system since the room surfaces will quickly absorb the radiant heat and the air will be heated not only by the radiant panel but also by the warmed room surfaces. In actual practice it will be found that air temperatures of only a few degrees below what is generally considered normal will be maintained. Thus the main problem in designing a radiant heating system is the same as the main problem in designing the conventional system; that is, to supply enough heat to the room to meet the heat loss.

There are some complicating factors which tend to make the design of a radiant heating system difficult. However, if certain assumptions are made these complications disappear and the problem becomes no more difficult than the design of a conventional system.

The use of Charts 2 and 3 will be of considerable help in reducing these complications. Chart No. 2 is designed for use in calculating a floor radiant heating system while Chart No. 3 is for use in calculating a ceiling radiant heating system. In both cases the heat released from a panel by radiation was calculated from the standard radiation formula and the heat released by convection was determined from test data.

Panel Temperature

In using these charts to determine the required ceiling or floor temperature it is necessary to know the desired room air temperature, the mean radiant temperature of the unheated walls, floor and ceiling, and the heat release of the panel.

The desired air temperature can be selected from the Radiant Heating Comfort Chart No. 1.

The mean radiant temperature of the unheated walls, floor and ceiling is determined by first calculating the surface temperature of the individual surfaces such as windows, exposed walls, interior walls and floors. Then the area of each surface is multiplied by the surface temperature and the products added. This sum is then divided by the total surface area to determine the mean surface temperature.

Actually this method gives only the approximate temperature. In those cases where there are large areas at greatly different temperatures the true mean radiant temperature can be calculated as follows:

The rate at which each surface radiates heat to surroundings is determined from Table 1. The radiating rate is then multiplied by the area of the surface to find the total radiating rate of the surface. The total radiating rates of all the surfaces are added together and the sum is divided by the total unheated area. This gives the mean radiating rate of unheated surface. This mean radiating rate is then used to find the mean wall temperature from Table 1.

The heat release is determined by dividing the total heat loss of the room by the area of the ceiling or floor which is to be heated. For example, if the heat loss of the room is 4000 B.t.u. per hr. and the ceiling area is 200 sq. ft., the heat release of the ceiling will have to be 4000 ÷ 200 or 20 B.t.u. per sq. ft. to meet the heat demand of the room. Knowing these three conditions we can determine from Chart 3 the required surface temperature.

In solving an actual problem there are eight steps to determine the required heated panel temperature:

1. Determine the comfort temperature from the Radiant Heating Comfort Chart 1.

2. Calculate the heat loss of the room assuming that the air temperature is the same as the comfort temperature which was found from Chart 1. Usually the air temperature in most radi-

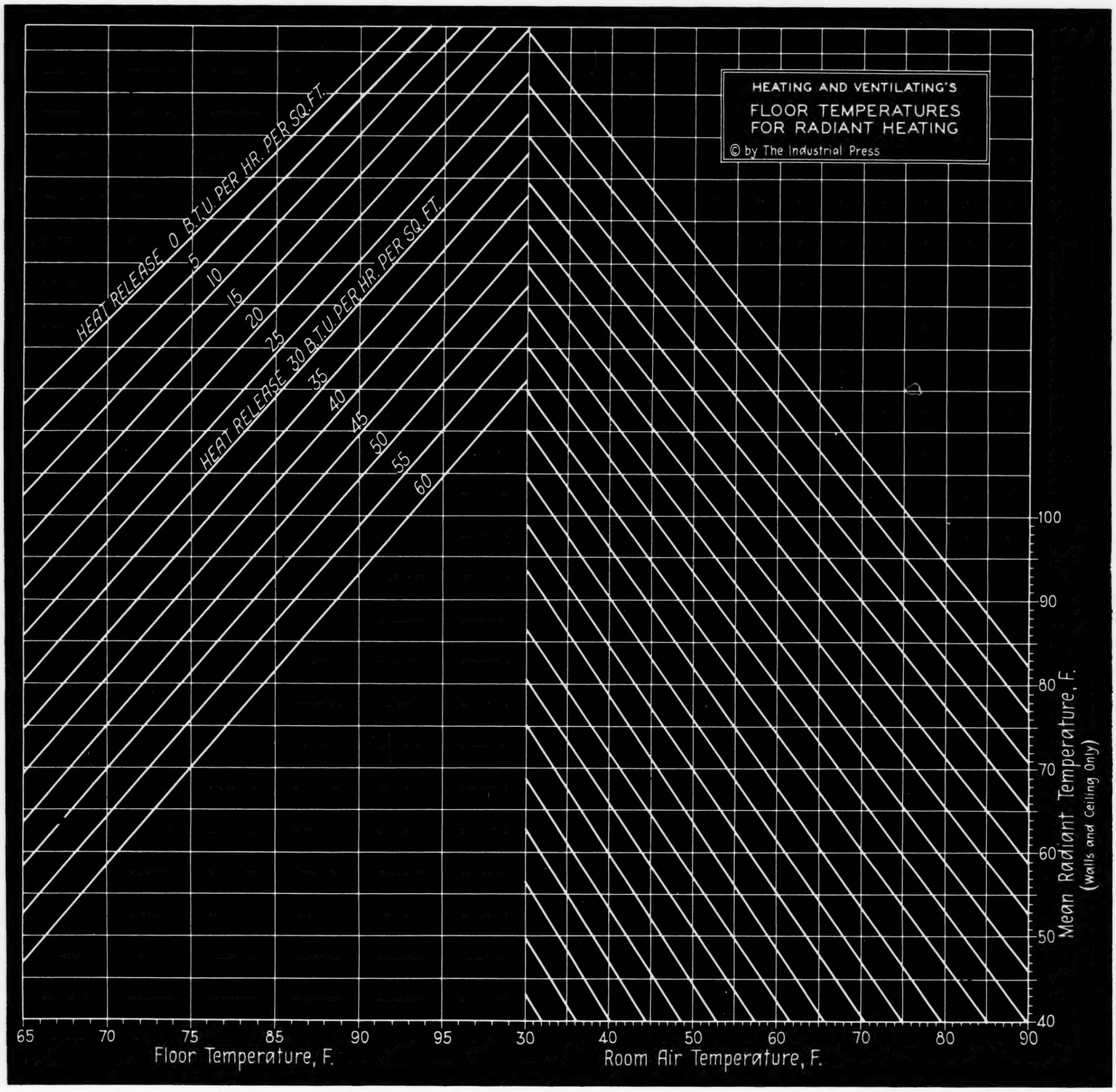


Chart 2, for determining floor temperatures in floor radiant heating systems. To use: locate intersection of predetermined mean radiant temperature and desired room air temperature. From intersection follow left along diagonal lines to vertical center line. Move horizontally to desired heat release, thence vertically down to floor temperature.

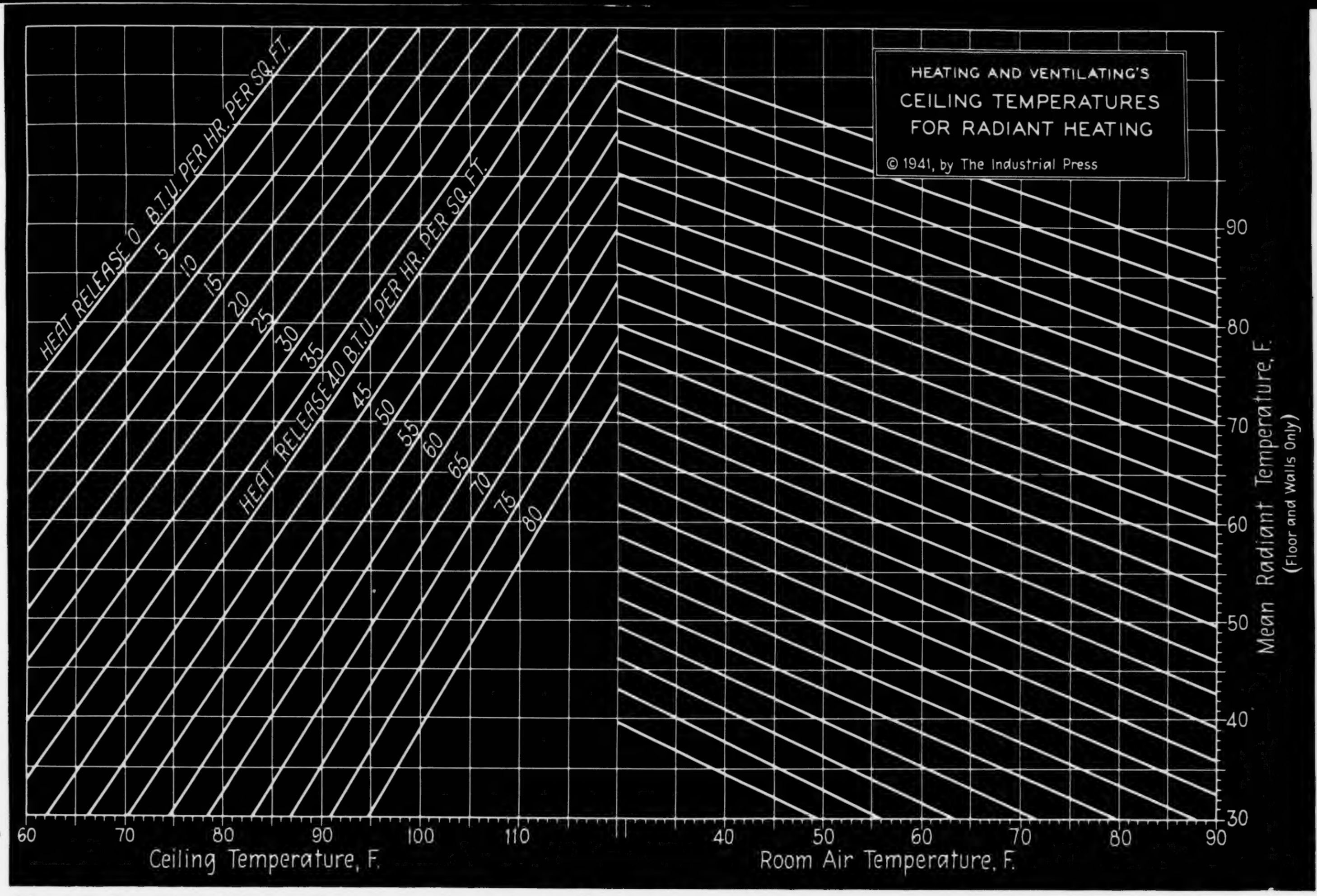


Chart 3, for determining ceiling temperatures for ceiling heating systems. To use: locate intersection of predetermined mean radiant temperature and desired room air temperature. From intersection follow left along diagonal lines to vertical center line. Move horizontally to desired heat release, thence vertically down to ceiling temperatures.

ant heating installations, will be somewhat lower than this, but due to the comparatively high rate of radiation in the room, wall temperatures will be higher than would normally be expected with the air temperature maintained. For this reason an air temperature equivalent to the comfort temperature will give a heat loss very close to the actual heat loss of the room. In these calculations, infiltration heat loss should be included.

3. Determine the desired heat release of the heating panel by dividing the heat loss of the room by the floor or ceiling area which is to be heated.

4. Determine the unheated wall, floor or ceiling temperatures. This is done by establishing the inside-outside temperature difference and the coefficient of the wall, window, floor or ceiling. When these two points are known Table 2 will show the temperature difference between the air and the wall surface. If this figure is then subtracted from the comfort air temperature the wall surface temperature is determined. This should be done for each unheated surface.

5. The mean radiant temperature of the unheated surface of the room should next be calculated. This is done by multiplying the area of each surface by its temperature. These products are then added together and the sum is divided by the total areas of the unheated surface. This gives the mean temperature of the unheated room surface. Where large areas of widely different temperatures are involved use the mean radiant temperature method explained previously.

6. The actual room air temperature to be maintained and the mean radiating rate are next selected from the Radiant Heating Comfort Chart 1.

7. Determine the required panel temperature from Chart 2 or 3 using the mean radiant temperature of the unheated room surfaces, the room air temperature and the desired panel heat release.

8. Determine the room mean radiant temperature to see if it agrees with the temperature selected in step 6. This is done by multiplying the mean radiating rate of the heated panel by the heated panel area and adding this to the product of the mean radiating rate of the unheated room surface and unheated room area. If this figure is then divided by the total room area, the mean radiating rate of the room is

TABLE 1.—HEAT RADIATION TO SURROUNDINGS

Surface Temperature, F	Radiation, B.t.u. per Sq. Ft. per Hr.	Surface Temperature, F	Radiation, B.t.u. per Sq. Ft. per Hr.	Surface Temperature, F	Radiation, B.t.u. per Sq. Ft. per Hr.
30	89.4	60	113.3	90	142.0
31	90.0	61	114.2	91	143.1
32	90.9	62	115.0	92	144.0
33	91.5	63	116.0	93	145.1
34	92.1	64	117.0	94	146.2
35	93.0	65	117.7	95	147.1
36	93.8	66	118.6	96	148.4
37	94.5	67	119.5	97	149.3
38	95.3	68	120.5	98	150.3
39	96.0	69	121.3	99	151.7
40	96.8	70	122.3	100	152.6
41	97.5	71	123.1	101	153.8
42	98.4	72	124.1	102	155.0
43	99.1	73	125.1	103	156.0
44	100.0	74	126.0	104	157.0
45	101.0	75	127.0	105	158.0
46	101.8	76	128.0	106	159.0
47	102.5	77	129.0	107	160.7
48	103.2	78	129.9	108	161.8
49	104.0	79	130.9	109	162.5
50	105.0	80	131.9	110	163.8
51	105.8	81	132.9	111	165.0
52	106.4	82	133.8	112	166.0
53	107.5	83	134.7	113	167.1
54	108.2	84	136.0	114	168.2
55	109.0	85	136.7	115	169.7
56	109.9	86	137.8	116	170.7
57	110.7	87	138.7	117	171.7
58	111.6	88	139.9	118	173.0
59	112.6	80	140.9	119	174.5

determined. Table 1 will then give the mean radiating temperature. If this temperature is higher than that selected in step 6, a lower air temperature should be selected and steps 7 and 8 should be made over again. If the mean radiant temperature is lower than desired, a higher air temperature should be selected and steps 7 and 8 should be made again.

Panel Temperature Example

As an example of how the panel temperature is calculated in actual practice a typical problem is outlined. Suppose a radiant heating system is to be installed in a living room which is 20 x 10 ft. in area and 8 ft. high. The room has one 10 x 8 ft. and one 20 x 8 ft. outside wall; there are two 3 x 5 ft. windows with storm sash; the wall coefficient is 0.20 B.t.u. per sq. ft. per hr. per degree temperature difference; the window coefficient is 0.58 B.t.u. per sq. ft. per hr. per degree temperature difference; and the cutside design temperature is 0F.

SOLUTION:

1. Select the comfort temperature. The Radiant Heating Comfort Chart shows that for a living room the comfort temperature should be 70F.

2. The heat loss of a house should be calculated. The exposed wall area is 210 sq. ft. and the heat lost through this wall area is 210 × 0.20 × 70F = 2940 B.t.u. per hr. The window loss is 30 sq. ft. × 0.58 × 70F = 1302 B.t.u. per hr. Infiltration loss is ½ × 2 windows × 19 ft. of crack × 30 B.t.u. per ft. = 570 B.t.u. per hr. The total heat loss, therefore, is 4812 B.t.u. per hr.

3. Determine the heat loss of the panel. If the ceiling is used for heating, the ceiling area is 10 × 20 or 200 sq. ft. and the heat release per sq. ft. of ceiling area is 4812 ÷ 200 or 24 B.t.u. per hr. per sq. ft. of ceiling.

4. Determine the temperatures of the unheated walls, floor and ceiling. The unheated wall temperature is found from Table 2 which shows that for an inside-outside temperature difference of 70F and for a wall coefficient of 0.20 the temperature difference is 8.8F and the wall temperature 62.2F. The window temperature is found, in a like manner from Table 2, to be 43.3F. The inside wall and floor areas are 70F since they are heated on both sides.

5. Determine the mean radiant temperature of the unheated surface of the room. This is done by multiplying the various room surface temperatures and dividing the sum by the total area:

SURFACE	TEMPERATURE F	AREA SQ. FT.	PRODUCT
Exposed walls	61.2	210	12,810
Windows	43.3	30	1,300
Interior walls	70	240	16,800
Floor	70	200	14,000
TOTALS		680	44,910

TABLE 2.—TABLE FOR COMPUTING UNHEATED WALL SURFACE TEMPERATURES FOR RADIANT HEATING

WALL TRANSMISSION COEFFICIENT B.T.U. PER HR. PER SQ. FT. PER DEG.	TEMPERATURE DIFFERENCE ACROSS WALL, F								
	20	30	40	50	60	70	80	90	100
	TEMPERATURE DIFFERENCE—ROOM AIR TO INSIDE WALL, F								
0.05	0.7	1.0	1.3	1.6	2.0	2.3	2.6	3.0	3.3
0.06	0.8	1.2	1.6	2.0	2.4	2.8	3.2	3.6	4.0
0.07	0.9	1.4	1.8	2.3	2.8	3.2	3.7	4.1	4.6
0.08	1.0	1.6	2.1	2.6	3.2	3.7	4.2	4.7	5.3
0.09	1.2	1.8	2.4	3.0	3.5	4.1	4.7	5.3	5.9
0.10	1.3	2.0	2.6	3.3	4.0	4.6	5.3	5.9	6.6
0.12	1.6	2.4	3.2	3.9	4.7	5.5	6.3	7.1	7.9
0.14	1.8	2.8	3.7	4.6	5.5	6.5	7.4	8.3	9.2
0.16	2.1	3.2	4.2	5.3	6.3	7.4	8.4	9.5	10.5
0.18	2.4	3.5	4.7	5.9	7.1	8.3	9.5	10.7	11.8
0.20	2.6	3.8	5.0	6.3	7.5	8.8	10.0	11.3	12.5
0.22	2.9	4.3	5.8	7.2	8.7	10.2	11.6	13.0	14.5
0.24	3.2	4.7	6.3	7.9	9.5	11.1	12.6	14.2	15.8
0.26	3.4	5.1	6.8	8.6	10.3	12.0	13.7	15.4	17.1
0.28	3.7	5.5	7.4	9.2	11.1	12.9	14.7	16.6	18.4
0.30	3.9	5.9	7.9	9.9	11.9	13.8	15.8	17.8	19.8
0.32	4.2	6.3	8.4	10.5	12.6	14.7	16.8	18.9	21.0
0.34	4.5	6.7	9.0	11.2	13.4	15.7	17.9	20.2	22.4
0.36	4.7	7.1	9.5	11.9	14.2	16.6	19.0	21.4	23.7
0.38	5.0	7.5	10.0	12.5	15.0	17.5	20.0	22.5	25.0
0.40	5.3	7.9	10.5	13.2	15.8	18.5	21.1	23.7	26.4
0.42	5.5	8.3	11.1	13.9	16.6	19.4	22.2	24.9	27.7
0.44	5.8	8.7	11.6	14.5	17.4	20.3	23.2	26.1	29.0
0.46	6.1	9.1	12.1	15.1	18.2	21.2	24.2	27.2	30.3
0.48	6.3	9.5	12.7	15.8	19.0	22.1	25.3	28.4	31.6
0.50	6.6	9.9	13.1	16.4	19.7	23.0	26.3	29.6	32.9
0.58	7.6	11.5	15.3	19.1	23.0	26.7	30.6	34.4	38.2
0.70	9.2	13.8	18.4	23.0	27.6	32.2	36.8	41.4	46.0
1.24	16.3	24.4	32.6	40.8	49.0	57.0	65.2	73.4	81.5

Mean temperature is 44,910 ÷ 680 or 66F. As mentioned previously this method is only approximately correct. If it is desired to determine the temperature more accurately it should be done by the mean radiant temperature method as follows:

SURFACE	TEMPERATURE F	RADIATION RATE, B.T.U. PER SQ. FT. PER HR.	AREA, SQ. FT.	PRODUCT, Rate x Area
Exposed walls	61.2	114.4	210	24,024
Windows	43.3	99.4	30	2,982
Interior walls	70.0	122.3	240	29,352
Floors	70.0	122.3	200	24,460
TOTALS			680	80,818

Therefore, the mean radiating rate is 80,818 ÷ 680 sq. ft. or 118.8 B.t.u. per hr.

Table 2 shows that this is equivalent to a mean room temperature of 66.2F. The short cut method in this case gives us a temperature of 66F as against the actual temperature of 66.2F. This is close enough for practical purposes and the mean radiating temperature method need only be used where large wall areas of widely different temperatures are encountered. A good example of such an instance would be a room which has one side of plate glass.

6. Select the actual room air temperature and mean radiating rate to be maintained. The Radiant Heat Comfort Chart will give us this figure and if we assume a 65F air temperature, the mean radiating temperature of the entire room will have to be 74F.

7. Determine the required panel temperature. For a ceiling surface we use Chart 3 which indicates that for a room air temperature of 65F, a mean radiant temperature of the *unheated* surface of 66F and a desired heat release of 24 B.t.u. per hr. per sq. ft., the ceiling temperature will have to be 84F.

8. Determine the room mean radiating temperature to find if it agrees with the temperature selected in step 6. The total radiating rate is found to be 80,818 in step 5 for the unheated surface. To this add the radiating rate of the ceiling which is 136.0 (from Table 2) multiplied by the area or 27,200. Adding this to 80,818 the total radiating rate of the room is found to be 108,018. The mean radiating rate is therefore 108,018 ÷ 880 (the total area of the room) or 122.8 B.t.u. per hr. This is equivalent to a 70.6F mean radiant temperature. Since the mean radiating temperature should be 74F with an air temperature of 65F, we will have to select a higher air temperature and calculate steps 7 and 8 again.

Recalculation

6. Let us assume a 69F air temperature. Then the Radiant Heating Comfort Chart shows that the mean radiant temperature should be 71F.

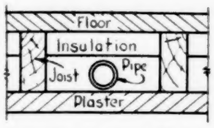
7. Chart 3 shows that for a 69F air temperature, a 66F mean radiating temperature of the *unheated* surface, and a panel heat release of 24 B.t.u. per hr. per sq. ft., a panel temperature of about 86F is necessary.

8. The mean radiating temperature is therefore 80,818 + (200 × 137.8) = 80,818 + 27,560 = 108,378. Divide this by the total room area of 880 to

find the mean radiating rate is 123.1. This is equivalent to a mean radiant temperature of approximately 71.0. This compares with a desired mean radiant temperature of 71. Therefore, if a panel temperature of approximately 86F is maintained in zero weather the room temperature will be kept at 69F and the room mean radiant temperature will be about 71.0F.

DESIGN OF HEATING SURFACE

Once the temperature of the radiant heating surface is determined the next step is to consider the methods available for heating the surface. American practice is to heat the floor with either steam or hot water coils or warm air ducts buried beneath or in the floor; or ceiling panels are heated with hot water or steam pipes buried either in the plaster or mortar or located in the air space above the plaster. Warm air ducts in the space between the ceiling and the floor above are also used. Wall heating, also, is not uncommon.



PIPES IN CEILING AIR SPACE. Steam or hot water pipes are located in the air space formed by joists. Pipes can be run either parallel or at right angle to joists. Iron or steel pipe is usually employed because the heat release to surrounding air is greater than if copper or brass pipe is used. Table 3 gives approximate heat release of iron, steel and copper tubes to air at temperatures usually encountered in these applications.

Only Room Below Heated. When the ceiling is used for heating the room below and the space above is unheated, the following method is used for designing the ceiling coil:

1. Ceiling heat release and ceiling surface temperature is determined as explained previously.

2. Determine the temperature of the air in the joist space. This is done by solving the equation

$$T_a = \frac{H_c}{U_c} + t_s$$

where

T_a = air temperature in ceiling space, F.
H_c = Desired heat release of ceiling, B.t.u. per hr. per sq. ft.
U_c = Heat transmission coefficient of ceiling, B.t.u. per hr. per sq. ft. per F.
t_s = Ceiling surface temperature, F.

The heat transmission coefficient of the ceiling can be taken as 1.2 B.t.u. per sq. ft. per hr. per F for metal lath and plaster; 1 B.t.u. for wood lath and plaster and 1.7 B.t.u. for plain plasterboard.

3. Determine the heat loss through top of air space. This is done by solving the formula

$$H_t = U_t\,(T_a - T)$$

where

H_t = the heat loss through top of air space in B.t.u. per sq. ft. per hr.
U_t = the heat transmission coefficient through the top of air space in B.t.u. per sq. ft. per hr. per degree temperature difference.
T_a = the temperature of the air in the joist space.
T = the temperature of the space above the ceiling.

4. Determine the heat release of the pipe per lineal foot. To do this first find the total release per sq. ft. of space which is the ceiling heat release plus the heat loss through the top of the air space. If joists are spaced on 12-in. centers this figure will be the required heat release of the pipe per lineal foot. If joists are on 16-in. centers the figure should be multiplied by 1.33 to find the required heat release per lineal foot of pipe. If joists are on 24-in. centers the figure should be multiplied by 2 to find the pipe heat release.

5. Determine the temperature of the heating medium in the pipe and pipe size. Knowing the heat release per lineal foot of pipe refer to Table 3 to find the required pipe size and temperature difference. The temperature of the heating medium will then be the sum of the temperature of the air in the joist space plus the temperature difference (from Table 3).

6. For hot water coils determine the amount of water to be circulated. This is done by dividing the total heat release per hour of the coil by the temperature drop (usually taken as 10 to 20F). This gives the number of pounds of water to be circulated per hour.

For steam coils, divide the total heat release per hour of the coil by the latent heat per pound of steam at the required temperature. This gives the number of pounds of steam to be supplied per hour.

Both Room Above and Room Below Heated. The method of calculating pipe coils for such applications is practically the same as outlined for ceiling coils heating only the room below. However, step 3 is omitted and in step 4 the total of the ceiling and floor heat release gives the heat release of the pipe per lineal foot.

An additional step is necessary, however, to determine the amount of insulation to install in the floor to provide the required floor temperature.

First determine the resistance to the floor. This is done by dividing the temperature difference between the joist air and the floor surface by the heat release of the floor.

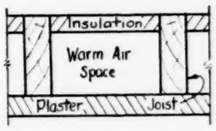
$$R_f = \frac{(T_a - T_s)}{H_f}$$

where

H_f = the resistance of the floor.
T_a = joist air temperature.
T_s = floor surface temperature.
H_f = heat release of the floor.

The total resistance of an ordinary floor consisting of a sub-floor and finished floor is 1.94. The difference between this and the calculated resistance is the resistance of the insulation to be added.

The average rigid insulating board has a resistance of about 3 per inch thickness while a sheet of aluminum foil will have a resistance of about 2.2.



CEILING HEATED BY WARM AIR. Usual practice is to pass warm air through joist space, although in some cases special ducts are used. Where space above is not heated insulation is placed at the top of the air space.

Only Room Below Heated. Warm air temperature and volume is calculated as follows:

1. The ceiling heat release and ceiling surface temperature is determined as explained previously.

2. Determine the temperature of the air in the joist space. This is done by solving the equation

$$T_a = \frac{H_c}{U_c} + T_s$$

where

T_a = the air temperature in the ceiling space.
H_c = the desired heat release of the ceiling in B.t.u. per hr. per sq. ft.

TABLE 3.—HEAT RELEASE OF PIPES IN AIR

Temp. Diff., Heating Medium to Surrounding Air	Iron or Steel Pipe						Copper Tube				
	Nominal Size, In.						Nominal Size, In.				
	½	¾	1	1¼	1½	2	½	¾	1	1¼	1½
	Heat Release, B.t.u. per Hr. per Lineal Foot										
5	2.0	2.7	3.8	4.3	5.4	6.4	1.3	1.9	2.3	2.6	3.1
10	4.0	5.1	7.1	8.6	10.4	12.5	2.8	3.4	4.5	5.1	6.1
15	6.5	8.0	10.8	13.0	15.8	19.0	4.1	5.3	6.9	7.9	9.5
20	9.1	11.0	14.2	17.6	21.0	25.4	5.6	7.1	9.1	10.5	12.9
25	11.6	14.0	17.9	22.2	26.3	32.1	7.0	9.1	11.7	13.5	16.1
30	14.5	17.5	22.0	27.0	31.0	38.0	8.6	11.0	14.0	16.3	19.4
35	16.6	20.0	25.3	31.5	36.5	44.8	10.2	13.0	16.8	19.5	23.0
40	19.4	23.1	29.3	36.5	42.0	51.2	12.0	15.1	19.3	23.0	26.8
45	22.0	26.1	33.0	41.3	47.5	58.0	13.5	17.2	22.0	26.0	30.3
50	23.7	29.5	37.3	46.1	53.0	64.2	15.0	19.3	24.9	29.5	34.0
55	27.2	33.0	41.1	51.1	58.5	71.0	17.0	21.7	27.8	33.0	38.2
60	30.0	36.3	45.1	56.3	64.0	77.2	18.5	24.0	30.5	36.4	42.0

U_c = the heat transmission coefficient of the ceiling in B.t.u. per sq. ft. per hr. per degree temperature F.

T_s = the space temperature.

The heat transmission coefficient for the ceiling can be taken as 3 B.t.u. per sq. ft. per degree temperature F where plaster on metal lath is used; 2 B.t.u. where plaster on wood lath is used and 2.2 B.t.u. where plain plasterboard is used.

3. Determine the heat loss through the top of the air space. This is done by solving the formula

$$H_t = U_t\ (T_a - T)$$

where

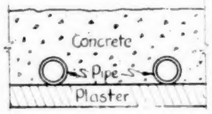
H_t = heat loss through the top of the air space in B. t.u. per sq. ft. per hr.

U_t = the heat transmission coefficient through the top of the air space in B.t.u. per sq. ft. per hr. per degree temperature F. In determining this heat transmission coefficient film should be included.

T_a = the temperature of the air in the joist space.

T = the temperature of the space or material above the floor.

4. Determine the volume of air to be supplied. To do this add the ceiling heat release to the heat loss through the top of the air space to find the total release per sq. ft. Multiply this total release per sq. ft. by the number of sq. ft. in the ceiling to find the total heat release of the ceiling panel. If this total heat release is divided by the temperature drop of the air and the specific heat of a cu. ft. of air the volume of air required will be determined.



PIPE COILS IN CEILING OF CONCRETE. Pipe coils are imbedded in concrete in the underside of the ceiling of the room to be heated and surface is plastered over coils. Usual practice is to use ½- or ¾-in. copper, wrought iron or steel pipe bent in coil form on 4 to 6 in. centers.

Only Room Below Heated. The method of designing pipe coils for radiant ceiling panels is as follows:

1—Determine panel heat release and ceiling surface temperature as outlined previously.

2—Determine average temperature of concrete in plane of pipe coils. This is done by solving the following equation:

$$T_p = \frac{H}{U} + T_s,$$

where

T_p = temperature of concrete in plane of pipe coils;

T_s = ceiling surface temperature;

H = the heat release of the ceiling in B.t.u. per sq. ft. per hr.,

U = the coefficient of heat transmission of the concrete—can be taken as 12 B.t.u. per sq. ft. per hr. per inch thickness per degree F.

3—Determine the heat loss to space above. This is done by solving the equation

$$H = U\,(T_p - T)$$

where

H = the heat loss to the space above in B.t.u. per sq. ft. per hr.

U = the coefficient of heat transmission from pipe coil plane to air space above.

T_p = the temperature of concrete in plane of coil.

T = the temperature of air space above.

In calculating U, note that there is only one surface coefficient to be included.

4—Determine the average temperature of heating medium in pipe. This can be estimated by solving the following equation

(For pipe on 6-in. centers)

$$T_m = 0.15H + T_s$$

(For pipe on 4-in. centers)

$$T_m = 0.12H + T_s$$

where

T_m = the average temperature of heating medium in pipe.

T_s = the average temperature of the ceiling.

H = the ceiling heat release, B.t.u. per sq. ft. per hr.

5—Determine the amount of heating medium to be supplied per hour. For hot water coils, divide the total heat release and heat loss of the panel by the temperature drop of the water (usually 10 to 20F) to find the pounds of water to be circulated per hour.

For steam coils, divide the total heat release and heat loss of the panel by the latent heat of the steam at the required temperature to find the pounds of steam to be supplied per hour.

Both Room Below and Room Above Heated. The method of calculating pipe coils for such applications is practically the same as outlined for ceiling coils heating only the room below. However, step 3 is omitted and in step 4 the total of the ceiling and floor heat release is used to determine the amount of heating medium to be supplied.

An additional step is necessary, however, to determine the amount of insulation which has to be placed between the coils and the floor above to provide the required floor temperature. The required coefficient of heat transmission of the construction above the pipe coil can be determined from the following formula:

$$U = H\,(T_p - T_f)$$

where

U = the required heat transmission coefficient of construction above the pipe coil.

H = the floor heat release.

T_p = the average temperature in the plane of the pipe coil.

T_f = the average floor temperature desired.

Once this coefficient is known a construction to give this coefficient can be selected.

This type of coil has been much used in English practice and some design data are available. If these data are used, it should be remembered that ceiling temperatures are higher and room temperatures lower than in American practice.

Tests on pipe coils in concrete ceilings showed that as the spacing of the pipe was increased the heat release of the surface decreased but the heat release per lineal foot of pipe increased. For iron pipes of ½ in. diameter the heat release of the panel in B.t.u. per sq. ft. per hr. for a 60F temperature difference between room air and pipe water temperature was for a spacing of 4½ in., 115; 6 in., 100; 9 in., 70; 12 in., 54; 18 in., 38. Usual practice in England is to space coils 6 in. apart.

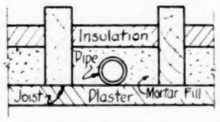
The heat release of ceiling panels has been listed by some English authorities as shown in Table 4.

TABLE 4.—HEAT RELEASE OF CEILING PANELS

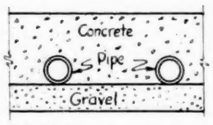
Temp. Diff. Room Air to Pipe Water	Heat Release B.t.u. per Hr. per Sq. Ft.
20	24
30	40
40	60
50	80
60	102
70	123
80	146

This data applies to pipe coils on 6-in. centers imbedded in concrete with ceiling plastered. Maximum hot water temperature is usually considered in England to be 135F.

PIPE COILS IN WOOD CONSTRUCTION WITH MORTAR FILL. This type of construction differs from the preceding type only in that pipe coils are surrounded by a mortar fill rather than concrete. Method outlined above can be used except the mortar transmission coefficient can be taken as 8 B.t.u. per hr. per sq. ft. per inch thickness per degree F.



PIPE COILS IN CONCRETE FLOOR. Pipe coils are either imbedded in concrete floor slab or in the gravel or sand base which is between concrete and ground. Pipe from ½ in. to 2 in. diam. on from 6 in. to 4 ft. centers has been used. Either steam or hot water is used for heating. Floor surface temperatures are usually limited to a maximum of 85F and some designers prefer as low as 75F. An English authority has stated that it is found to be unpleasant and tiring to sit for long with the feet in contact with a heated floor even though the surface may be no more than 80 or 85F. In England the maximum floor temperature for comfort with continuous use is usually taken as 75F.



A method of designing pipe coils for floor heating panels is as follows:

1—Determine floor heat release and surface temperature.

2—Determine average mean temperature of concrete in plane of coils.

This is done by solving the following:

$$T_p = \frac{H}{U} + T_s$$

where

T_p = the temperature in plane of coils.

T_s = the floor surface temperature.

H = the heat release of the floor in B.t.u. per sq. ft. per hr.

U = the coefficient of heat transmission of floor between pipe coils and floor.

If the material is concrete, a heat transmission coefficient of 12 B.t.u. per hr. per sq. ft. per inch thickness per degree F can be used. Sand has a coefficient of 8, while gravel has a coefficient of 2.5.

3—Determine the heat loss to the ground below. This is done by solving

$$H = U(T_p - T_g)$$

where

H = the heat loss to ground below in B.t.u. per sq. ft. per hr.

U = the combined heat transmission coefficient of material below pipe coil.

T_p = the temperature of the concrete slab in plane of pipe coil.

T_g = the temperature of the ground.

The heat transmission coefficient of earth will vary considerably but 10 B.t.u. per sq. ft. per hr. per inch thickness per degree F for moist earth may be used. The temperature of the ground is difficult to predict because of both the lack of actual tests and the many variables. However, a ground temperature of 70F five feet below the slab is probably a good approximation of what may be expected. Therefore, the coefficient U can be calculated using this thickness and coefficient together with the coefficient of other material between pipe coil and earth.

4—Determine the average temperature of the heating medium in the pipe. This temperature can be found approximately by solving the following equation:

$$T_m = \frac{H}{U}\sqrt{d^2 + 0.06c^2} + T_s$$

where

T_m = the temperature of heating medium.

H = the heat release of the floor in B.t.u. per sq. ft. per hr.

U = Coefficient of heat transmission of the material between the pipe coils and the floor surface in B.t.u. per sq. ft. per hr. per degree F per inch of thickness. (Note that this coefficient includes no surface coefficients).

d = the thickness in inches of the floor above the pipe coils.

c = the distance between the pipes in the coils in inches.

T_s = the temperature of the floor surface.

5—Determine the amount of hot water or steam to be supplied per hour. For hot water, divide the total of the floor release and ground loss by the temperature drop of the water (usually 10 to 20F) to find the number of pounds of water to be circulated per hour.

For steam coils divide the total heat release and heat loss of the floor by the latent heat of steam at the required temperature to find the number of pounds of steam to be supplied per hour.

Hot Water System Design

While either forced circulation or gravity circulation systems may be used with radiant heated systems, gravity circulation is limited to small systems.

Sizing the mains and returns is the same as for any hot water system. However, the resistance of the pipe coils is very much larger than the resistance of radiators or convectors. A 120 ft. coil of ½ in. pipe with welded ells may have a resistance of 6 ft. of water. Thus, the load which the circulating pump has to work against is greater than for convectional systems.

An advantage of this high resistance is that the system is self-balancing to a considerable extent. This is so because a coil of a given size at the end of a main will have only a slightly higher total resistance than a coil of the same size at the beginning of a main since the main resistance is small compared to the coil resistance.

Balancing of these systems need not be highly accurate because of the small temperature drop in the coils. For example, suppose two coils are made of the same size pipe but one is twice the length of the other. Roughly, 50% more water will be forced through the shorter coil which would mean that the temperature drop instead of being, say, from 120 to 110F would only be from 120 to 113F. Therefore, the mean temperature would be 115F in the large coil and 116.5F in the small coil and the difference in heat emission would be quite small.

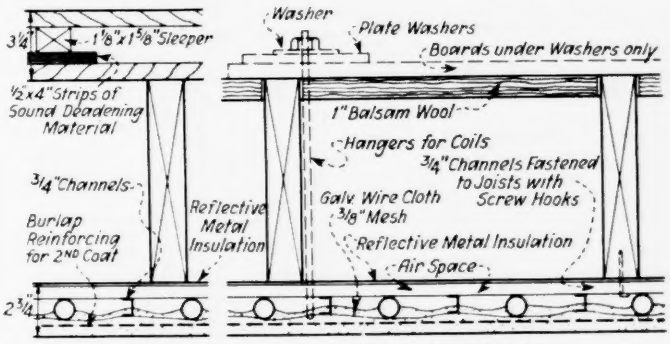
U. S. APPLICATIONS OF RADIANT HEATING

There are 84 known radiant heating installations in the U.S., and at least 13 proposed, all of which are included in the listing of applications on this and the following pages. These cover a wide range of building types, show use of steam, hot water, warm air, and electricity as the medium, and illustrate applications of steel, copper and wrought iron pipe.

The number of each project given at the left has no significance except that it assists in cross-referencing.

1 - RESIDENCE

RESIDENCE OF WINSTON ELTING, Lake Forest, Ill. Owner, an architect, designed radiant heating for ceiling throughout, as shown in accompanying sketches. Pipe coils are of standard seamless tubes welded into Tube-Turns and reenforced with steel braces welded across the lines. Water temperature control provided by automatic mixing of water from boiler with return water, with a maximum water temperature of 130F under control of outside-inside thermostat. Each coil manually controlled to balance system.



FLOOR CONSTRUCTION CEILING CONSTRUCTION

HEATING AND VENTILATING
REFERENCE SECTION
No.
1
OF A SERIES

U. S. APPLICATIONS

2 - RESIDENCE

RESIDENCE OF H. F. JOHNSON, JR., "Wing Spread," near Racine, Wis. House designed by Frank Lloyd Wright, steam in pipe coils.

3 - RESIDENCE

RESIDENCE OF JOSEPH McHARRIS, Utica, N. Y. Warm air ducts in ceiling similar to that described in project 7. This installation was made about 1938 and its success led to the building of project 7. Warm air from an International No. 125 direct fired unit with an oil burner is supplied to the ceiling ducts by a fan under the control of an outdoor bulb with a control in the warm air duct for burner operation. Fan ran constantly when outdoor temperature was below 65F under this control and resulted in overheating in mild weather. Later a thermostat was installed as a high limit control so that when it was satisfied both the blower and the burner would stop. Two winters of operation have been satisfactory to the owner. Fuel bills are reported as low. System designed by H. F. Randolph, International Heater Company, in collaboration with P. W. Hamjy, Utica contractor who installed the job.

4 - POOL

ST. GEORGE HOTEL, Brooklyn, N. Y. Radiant heating installation under floor in area around the edges of swimming pool, under the tiling. Hot water is heating medium, maintained at 106 to 110F. Heating designed by Jaros & Baum, New York. Installed by Baker, Smith & Co., Inc., New York.

5 - EXPERIMENTAL

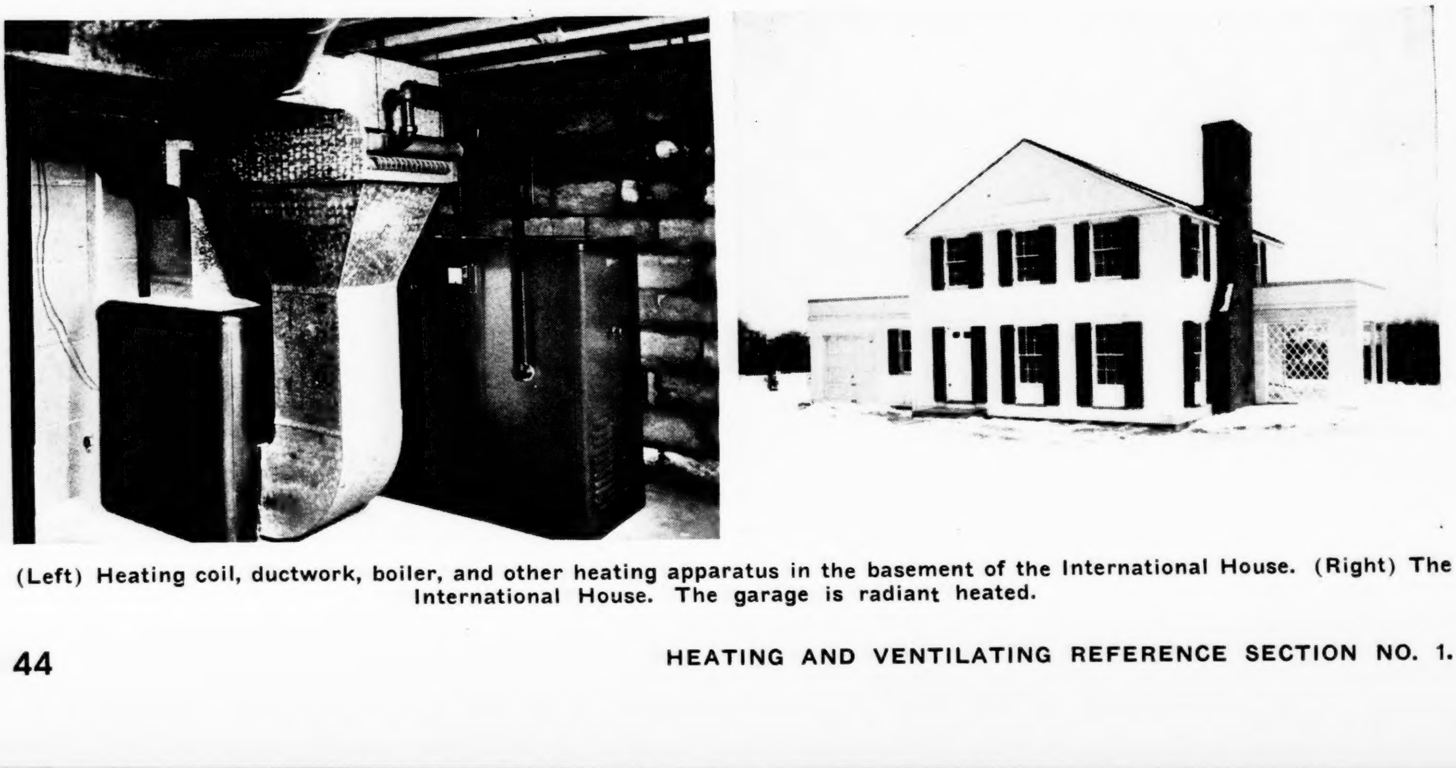
JOHN B. PIERCE LABORATORY OF HYGIENE, New Haven, Conn. A test room in this laboratory for special research work uses high temperature electric radiant heaters. Radiation is reflected back and forth on copper enclosures so that the subject in the enclosures is subjected to reflected and consequently low temperature radiation. Application is highly specialized and fully described in publications of the Laboratory.

6 - RESIDENCES

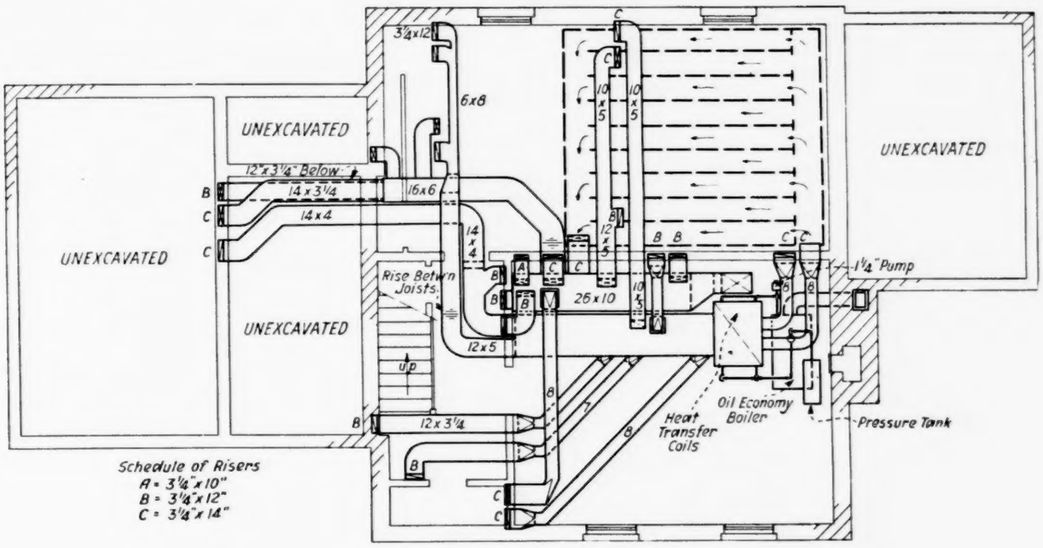
RESIDENCES, DETROIT, Mich. Fourteen residences designed and built by C. A. Robinson in Detroit and Dearborn. System uses a gravity warm air furnace. Air, leaving the furnace, is carried up through a central duct to the space between the first and second floors. The air then flows across through the steel joists by gravity down through the outer walls and back under the first floor through the return to the heater. An ordinary room thermostat is used for controlling the operation of the burner. Since no fire stops are used in the wall the entire basement ceiling is covered with plaster or plasterboard.

7 - EXPERIMENTAL

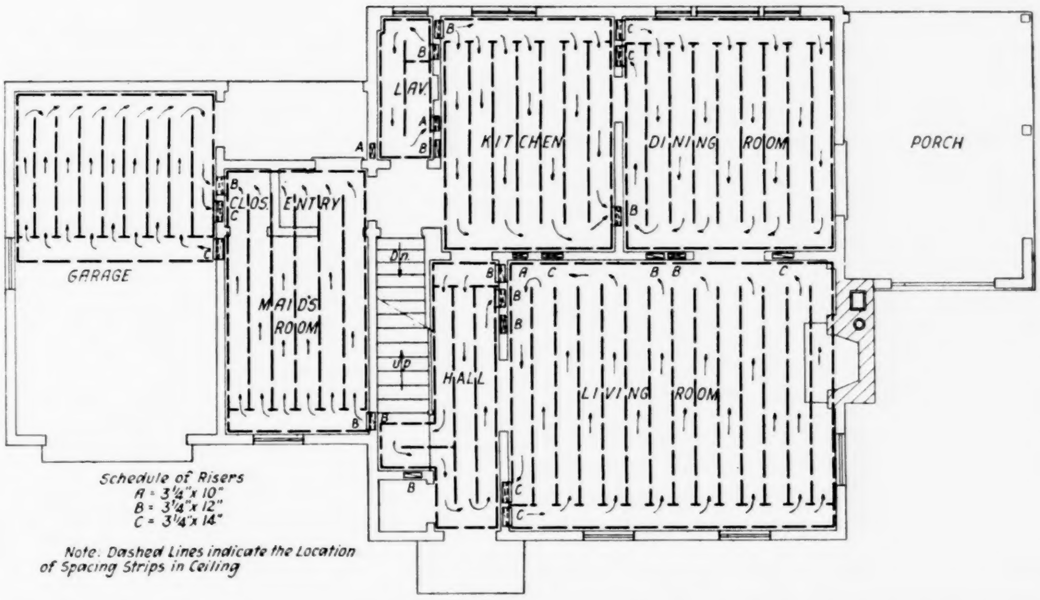
INTERNATIONAL HEATER COMPANY RESIDENCE, Utica, N. Y. Success with another residence (project 3) led International Heater to build this project for experimental purposes. Residence has eight rooms and attached garage. Company's No. 40 Oil Economy boiler fired by a Roberts-Gordon gas burner supplies water to



(Left) Heating coil, ductwork, boiler, and other heating apparatus in the basement of the International House. (Right) The International House. The garage is radiant heated.

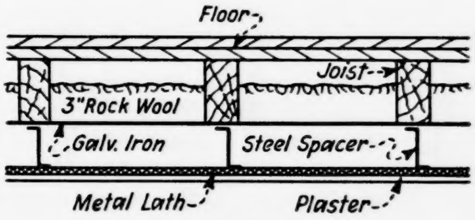


Basement plan of the International Heater Company experimental residence.



First floor plan of the International residence. Heating of second floor is largely similar.

a Trane blast coil. Warm air is circulated through spaces above ceilings as indicated on accompanying drawings. In addition, half of the garage ceiling is similarly heated, that part being the portion which extends over the car hood. Details of ceiling construction are shown in accompanying drawing. Thermocouples were placed at 170 points through the house. In most rooms five couples are imbedded in the ceiling and five in the floor. Couples are located in each supply and return riser adjacent to the ceiling. Three couples are located throughout each room to furnish information on ambient air temperatures. Readings from thermocouples can be taken at the selector switchboard in basement. Minneapolis-Honeywell is cooperating with International in the project. Architect, Charles R. Greenidge, Utica. System designed by H. F. Randolph of International Heater Company. Completion of house scheduled for March, 1941.



Not to Scale

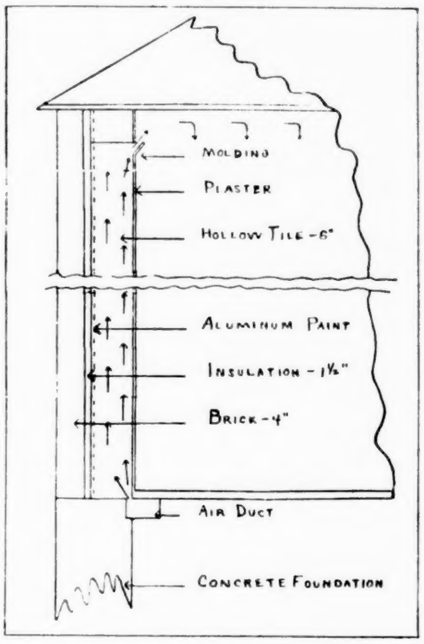
8-RESIDENCE

RESIDENCE OF BENJAMIN REHBUHN, Great Neck, N. Y. Installation completed in 1938. Coils are 1¼ in. wrought iron pipe, welded, laid on gravel under the concrete slab which forms the floor. Pipe connected to cross headers by welded joints and these in turn connected to oil-burning boiler in basement. Boiler heats water to 85F in the morning and when room temperature has reached the desired temperature boiler is turned off and heat storage in floor slab and in the water maintains comfortable conditions for considerable length of time. Owner reports house economical to heat. Frank Lloyd Wright was the architect; Long Island Air Conditioning Co. made the radiant heating installation.

U. S. APPLICATIONS

9 - RESIDENCE

RESIDENCE IN DENVER, Colorado. A single story house completed about 1935 and with walls of 4 in. brick, 1½ in. insulating material, and 6 in. hollow tiles finished with plaster. The hollow tiles are placed end to end to form a continuous series of vertical air ducts around the outside walls. Sheet metal ducts from the heating plant supply warm air to tiles. The walls thus emit radiant heat. System is a combination radiant and convection plant inasmuch as the warm air is introduced into the rooms through a ⅝ in. opening in the tile at the ceiling. System designed by Lester L. Jones, architect, Denver.

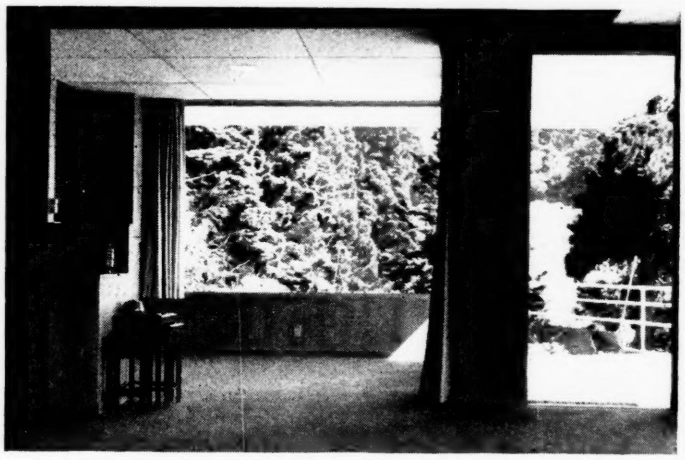


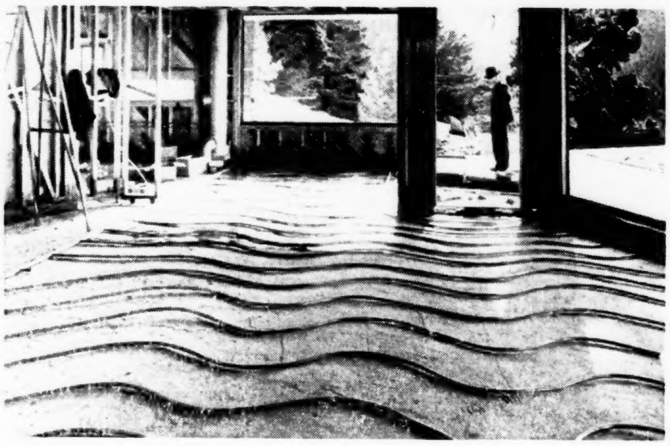
10 - RESIDENCE

RESIDENCE OF RALPH POMERANCE, Cos Cob, Conn. A 1940 installation of Wolff & Munier, Inc., using 130F water circulated under pressure.

11 - RESIDENCE

RESIDENCE OF EDW. A. SMITH, Oakland, Calif. A one-story modern frame bungalow, five rooms, cost $7500. A 4-in. concrete slab was poured directly on ground and on this slab coils of ¾ in. soft copper tubing, in wavy formation on 18-in. centers, were laid with supply opening to each lateral being orificed to balance the flow. Sand filled in between coils to top level, then a 2-in. colored cement slab poured on top. Floors are heavily carpeted. Heater is a 150,000 B.t.u. Big 4 Booster (General Water Heater Corp.), controlled at 160F. Flow control valve above heater; Triplex 1½-in. circulator in return. Control from air thermostat in series with plug type thermostat immersed in cement slab, air thermostat set at 64F and slab thermostat set at 80F. From a cold start four hours required to heat slab, after which thermostat operates only about once every five hours. In practice owner finds that circulator is operating from four to six hours daily (24 hours). Slab is never allowed to cool during winter. Operating costs reported running sharply lower than similar sized conventionally heated jobs. Architect Harry A. Bruno, Oakland, Calif. System installed by Robert Bruen, Oakland, Calif.

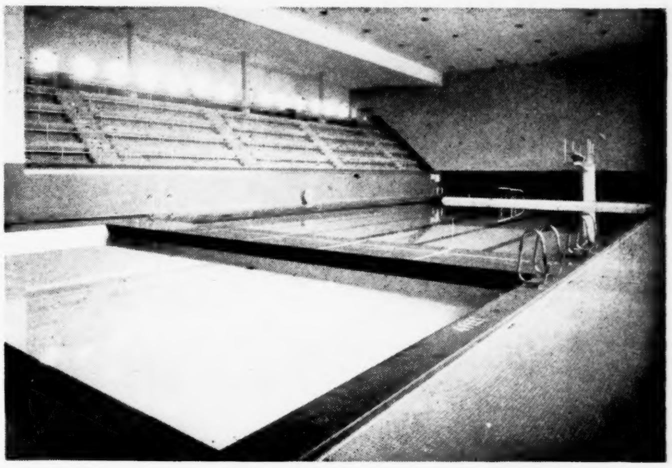




U. S. APPLICATIONS

12 - POOL

ALUMNI SWIMMING POOL, Massachusetts Institute of Technology, Cambridge, Mass. Radiant heat for deck of pool and in ceiling over pool deck. Has an enormous window (90 ft. x 30 ft.) of plate glass. Outside walls of 13 in. of light brick for the exterior, two inches of cork, terra cotta, two layers of asphalt, acoustic plaster and finally ceramic tile six feet high, around the walls. Heating system is a down-feed, forced hot water system for the pool room, the water being heated by steam from M.I.T. central heating plant. Water supply rises to ceiling by a main between the roof trusses. Hot water take-offs from the U-type water tube heater supply pool heating and feed the deck coils, circuits being balanced. Similarly, the panels in the ceiling are heated. Coils for deck and ceiling panel heating were pre-fabricated at the factory, of extra heavy black iron pipe, in ½, ¾ and mainly 1 in. sizes. Deck coils are buried beneath the deck surface, while ceiling coils are suspended from structural steel that supports roof, with separate wire sling for each coil so supported and to which wire lath is fastened. All ceiling coils are set dead level, but those in the deck pitch from the pool level out toward the pool, 3 in. below the finished deck surface. Ceiling coils were back plastered, after plaster was forced around the pipe, and two inches of cork laid on top of each panel. Expansion and contraction of the ceiling was problem, but test panel was tried for cracking, and coefficients of plaster and pipe were found to be nearly the same. Construction was dried out very slowly. In all there is a mile of pipe in the panel heating system, consisting of 30 coils in the ceiling on 6 in. centers and 19 in the deck floor, on 9 in. centers. In providing the radiant heat in floor, structural concrete was depressed 5½ in. and then concrete fill was poured around pipes, securing a good bond and at same time providing separate slab so that should it be necessary to make an opening in floor, or to tear up a part, entire deck would not have to be taken up. Heating coils act as reinforcement for upper slab, but steel bars reinforce lower one. One coil of ceiling panel extends directly over the two spring-boards, while in deck panels a coil is located on either side of these diving boards, with none beneath, for lack of room. Panel temperature is compensated to outside temperature by thermostat in fresh air duct. Structure completed in 1940. Architects, Professors Lawrence B. Anderson and Herbert L. Beckwith. Radiant heating designed by Professor James Holt of M.I.T. A Wolff & Munier, Inc., system.



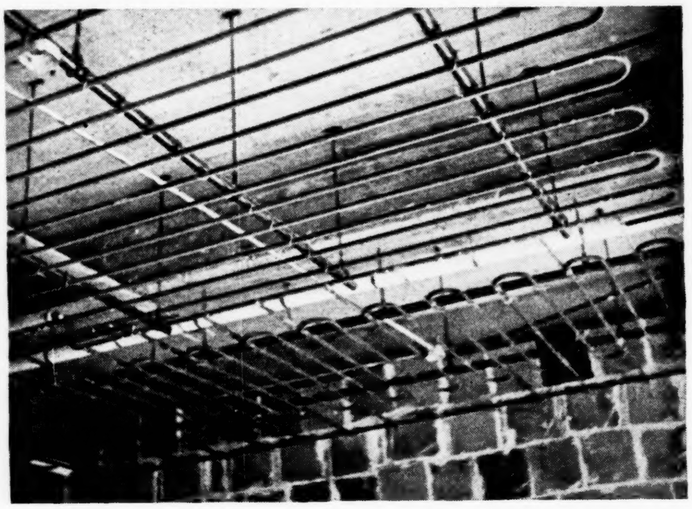
13 - OFFICE

OFFICE EXTENSION, JOHN B. PIERCE LABORATORY OF HYGIENE, New Haven, Conn. The extension to the Pierce Laboratory consists of one floor and basement and is used for offices, conference room and drafting room. Total floor area about 1200 sq. ft. System is largely similar to that in project 70 but instead of allowing the walls to heat their entire height from floor to ceiling, stops were installed at the 6 ft. level so that the walls are not heated above 6 ft. This was done due to the high (11 ft.) ceilings. System designed by E. J. Rodee.

U. S. APPLICATIONS

14 - SCHOOL OFFICE

HIGH SCHOOL, Everett, Wash. Offices of school board located in high school building have radiant heating using hot water from a 4000 gal. storage tank connected to school's boilers. Water supply thermostatically controlled to give a circulating temperature of 115F with a maximum of 130F. Panel system has capacity of about 550,000 B.t.u. per hour. Coils contain approximately 11,000 lineal feet of ½ and ¾ in. pipe on 6 in. centers. System designed by Erwin L. Weber, Seattle, using a Wolff & Munier, Inc. system.



15 - HANGAR

AIRPLANE HANGAR, California. Electrical heating system in concrete floor of airplane hangar. 53,000 ft. of electric wire spaced by 1-in. porcelain insulation tubes in ¾ in. galvanized conduit filled with oil is run across the floor in 300 ft. loops, each 2 ft. in width, and arranged in 12 zones each under thermostatic control.

16 - EMBASSY

BRITISH EMBASSY, Washington, D. C. A 700,000 cu. ft. combined residence and office and consisting of two main buildings joined by a connecting space. One of the earliest and best publicized radiant installations in this country. Installation made early in 1930. (See H & V, May and June, 1930.) Crittall system designed and installed by Wolff & Munier, Inc., New York, using ceiling coils of steel pipe through which hot water is circulated.

17 - HOSPITAL

BLODGETT MEMORIAL HOSPITAL, Grand Rapids, Mich. An operating room radiant heated in 1938. Room is approximately 12 x 17 x 10 ft. high with cold plates on opposite walls and the entire wall and ceiling surface faced with aluminum foil covering.

18 - RESIDENCE

RESIDENCE OF MR. SCHWARTZ, Two Rivers, Wis. Pipe coils were used in the floors on both the first and second floors of this residence designed by Frank Lloyd Wright. Westerlin and Campbell, Chicago, fabricated the coils which consist of over 300 ft. of 2-in. pipe.

19 - RESIDENCE

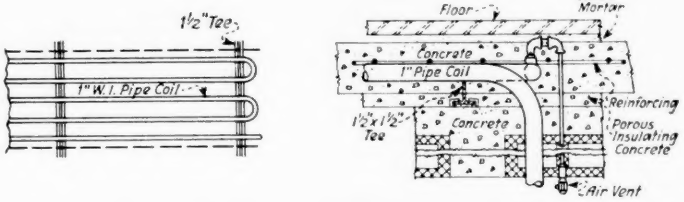
RESIDENCE OF B. L. KILGOUR, Cincinnati, Ohio. This installation, made late in 1940, in a large residence where the glass enclosed porch only is radiant heated, uses 800 ft. of ¾-in. pipe coils in the ceiling. Coils, welded, have provisions for venting. Plaster beneath the coils acts as a radiating surface. Hot water is the heating medium. Residence was designed by Harry Hake and Harry Hake, Jr., architects. Coils fabricated by Peck, Hannaford & Briggs, Cincinnati.

20 - RESIDENCE

MISSOURI RESIDENCE, exact location unknown. A Frank Lloyd Wright project using prefabricated 2-in. wrought iron coils imbedded halfway between top and bottom of a 9-in. coarse gravel fill topped by a reinforced concrete mat and finished floor. Heating medium is 130F water.

21 - CHURCH

THE SACRED HEART CHURCH, Pittsburgh, Pa. One of the early radiant heating systems in the country, installed in 1928. Wrought iron pipe coils laid in floor as shown in accompanying sketch. System designed by Carlton Strong, architect, and Kaiser, Neal and Reid. Coil is of 1-in. pipe; heating medium is hot water.



Heating details in Sacred Heart Church installation.

U. S. APPLICATIONS

22 - RESIDENCE

RESIDENCE OF LLOYD LEWIS, Libertyville, Ill. Installation made during summer of 1940. Hot water is circulated through coils of 3, 2½ and 2-in. pipe. Part of house is two stories; remainder one. Coils run at right angles to wood joists and through them. Each joist in two parts, the split being horizontally through the center; the holes provide for passage of coil. Frank Lloyd Wright was the architect. Pipe coils fabricated by Chicago Nipple Company.

23 - RESIDENCE

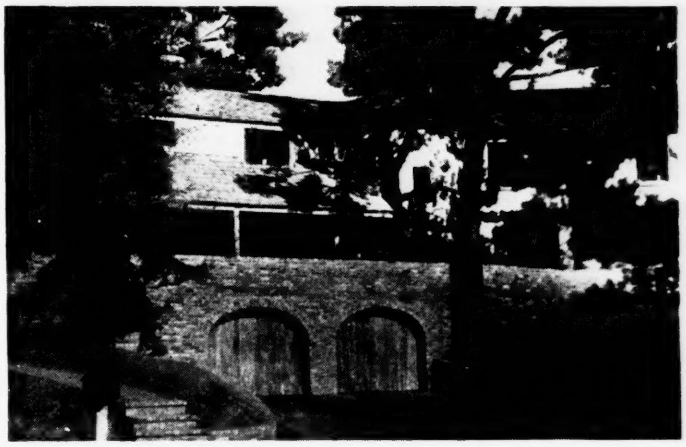
RESIDENCE OF E. B. SULLIVAN, Reading, Pa. System installed in 1939 under the direction of Wolff & Munier, Inc., U. S. licensees for the Crittall system.

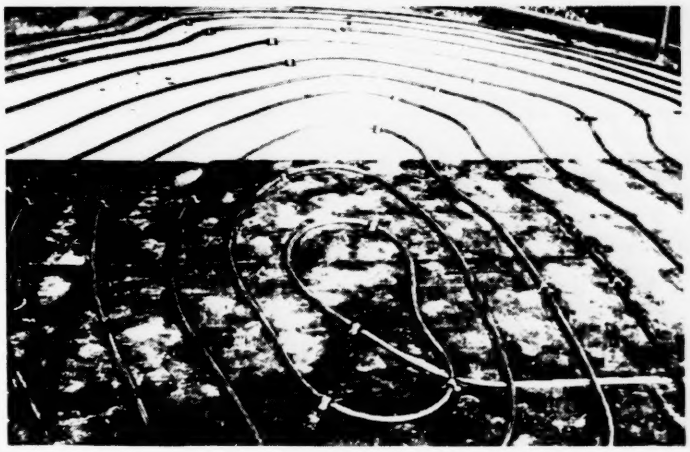
24 - RESIDENCE

RESIDENCE OF D. J. DALZIEL, Waukegan, Ill. System consists of 1¼ in. wrought iron pipe coils laid on a bed of gravel and covered with cement. System designed by Wm. A. Ganster and installed by McDonough Plumbing & Heating Co. Coils fabricated by Crane Company from 600 lineal foot of pipe.

25 - RESIDENCE

RESIDENCE OF H. H. EGGLESTON, Piedmont, Calif. A two-story brick residence, 12 rooms, 3 baths, cost about $35,000. On first floor a 4-in. concrete slab was poured directly on ground. Heating coils of ½ in. type L copper, in spiral formation, averaging 9 in. on centers; flow and return main to each coil individually taken from a header in heater room, where balancing fittings and control valves are conveniently located. Cold asphalt packed between coils to top level, then cement grouting (½ in.) topped with terra cotta tile, filled, waxed, and polished. On second floor, floor consists of planks 2½ in. thick. Coil installation on planks similar to first floor layout. Heater is a 150,000 B.t.u. Big 4 Booster. All other details of control and operation identical with project 11, except plug thermostat omitted, and multi-movement time switch substituted. Operating costs exceptionally low; for instance in December, 1940, $5.49, and in January, 1941, $7.82. Approximately 3300 lineal ft. of ½ in. copper tubing was used. Carr C. Jones, Piedmont, Calif., was the designer and builder; radiant heating system was designed and installed by Robert Bruen, Oakland, Calif.



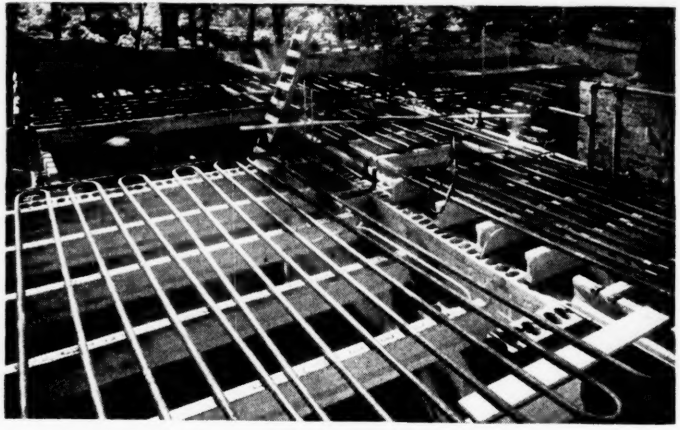


HEATING
AND
VENTILATING
REFERENCE SECTION
No.
1
OF A SERIES

U. S. APPLICATIONS

26 - RESIDENCE

RESIDENCE OF C. M. STANLEY, Muscatine, Iowa. Owner, associated with Stanley Engineering Company, acted as designer of this system for his own home. Hot water is circulated through wrought iron pipe coils. Coils on first floor are 1200 ft. of 2-inch pipe laid on gravel under a concrete floor slab. All joints welded. Coils for the second floor totaled 1200 ft. of 1¼-inch pipe laid on joists as indicated in photograph. Installation by Sanitary Plumbing and Heating Company of Muscatine.



27 - RESIDENCE

RESIDENCE OF P. W. YEAGER, Waukegan, Ill. System, installed about 1940, has coils in space between first floor, ceiling, and second floor; radiant heat in both directions. Supplementing this, wrought iron pipe coils are also laid under the concrete floor slab of one of the bedrooms of the first floor. System designed by the architect, William A Ganster, with William McDonough, heating contractor.

28 - RESIDENCE

RESIDENCE OF DR. MERIT SCOTT, Nittany Village, State College, Pa. System was designed by R. V. Hall, Port Allegheny, Pa., and is apparently similar to system described as project 53.

29 - RESIDENCE

RESIDENCE OF WILLIAM J. KERR, Waterford, Conn. System was designed by R. V. Hall and is apparently similar to system described as project 53.

30 - RESIDENCE

RESIDENCE OF R. A. PETERMAN, Wexford, Pa. System was designed by R. V. Hall and is apparently similar to system described as project 53.

31 - RESIDENCE

RESIDENCE OF WALTER L. KRAHL, Emporium, Pa. System was designed by R. V. Hall and apparently is similar to system described as project 53.

32 - GARAGE

GARAGE FOR CHIEF MOTORS, Olean, N. Y. Installation made in 1940. System designed by R. V. Hall and is apparently similar to system described as project 53.

33 - OFFICE

FACTORY OFFICE, SARCO COMPANY, INC., Bethlehem, Pa., reported by T. Napier Adlam to be radiant heated. In addition, Mr. Adlam states that one or two of his friends in Bethlehem have had partial heating by radiant panels.

34 - OFFICES

YAKIMA COUNTY COURT HOUSE, Yakima, Wash. A 1941 installation of a Wolff & Munier, Inc., system.

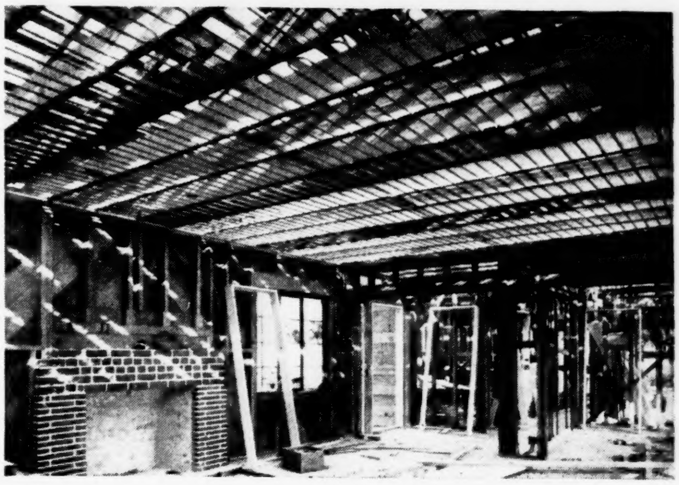
35 - RESIDENCE

RESIDENCE OF ALICE SHINN MINER, Hidden Valley, Contra Costa County, Cal. Building is a one-story frame and rustic dwelling, seven rooms, three baths, cost $10,000. Similar in all respects to project No. 36 except that there are no zone controls, plug thermostats or electric time switches, no floor panels, and insulation consists of 2 in. thickness rock wool batts. Architect Nathan Lindell Coleman, Oakland, Calif. Heating system designed and installed by Robert Bruen, Oakland, Calif.

U. S. APPLICATIONS

36 - RESIDENCE

RESIDENCE OF WM. J. BESLER, Happy Valley, Contra Costa County, Cal. A one-story frame dwelling; stucco exterior, 12 rooms, three baths, cost $17,000. Ceiling panels covering approximately 50% of ceiling area in all rooms and hallways, except bathrooms. Panels designed by Robert Bruen (patent applied for), constructed of ¼ in. type M hard copper tubing, 96 lineal ft. to the panel. Panel measures 4 ft. 6 in. wide by 8 ft. long. Tubing in 4-ft. lengths with U-bends on 5-in. centers. Coils fabricated, then applied to expanded metal lath by patented process. After erection of panels, all ceiling area not panel-covered is conventionally lathed with metal lath. Then the entire ceiling area is plastered with the conventional three coats of plaster, no special mix being used. After "scratch" coat of plaster has set, a very thin (creamy) mix is poured over the panels from the top side, just sufficient to fill up holes between the plaster "keys" insuring that coils are thoroughly imbedded, and providing a fairly smooth upper surface to reduce radiating surface on the top side of panel. Insulation then applied to the top side of all panels. On this job, experimentally, a 1-in. dead air space was left, then a sheet of metal aluminum foil face down secured between joists, then 4 in. thickness rock wool batts, then another 1-in. dead air space, then another layer of aluminum foil. Panels identical to the ceiling panels were laid on the sub-floor, grouting poured on, and ceramic tile laid atop same. In addition, there is an open-air front porch, 14 ft. x 42 ft. roofed, and with cement floor. About 800 lineal ft. of ½ in. Type L copper tubing is imbedded in this cement. Owner expects to grow exotic flowers on this open air porch in mid-winter. Heater is a No. 300 Burkay, 190,000 B.t.u. capacity. Triplex 1½ in. circulator. Job is zone-controlled, living room, dining room, etc., being on one zone; bedrooms on the other zone. B & G motorized valves, controlled by air and plug thermostats (immersed in plaster of ceiling) in series. Architect, B. Reede Hardman, Berkeley, Calif., heating system designed and installed by Robert Bruen, Oakland, Calif.



37 - RESIDENCE

RESIDENCE OF DR. FLORIAN HEISER, Storrs, Conn. A wall heating radiant system using hot water instead of steam but otherwise similar to project 70. System designed by E. J. Rodee. System has been in operation for three winters.

38 - RESIDENCE

RESIDENCE OF DR. C. H. WALLACE, Storrs, Conn. Building is a modern type log cabin. Has a floor heating system using forced hot water circulating through ¾ in. steel pipe with threaded connections imbedded just below the surface in the concrete floor. Building has been occupied since the fall of 1940 and occupants report very comfortable conditions. In order to compensate for the high heat loss from side walls, windows and roof, rather high floor temperatures are necessary—about 90F at outer edges of the room and 85F near center. There is no measurable air temperature difference between 6 in. above the floor and peak of roof which is 17 ft. above floor. System designed by E. J. Rodee.

39 - RESIDENCE

RESIDENCE NEAR HARTFORD, Conn. A wall heating system similar to that described in project 70. System now in process of installation. System designed by E. J. Rodee.

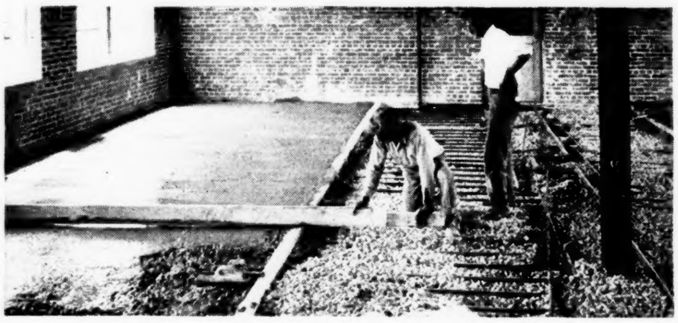
40 - RESIDENCE

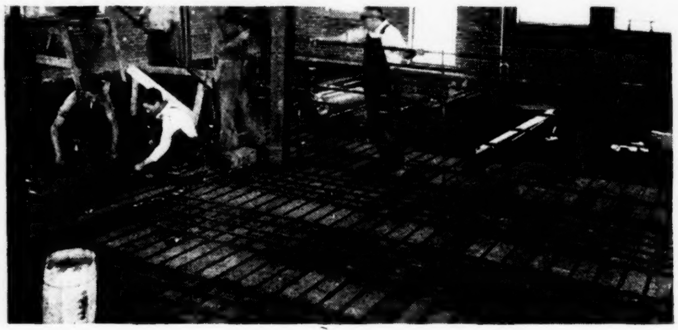
RESIDENCE OF CARL F. BOESTER, St. Louis, Mo. House heated and cooled by an unusual air conditioning system but panels for radiant heating and cooling are installed in the ceiling and walls of some rooms and the walls of the two bathrooms. System designed by the owner.

U. S. APPLICATIONS

41 - OFFICE

OFFICE OF GREENVILLE STEEL & FOUNDRY COMPANY, Greenville, S. C. Building 40 x 50 ft. of 2 stories. Hot water circulated by ¼ hp. pumps flows through ½ and ¾ in. wrought iron pipe coils. Broken stone fill was laid on ground and coils laid on this (see photo at right). Additional stone was placed around coils and 2-in. concrete slab poured to form floor. Pipe slightly pitched to provide for drainage. On second floor, pipe coils laid over cork insulation (see lower photo) supported by steel beams. Sarco control equipment was used with an indoor-outdoor thermostat controlling water temperature. System designed by T. Napier Adlam, of Sarco Company, Inc., in collaboration with H. G. Faust of Crane Company.





42 - RESIDENCE

RESIDENCE OF R. J. SULLIVAN, Redding, Conn. A ceiling coil job with insulation between the joists, wire lath and plaster beneath coils and in contact with them. A. E. Johnson, New York engineer, designed the system.

43 - RESIDENCE

RESIDENCE OF JUDGE DECHANT, Lebanon, Ohio. A one-story house with a number of wings and large group windows. Heating medium is steam provided by oil-fired boiler under control of room thermostat. Coils are laid on gravel over which a 4-in. reinforced concrete slab was poured and floated. System may be switched to hot water. Owner reports an advantage of system is its cleanliness. Albert L. Harmon, architect.

44 - FACTORY

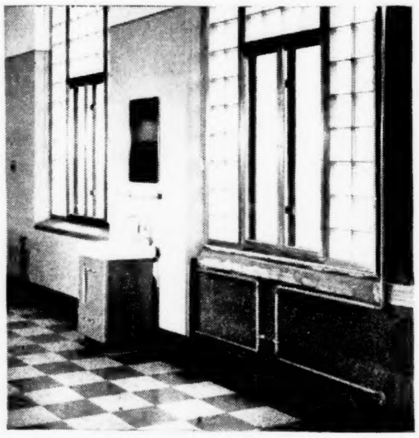
INDUSTRIAL BUILDING, Detroit, Mich. An experimental installation by Detroit Edison Company using electric radiant heaters. An identical room was heated by conventional steam radiators. Results show a great reduction in temperatures near the ceiling with the radiant heating system.

45 - RESIDENCE

RESIDENCE, Seattle, Wash. A large home using about 12,000 ft. of ½ and ¾ in. pipe suspended from ceiling and also in floor slabs. Installation designed by Erwin L. Weber, consulting engineer, Seattle.

46 - OFFICES

BANKERS LIFE CO., Des Moines, Ia. Modern office building of six main floors, two basements and one penthouse. Masonry walls insulated by 2 in. of cork. One inch of bare copper piping loops around and under the windows carrying hot water for forced circulation. Sectional steel wall panels are used and act as radiating surface. Wall temperatures are maintained at about 70F. Results indicate widest temperature variation of wall from floor to ceiling as 3F. Architects, Tinsley, McBroom & Higgins, Des Moines; heating and air conditioning designed by Charles S. Leopold, consulting engineer of Philadelphia. In summer walls are used as radiant cooling surface. Photo shows panels removed.



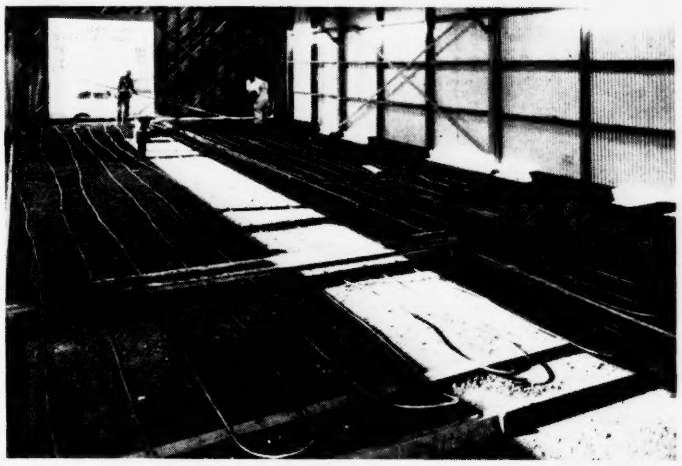
U. S. APPLICATIONS

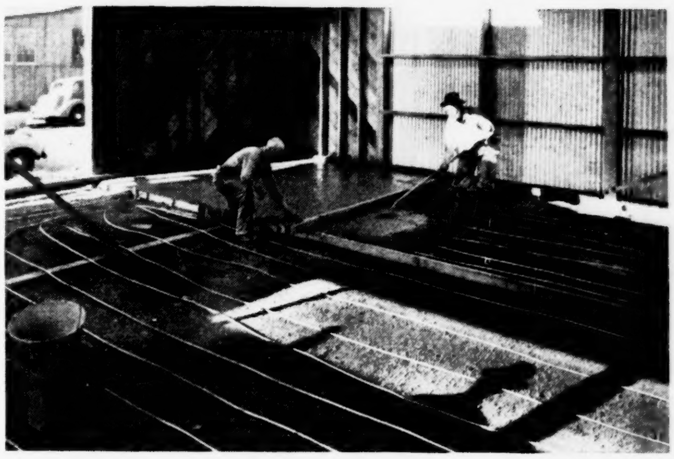
47 - RESIDENCE

RESIDENCE OF DR. C. W. SONDERN, Kansas City, Mo. Pipe coils of wrought iron laid on broken stone under a concrete floor slab using hot water as the heating medium. Frank Lloyd Wright, architect; coils fabricated by the Chicago Nipple Manufacturing Company.

48 - METAL SHOP

MELROSE SHEET METAL WORKS, Shop Building, Oakland, Calif. A corrugated galvanized building about 160 ft. sq. with high ceiling, numerous skylights with ventilators and concrete floor. Proprietor George Lauer requested Robert Bruen, Oakland, Calif., engineer and contractor, to lay out a low cost heating installation to take chill off the shop building. Designer suggested copper tubing in coils approximately 12 x 12 ft. square, but owner laid tubing in long bends, more than 100 ft. to each lateral, on about 2 ft. centers. Coil, ½ in. type L copper, was laid out and tested on the ground, is embedded in the cement about 2 in. below the surface. Due to extreme length of each coil, frictional resistance, etc., it took about 72 hours of constant operation before there was noticeable warmth to the concrete slab. The plant now operates continuously day and night with 100F water circulating through the coils. Slab temperature has not been tested, but is probably averaging 70 to 74F. During coldest winter weather this year plant was comfortable as any heated house. Union mechanics working there have publicized the job through their union, with result that the job has had considerable local publicity, and has been inspected by dozens of factory superintendents. Two additional plants are planning on equipping their plants similarly this year.





HEATING AND VENTILATING
REFERENCE SECTION
No. 1
OF A SERIES

U. S. APPLICATIONS

49 - HOSPITAL

METROPOLITAN STATE HOSPITAL, Waltham, Mass. Tubercular wing, top floor only, radiant heated in 1935, operating suite in 1932. Cost of ceiling radiant heating for these top floor wings was $20,000. Lower two floors heated with forced warm air costing $71,000. A Wolff & Munier, Inc., system.

50 - RESIDENCE

RESIDENCE OF PIERRE S. DuPONT, III, Rockland, Del. A large home insulated with 2 in. of cork and heated with pipe coils in hung ceiling. Steam, generated for other uses in house, is used to heat water which is circulated at temperatures from 100 to 110F. A 1940 installation of a Wolff & Munier, Inc., system designed by Jaros, Baum & Bolles, New York.

51 - RESIDENCE

RESIDENCE OF JOHN BALTZ, Larchmont, N. Y. System installed in 1936 by Wolff & Munier, Inc.

52 - RESIDENCE

RESIDENCE OF JAMES LAWRENCE, Brookline, Mass. System installed in 1938 under the direction of Wolff & Munier, Inc.

53 - RESIDENCE

RESIDENCE OF CLIFFORD M. LEWIS, Boalsburg, Pa. Installation, completed in 1939, consists of floor coils, fabricated from 1½ in. wrought iron pipe and using steam as a heating medium. R. V. Hall, Port Allegany, Pa., was the architect for the house which is a one-story affair. Coils were apparently laid on gravel with concrete on top.

54 - RESIDENCE

RESIDENCE OF MR. ROSENBAUM, Florence, Ala. System is forced hot water heated by electricity. Prefabricated coils are laid on gravel with a cement slab on top. Approximately one-quarter of the installation's cost was for pipe.

55 - RESIDENCE

RESIDENCE OF DENISON B. HULL, Wilmette, Ill. A large house insulated with cork. Believed to have been built about four years ago. Hot water is medium used, maintained at 120F. System was designed by Irving E. Brooke, consulting engineer of Chicago and the owner, who is an architect. Owner reports radiant heating quite economical.

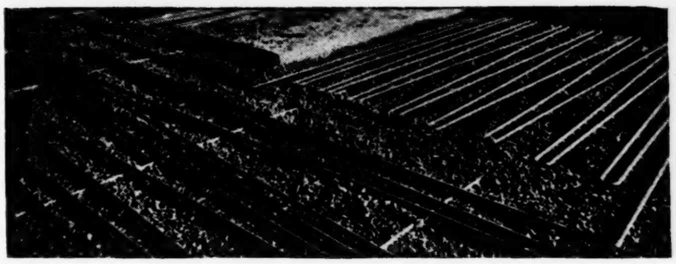
56 - OFFICES

S. C. JOHNSON & SON, INC., Office Building, Racine, Wis. A floor heating system using steam as the heating medium due to its availability from company's power plant. Steam is controlled by a sub-atmospheric system. Pipe coils were covered with coarse gravel about 5 in. thick below the pipes and 3 in. above. A 6-in. concrete slab was poured over the gravel and finished with felt and rubber tile. System divided into five zones, each with two controls to maintain a slab temperature of 60 to 75F. Architect, Frank Lloyd Wright. Heating system designed by Victor Walters, Westerlin and Campbell Company, Chicago, in collaboration with Mr. Wright.

U. S. APPLICATIONS

57 - RESIDENCE

RESIDENCE OF HUDSON TALBOT, St. Louis, Mo. One-story brick residence. Eaves designed to admit winter sunlight, keep out summer sun. House heated by copper pipe coils laid under concrete floor. E. J. Mutrux, architect.



58 - RESIDENCE

RESIDENCE OF PAUL VOIGT, Jeffersonville, Ind. Installed in 1939 under the direction of Wolff & Munier, Inc.

59 - TEST ROOM

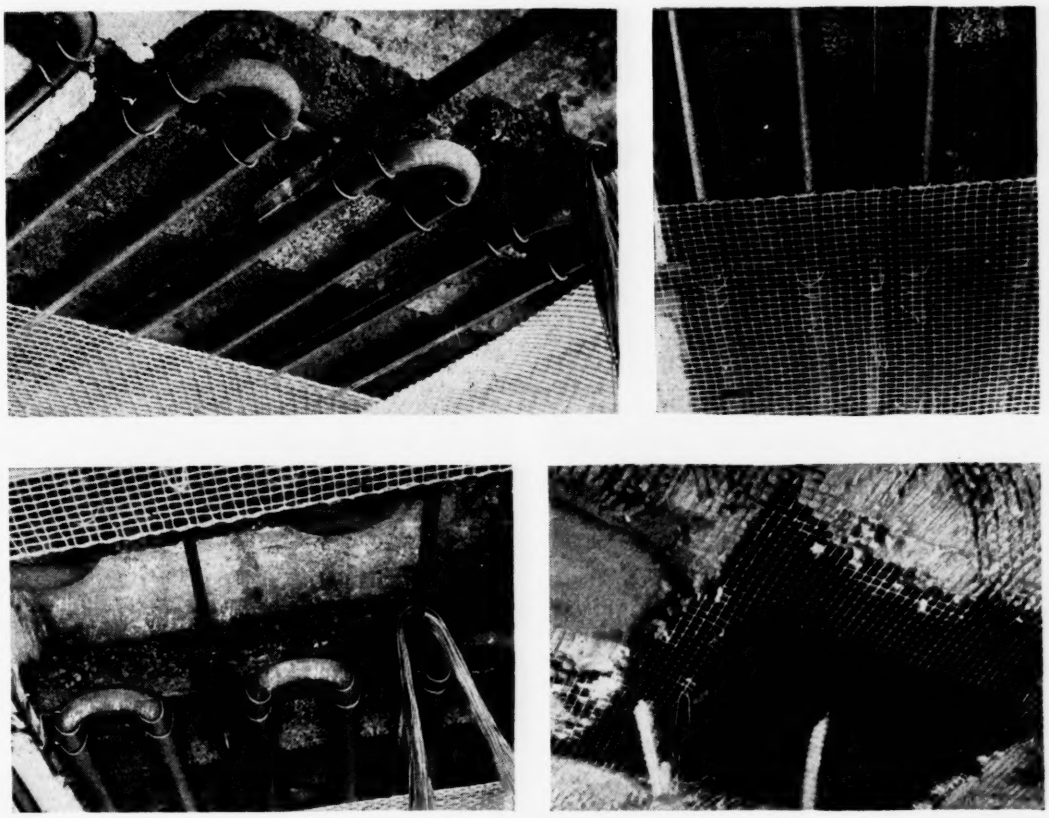
EXPERIMENTAL ROOM, University of California, Berkeley, Calif. A test set-up for conducting research on radiant heating is under way at this University under the direction of Prof. B. F. Raber and F. W. Hutchinson and in collaboration with the ASHVE. Room is heated by Dulrae (electrical) system. Results anticipated next summer but further information not available at this time.

60 - RESIDENCE

RESIDENCE OF THE MISSES NOTZ, Mifflin Township, near Pittsburgh, Pa. A one-story frame and brick house with five rooms and bath, no basement. System, completed during the summer of 1940, consists of wrought iron pipe coils laid in gravel over which a concrete slab was poured. Water heated by gas boiler to 130F. Floor maintained at 80 to 85F with room temperature at about 65F. Field welding to complete assembly was handled by welder and his helper in under two days. Coils fabricated at the Cleveland plant of York Ice Machinery Corp. System designed by Frank Lloyd Wright and Associates, architect, and Edward Brown, Jr., associate architect.

61 - RESIDENCE

RESIDENCE OF MRS. WILLIAM GOODBY LOEW, New York, N. Y. Radiant heat is provided to carry about 40% of the total load, remainder being steam heated with direct radiators. Residence is four stories, about 80 x 100 ft. and uses ¾ in. extra heavy steel pipe. This was the second Crittall system installed in this country by Wolff & Munier, Inc., and was made in the early spring of 1932. Heating medium hot water at 135F maximum. Architect, Walker & Gillette; heating engineers, Jaros & Baum. (See H & V, April, 1932).

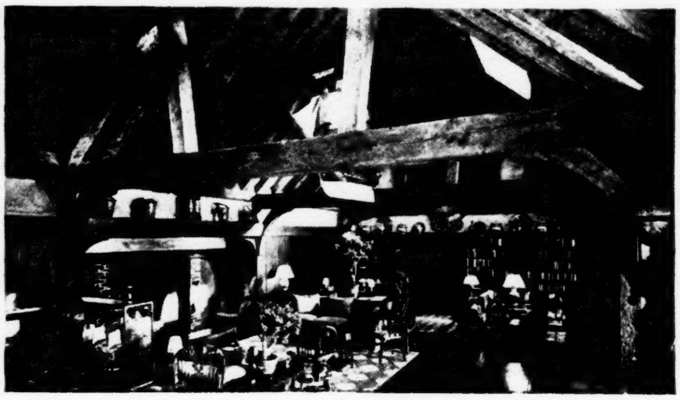


Close-up of Loew residence installation during construction. (Upper left) Coils attached to ceiling. (Upper right) Note 2-in. cork insulation above coils. (Lower left) View before plastering. (Lower right) Showing rough plastering applied.

U. S. APPLICATIONS

62-RESIDENCE

RESIDENCE OF COL. H. H. ROGERS, Port of Missing Men, Southampton, L. I., N. Y. Installation made in 1934. In certain portions, installers faced problem of simulating old beams. (See accompanying photograph.) This was done by making plaster casts of the original beams. Installation under direction of Wolff & Munier, Inc., U. S. licensees for Crittall system.



63-HOSPITAL

PHIPPS PSYCHIATRIC CLINIC, Johns Hopkins Hospital, Baltimore, Md. Project reported but no information available.

64-RESIDENCE

RESIDENCE OF MR. ROSEBROOK, Alamo, Contra Costa County, Calif. A two-story brick residence, five rooms, two baths with floor and ceiling radiant heating now being installed. Lower floor is an old existing stone house of a large room with kitchenette and bath. New brick and stone structure being erected above. Lower floor will later be used as guest quarters. Ceiling panels as described in project 36 will be applied directly to the old plastered ceiling with no insulation above and with new plaster applied over the panels. In the new portion of the house, second floor, floor panels, as described in project 25, will be laid with tile topping. Designer and builder, Carr C. Jones, Berkeley, Calif. Heating system designed and installed by Robert Bruen, Oakland, Calif.

65-LIBRARY

CONNECTICUT STATE COLLEGE FOR WOMEN, New London, Conn. The space heated consists of reading spaces in the book stacks of the library which are heated by hot water panels. The stacks themselves are heated by forced air. System designed by Hubbard, Rickerd & Blakely, consulting engineers of Boston and New Haven.

66-RESIDENCE

RESIDENCE OF L. WINKLE, Scarsdale, N. Y. A 1940 installation of Wolff & Munier, Inc., U. S. licensees for the Crittall system.

67-RESIDENCE

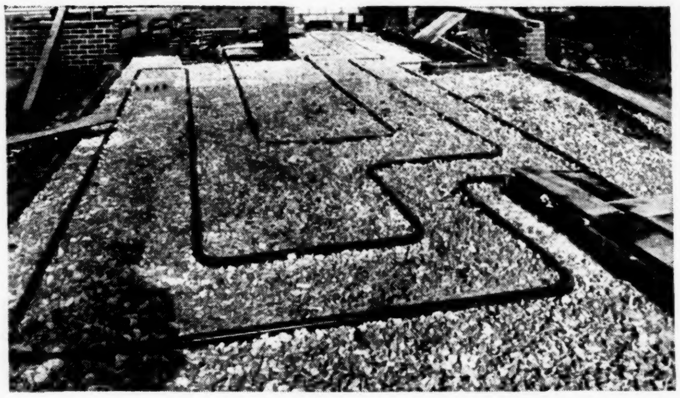
RESIDENCE OF ANDREW ARMSTRONG, Gary, Ind. Two-in. pipe made up into coils was laid in sandy floor. Four inches of sand was packed over the coils and a 2-in. concrete mat was poured. Finished floor, supported by this mat, consisted of 4-in. thick concrete slabs in 4-ft. wall sections. Heating medium is hot water circulated through the pipe coils. System designed by Frank Lloyd Wright with Edward J. Munkhoff, Gary, as heating contractor. Coils consist of 2-in. pipe, welded.

68-SCHOOL

CLOVER PARK HIGH SCHOOL, Tacoma, Wash. A 1940 installation of a Wolff & Munier, Inc., system.

69-RESIDENCE

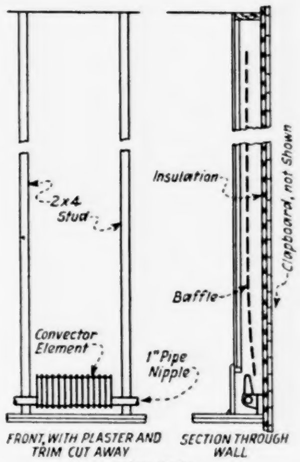
RESIDENCE OF JOSEPH EUCHTMAN, Baltimore, Md. Six-room one-story house. Five hundred feet of pipe was laid on gravel over which concrete floor was poured. Frank Lloyd Wright, architect and R. H. Bosman Bros., Inc., of Baltimore, fabricated the coils. See accompanying photo.



U. S. APPLICATIONS

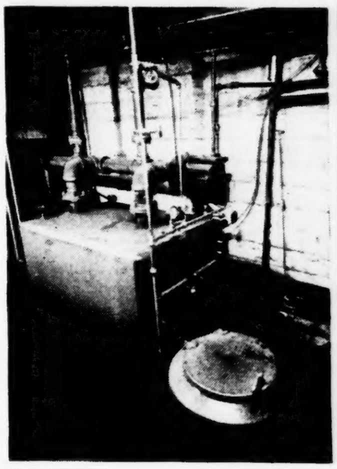
70 - RESIDENCE

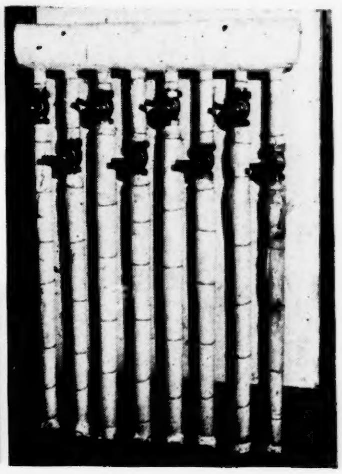
RESIDENCE OF E. J. RODEE, West Haven, Conn. Owner, engineer of John B. Pierce Laboratory of Hygiene, designed own system which is of the wall type with heated air in the stud space of either outside or inside walls; air in turn heats the plaster which becomes the radiating surface. Cast-iron convector elements set between the studs and are so arranged that by removing the base-board the heating sections can be removed without disturbing the plaster. Baffle is installed within the stud space behind and above the convector through which steam circulates (see sketch) to separate the upward and downward currents of air and causes the air to circulate. Cold air travel is on the outside edge of the baffle. Owner reported operating results of this system in recent ASHVE paper in which he concluded that the system can be installed without increase in wall thickness, without more than a normal amount of insulation, and that the results will compare favorably with free standing radiators from the standpoint of operating costs. Temperature difference between floor and ceiling is low.



71 - CHURCH

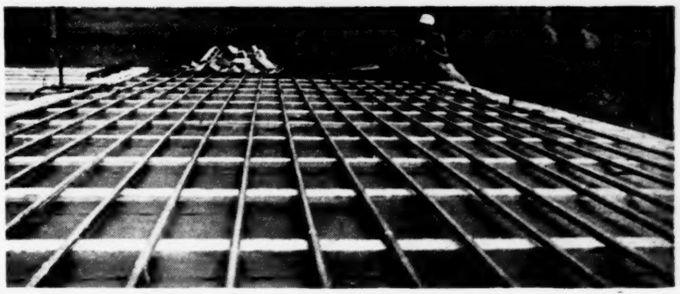
ST. MARY'S CHURCH, Wytheville, Va. System installed about 1939. Fifteen pound steam is the heating medium. Eight supply lines, separately valved, run from central control point in the sacristy to supply headers along one side of the building under the floor. Return headers run close to the supply headers and five coils of 1¼-in. pipe run from each supply header across the church between the open-work floor joint and back to the return headers. An unusual feature of this installation is that the coils are laid on edge, with supply at the top. Wire lath is laid under the floor joists and reenforcing bars under the lath. Floor is divided into 8-ft. sections for expansion. Floor consists of a 2-in. concrete slab poured over the lath. Result is that the steam coils warm air between the joists and this air warms the concrete floor. The warm air is circulated from grilles at the sanctuary steps at one end of the building and between wall studs to grilles at the far end of the church. Photos show boiler and supply header. System designed by Father Michiel, Belmont, N. C. Installed by E. Y. Spraker, Wytheville.





72 - RESIDENCE

RESIDENCE OF L. C. STRYER, Emmaus, Pa. Owner is contractor and has built several houses equipped with radiant heating. All are of concrete and masonry block construction. An oil burning forced circulation hot water system is used in connection with coils fabricated with ¾-in. pipe on 6-in. centers at right angles to the concrete joists and set in a 4-in. concrete floor slab. Owner reports temperature differences of only 2F between floor and ceiling and reports system economical and comfortable.



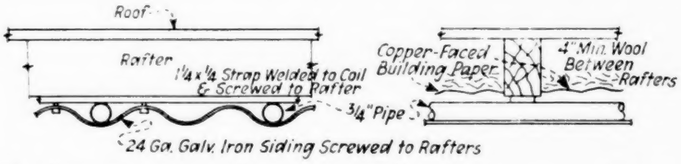
73 - RESIDENCES

FIVE RESIDENCES, Emmaus, Pa. Approximately five other residences in addition to his own and built by L. C. Stryer whose house is described as project 72, are reported to have radiant heating.

U. S. APPLICATIONS

74 - RESIDENCE

RESIDENCE OF J. H. HANSEN, Redding, Conn. System installed late in 1939. Forced hot water system, using Taco specialties with black iron pipe coils in both floor and ceiling, as shown in accompanying sketch. Owner reports this winter, with temperatures as low as 20F, and with both coils operating, that a 60F setting of thermostat is favorable. System designed by H. M. Wright.



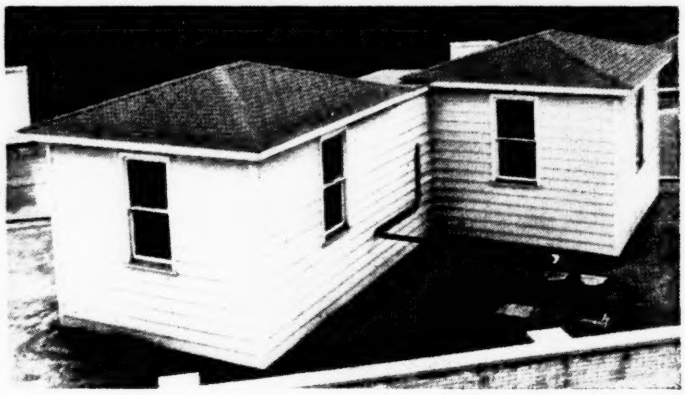
Sections showing ceiling construction in Hansen residence

75 - APARTMENT

SUNTOP HOMES, Ardmore, Pa. A modern four-family project built in separate units. Frank Lloyd Wright, architect. Steam generated in the basement is circulated through 175 ft. of coils laid in the concrete floor of the living room, carport (garage), and storeroom.

76 - EXPERIMENTAL

ASHVE RESEARCH LABORATORY, Pittsburgh, Pa. At Research Laboratory of American Society of Heating and Ventilating Engineers two rooms are being used for studies to determine effect of radiation in comfort. These rooms, similarly oriented, are constructed of materials simulating average residential application. In addition to being heated by radiators and convectors, heat panels have been installed in the ceiling of one room. Panels are of continuous wrought iron pipe and the plaster is applied to prevent cracking and at the same time obtain greatest efficiency of heat output from embedded pipes. Insulation is applied to ceiling.



77 - RESIDENCE

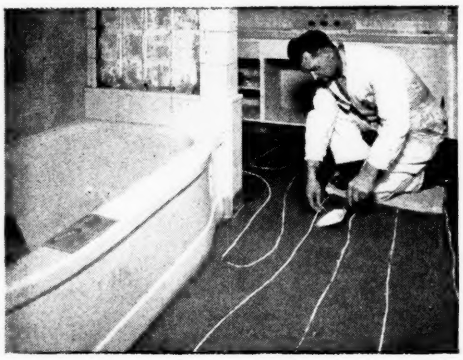
RESIDENCE OF MR. KRESLING, Laporte, Ind. A floor coil installation, using screwed fittings. Hot water is heating medium, installation being a Hoffman hot water controlled heat job. Pipe coil is laid in the concrete floor and the water temperature is controlled from an outdoor temperature bulb. System designed by Samuel R. Lewis, consulting engineer, Chicago.

78 - EXPERIMENTAL

RESIDENCE IN BOISE, Idaho. An installation of late 1938 or early 1939 and in which reverse cycle refrigeration is combined with radiant heating. The house was especially designed to test the system. Small copper tubing is imbedded in the plaster walls of heated spaces. Walls and ceiling are insulated with 3 in. of mineral wool and one ply of aluminum foil paper facing the interior with a heat transmission of 0.10 per hr. per sq. ft. per deg. F. The copper tubing was mounted on the face of the lath in an irregular curved fashion. At last report system was to have been used for radiant cooling in the summer. System designed by C. E. Boggs.

79 - MODEL HOME

MATCHLESS HOME, Los Angeles, Calif. Building is modern house maintained by the Electrical Development League of Southern California, built during 1939. The bathroom is electrically heated by 160 ft. of soil heating type electrical coils. System was installed as a novelty but proved practical. Current is controlled from a switch on the bathroom wall or from a wall switch beside the bed in adjoining bedroom. Purpose was to make the tile floor comfortable for bare feet. System designed by W. W. Pearce, Los Angeles Bureau of Power & Light.



U. S. APPLICATIONS

80 -SCHOOLROOM

ARSENAL HIGH SCHOOL, Washington Crossing, Pittsburgh, Pa. Installation, eight or nine years old, in a room of the school which is especially provided for sick and undernourished children. Reported as highly successful. Tunnel under rooms for this department contains steam coils for steam and return lines, 2½ lb. steam being provided from school boilers. Pipe is accessible in tunnel and not imbedded in concrete. Floor temperatures maintained at 53 to 56F.

81 - HOSPITAL

EIEL CLINIC, Osage, Iowa. Consists of continuous sinuous coils laid in gravel. After the coils were in place a concrete slab for the floor was poured on top. System was designed by Robert Blandon, architect, while O. D. Kingsbury, heating contractor at Osage, installed the system.

82 - RESIDENCE

RESIDENCE OF PROF. THEODORE BAIRD, Amherst, Mass. A nine-room one-story residence with wrought iron pipe coils laid in gravel under a slab of concrete forming the floor. Provision was made for draining. Coils were fabricated by Westerlin and Campbell, Chicago, from 430 ft. of 2-in. pipe; 30 field welds were made to complete the installation. System was designed by Victor Walter, engineer, Westerlin and Campbell, and Frank Lloyd Wright, architect.

83 - RESIDENCE

RESIDENCE OF WILLIAM A. GANSTER, Waukegan, Ill. Owner was the architect of this house completed late in 1940. Coils are of 900 ft. of 1-in. pipe welded. Provision is made for venting and draining. Heating medium is hot water; expansion tank provided. Coils laid under floor. McDonough Plumbing and Heating Co., Chicago, fabricated coils and made installation.

84 - GARAGE

SPECIALIZED SERVICE, INC., GARAGE, Klamath Falls, Ore. Unusual installation since heating medium, hot water, is supplied from underground hot water available in that vicinity. Water temperature in system is nearly 120F. Approximately 9000 ft. of pipe coils used in floor of garage area and in ceiling of office portion. Howard R. Perrin, architect; heating system designed by Erwin L. Weber, consulting engineer of Seattle, and installed by Lorenz Company under the direction of Wolff & Munier, Inc.

85 - 97 (PROPOSED)

FOURTEEN PROJECTS with radiant heating are, at last available reports, proposed or under way: (85) Eight residences at E. Lansing, Mich., with floor heating, under direction of Frank Lloyd Wright; (86) Five 2-story houses in Cambridge, Mass., Carl Koch, architect; (87) Residence of Robert L. Parkinson, Columbus, O., to use 1¼-in. pipe coils on gravel under 6 in. of concrete; (88) Three residences in California, radiant heating designed by Robert Bruen; (89) Radiant heating may be used in 4-story concrete dormitory for University of California; (90) Residence of J. G. Rideout, Cleveland, O.; (91) Residence of J. F. Kern, Jr., Massapequa, N. Y., hot water coils in ceiling of first floor to heat both floors of seven-room insulated frame house; (92) Church in Kansas City, Mo., a $200,000 project, Frank Lloyd Wright, architect, radiant heating and cooling; (93) Residence of Otto Trzos, Pontiac, Mich., radiant heated bathroom and entry, designed by owner, a heating engineer; (94) Residence of George J. Sheers, Camden, N. J., to have hot water coils laid on rock; (95) Residence of Miss Winkle, East Lansing, Mich.; (96) Project to consist of two apartments and perhaps four low cost dwellings, Youngstown, O., F. P. Villani, owner; (97) High School for Consolidated School District, Pleasantville, Iowa.

Grateful acknowledgment is made to the following individuals and organizations for their cooperation in furnishing information and/or illustrations for this section as follows (number is that of project): Robert Bruen, heating engineer and contractor, Oakland, Calif. (11, 25, 35, 36, 48, 64, 88); Copper & Brass Research Association, New York (11, 25, 57); George B. Cushing and J. B. Fullman, A. M. Byers Co., Pittsburgh (1, 8, 18 to 22, 24, 26 to 32, 41, 43, 47, 53 to 55, 57, 60, 67, 69, 71 to 73, 75, 80 to 83, 85 to 87, 90, 92 to 96); H. F. Randolph and J. B. Wallace, International Heater Co., Utica (3, 7); Professor James Holt, Massachusetts Institute of Technology (12); Erwin L. Weber, consulting engineer, Seattle (14, 45, 84); Bankers Life Co., Des Moines (46); Leon L. Munier, Wolff & Munier, Inc., New York (10, 23, 34, 49, 50 to 52, 58, 62, 66, 68); York Ice Machinery Corp., York, Pa. (56); John James, American Society of Heating and Ventilating Engineers, New York (76); T. Napier Adlam, Sarco Co., Inc., New York (33, 41, 73), and E. J. Rodee, John B. Pierce Laboratory of Hygiene, New Haven (13, 37 to 39, 70).

RADIANT HEATING IN ENGLAND

By F. W. HUTCHINSON†

The author here discusses low temperature radiant heating particularly as applied to England, which he visited on extensive tours during which he thoroughly investigated present practice in radiant heating in that country. At present the author, together with Professor B. F. Raber, is at work on a comprehensive program of panel heating research at the University of California.

Low temperature panel warming is a special form of heating by direct transfer of radiant energy. As such it is but one of many methods of radiant heating, among which are the conventional type of electric reflecting "spot" heater and the open fireplace. Like all other methods of radiant heating, the panel system operates on the principle of transferring energy directly from the source to the person or object acting as the receiver; unlike most other radiant methods, the low temperature panel system is designed not only to supply the requisite quantity of energy, but to supply it at an intensity and with a distribution such that the occupants will experience comfort.

The low temperature panel method offers a solution to the two basic problems of radiant heating. The first problem, intensity of irradiation, defied solution so long as man was obliged to use an open fire as the direct source of energy. The amount of radiant energy leaving a source is dependent on the temperature and on the emissivity (surface characteristic) of that source; the range over which control of either of these variables is possible is, in the case of an open fire, so small that effective control is practically impossible. Adjustment of the energy absorption rate at the receiver is likewise difficult and is further complicated by the fact that with humans as receivers the control exercised by the heating engineer over the variables (as texture, color, and character of clothing) is reduced almost to zero. It is evident, then, that the open fire should not itself be the radiating source, but should supply energy to some other device, radiation from which would be subject to more rigorous control. A panel is such a device. The quantity of energy which leaves its surface is a function of the surface temperature and this, in turn, is dependent on both the rate of flow and the temperature of the circulating fluid. Thus the energy leaving the panel is readily subject to control and the intensity of radiation from such a source need not approach a value that would be objectionable.

Solution of the second problem, distribution, follows directly from the use of lower radiating temperatures. With high temperature radiation the requisite total heating effect is obtained by irradiation from a source of small area. In such a case it would not be possible to correct the poor distribution by placing a second source behind the occupant as to do so would result in a heating effect too great for comfort. English tests show that a person will be comfortable in front of a fire with the "front effective temperature" 70F and the "back effective temperature" 60F, but will become uncomfortably warm if the back temperature is permitted to rise even to 67F. Low temperature panels correct this difficulty by providing irradiation at lower intensity from a larger area and from many directions. The total quantity of energy necessary for comfort is practically the same as with an open fire, but its intensity is much less.

The term "panel heating" is usually restricted to those installations in which the greater fraction of the heating effect is realized by direct irradiation of the occupants rather than by the heating of the air in the occupied space. If this limitation in nomenclature is accepted it immediately follows that wall and floor installations do not constitute true panel warming since, in such cases, pronounced air movement is set up (due to the tendency of warm air to rise) and the sweep of cold air across the panels increases the film transfer coefficient with a consequent marked increase in the fraction of heat dissipated from the panels by convection.

HEATING
AND
VENTILATING
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1
OF A SERIES

Except in very unusual circumstances the characteristics, and particularly the economies, of panel heating will be realized only where the panels are constructed in, or installed flush with, the ceiling surface. This statement does not imply that the use of wall or floor panels is either incorrect or inadvisable. It means only that panels installed in this way should be designed as part of a convective heating system and must be expected to perform in such a way that the comfort requirements of convective rather than radiant heating will be realized. When used on this basis, wall and floor panels often meet a genuine need and provide an excellent solution to certain special problems of space heating. Floor panels are sometimes used in entrances of ceiling heated structures to temper the infiltration.

PRINCIPLES OF RADIATION

What is radiant heating and how does it differ from heating by convection? In the first place, there is no such thing as a radiant heat. A heated object gives off heat by convection to the air surrounding it, but it also emits energy—not heat—which travels with the velocity of light from the surface of the heated object in all directions; some of this energy strikes the receiver and is absorbed at its surface (or beyond). The characteristics of radiant energy are a function of its wave length which, in turn, is a function of the temperature of the radiating surface and the material of which it is constituted. Heat, light, sunburn, photoelectric response, are all special effects which may or may not result from the absorption of radiant energy, depending upon the wave length of the radiation and the response characteristic of the receiving surface. Since the nature of all radiant energy, regardless of wave length, is the same it follows that the equations used by the illuminating engineer to investigate the uniformity of lighting are equally valid when used by the heating engineer to investigate the uniformity of radiant heating.

The analogy between lighting and radiant heating permits of a number of comparisons which may assist in visualizing the radiant process. The low temperature panel system, for example, is to an open fire as diffused indirect lighting is to a single, exposed, high intensity lamp. The dark spots, shadows, points of excessive light or glare found in a room with

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†Instructor, Department of Mechanical Engineering, University of California.

improperly designed lighting all have their counterparts in a room heated by poorly distributed panels of the high temperature type. With lighting, proper selection of colors and wall surfaces will permit maintenance of a higher lighting level with reduced power input; with heating, a careful choice of wall surfaces and colors (where such a choice is aesthetically or architecturally permissible) will reduce the necessary energy input of the panel heating system.

So, also, as with lighting, the panel system readily lends itself to the establishment of a relatively low radiant flux throughout the heated space with provision for supplying additional energy at the particular points of use. Another similarity which might be expected would be an immediacy of response following the turning on or off of the source of energy. This would be realized if the thermal capacity of the panels were sufficiently low and the reflectivity of other inside surfaces sufficiently high, but in practice, with existing materials and methods of construction, the response of a panel system is surprisingly low, often less than that of a heating plant of the conventional type. In respect both to thermal capacity and to integration of design with selected wall surfaces, the panel system offers possibilities for economy of operation and flexibility of control which are at present largely theoretical, but give promise of practical development that may far surpass present methods of central heating.

CONVECTION vs. RADIATION

Since radiant heating endeavors to supply heat directly to the occupants, the design of such a system presupposes a knowledge of the physiological basis of comfort heating. This subject will now be reviewed briefly.

Physiological Basis of Comfort Heating. The human body operates essentially as a heat engine. Its thermal efficiency is relatively low when compared with the efficiencies of certain man-made engines, but if the comparison takes into account the amazingly small temperature range over which the body operates it then becomes evident that the human engine more effectively performs its task than do most other heat machines. As is true of all other engines, so also with the human body does efficiency vary with load. Fortunately for heating engineers, however, the work necessary to sustain life—that is, energy required by a passive individual whose processes of digestion and thinking are taking place at an average rate and without evidence of accomplishing external work—is not subject to wide variation among individuals; neither does the thermal efficiency at *idling load* vary greatly. In consequence, the energy input (heat supplied) and the heat rejected are determinable with a fair degree of accuracy.

An average adult, seated and comfortably at rest, must lose heat at a rate approximating 400 B.t.u. per hour. This heat is the energy rejected from the human heat engine as a necessary consequence of securing the work needed to sustain life. If the efficiency of the body be taken as 20%, it is evident that 500 B.t.u. per hour must be the heat input of which 100 B.t.u. per hour is converted to work and the remaining 400 B.t.u. per hour represent degraded energy or waste heat which must be dissipated from the body surface.

The basic problem of comfort conditioning, whether for heating or cooling, by means of convection or radiation, is to establish environmental conditions such that the human body can dissipate its waste heat at a rate corresponding exactly to that at which it is being produced.

There are three mechanisms of heat transfer which can be used to bring about the necessary flow of heat from the body: evaporation, convection, and radiation.

The evaporative loss is limited to a maximum represented by the heat needed to evaporate the quantity of moisture reaching the surface of the body. Except in the range of extreme discomfort, where the emergency action of the sweat glands comes into play, the rate of moisture production is determined by the output of the subcutaneous glands which are responsible for insensible perspiration. Throughout the comfort range the limitation of evaporative loss is due to restricted flow of insensible perspiration rather than inability of the evaporative process to remove moisture from the surface of the body. This fact is clearly demonstrated by the absence of sensible moisture on the skin of a person in a comfortable environment, even though evaporation from the skin surface is constantly occurring at a rate equal to approximately 1.5 oz. per hour. From this it follows that evaporation is not available to the heating engineer as a mechanism of regulation; although more than one-fifth of the heat dissipated from the body is by evaporation, the magnitude of this loss and variation in the rate are subject to physiological rather than engineering control.

Of the heat to be dissipated by methods other than evaporation, the convective process usually accounts for about 40% and radiation for the remaining 60% (for a sedentary adult in room with walls and air at 70F). A convective system for either heating or cooling attempts to vary the fractional convective loss without change in the radiant loss, while a radiant system attempts to control the radiant fraction only.

Basis of Convective Heating. Analysis of the needs of convective control can be easily visualized in terms of the equation,

$$Q_c = \frac{A}{R}(t_b - t_a) \ldots \text{Equation (1)}$$

where Q_c is the rate of convective heat loss in B.t.u. per hour, R is the resistance to heat flow, A the area in square feet of the body surface, t_b is the body surface temperature and t_a the temperature of the air in the enclosure. The value of Q_c is equal to $Q_p - Q_e - Q_r$ where Q_p, Q_e, Q_r are respectively the rates of heat production, loss by evaporation and loss by radiation. With a convection system all of these terms are beyond control and, hence, Q_c has a particular value for each given case and the terms on the right side of equation 1 must be so arranged that the equation will be satisfied. But A and t_b are obviously not subject to manipulation. Then, t_a and R are the sole remaining variables available for control of convective heat loss.

The resistance, R, is the variable which has unwittingly been used by laymen from time immemorial for controlling personal convective loss; thus adding or removing clothing or blankets is a common and effective means of varying resistance and thereby restoring equation (1) to a condition of balance. This method is applicable

Office building of the Raleigh Cycle Company, England.

A French greenhouse radiant heated. Note the reflectors over the pipes from which the radiant heat is derived.

over wide limits, but is subject to the disadvantages attendant upon the inconvenience of being obliged to make frequent changes in one's apparel and the definite discomfort, psychological if not rational, resulting from wearing heavy clothing indoors. The practicability of heat loss regulation by this method is important, however, in that it emphasizes the fact that it is not essential that heat be supplied to the human body. Under all conditions of comfort the body must be subject to a net loss rather than gain of heat and from this it immediately follows that the energy used by a heating system is used not to warm the occupants, but rather to prevent them from suffering an undue loss of heat. In consequence, the writer suggests that any system which establishes comfort through the input of heat is, *prima facie*, irrational and uneconomical. This statement does not imply that such systems are not desirable; at present there is no method, other than the uncomfortable one of controlling resistance, through which heat loss can be regulated except by the input or removal of heat from the environment. But the necessary energy requirements do not vary greatly with the method.

Returning, now, to equation (1), it is evident that control of body heat loss without subjecting the occupants to the need of wearing a particular kind of clothing is limited to the single variable t_a. With all other terms in equation (1) fixed by considerations not subject to engineering control, it follows that the requirements of a heating system which operates through control of the convective fraction is that it establish in the occupied space a value of t_a such that equation (1) will balance.

But when t_a is controlled at a value different from that of the outside temperature it becomes immediately evident that convection must again be considered, this time as it applies to heat transfer between the conditioned enclosure and the outside air. When so considered, both temperatures are fixed by comfort or weather considerations, A is fixed by the structure to be heated and the engineer must therefore use the two indirect variables, R, the resistance to heat flow or insulating value of the structure and Q, the heat added or removed from the structure by the conditioning equipment, as the only terms subject to his control. Thus the basis of convective heating is to hold the air temperature at a value such that the heat loss from the body will be at a rate equal to that of its production; that is accomplished by varying the heat input to the structure. Thus the convective solution of the problem is by supplying sufficient energy to "heat the house" and it is obvious that no direct connection exists between the energy requirements of such a system and the comfort needs of the occupants.

Basis of Radiant Heating. Suppose, however, that the original assumption is re-examined. Is it essential that the equation $Q_c = \frac{A}{R}(t_b - t_a)$ be satisfied with a value of $Q_c = Q_p - Q_e - Q_r$ where Q_r and Q_e remain uncontrollable? Is it necessary that the gross heat loss from the body equal the rate of heat production? The radiant system was designed on the assumption that the above conditions are not essential to comfort and it has been shown by subsequent experience that, over a reasonable range, this assumption is valid. The true requisite of comfort is that the net, rather than gross, rate of heat dissipation be equal to the rate of heat production. The importance of this distinction can not be over emphasized.

Consider, one again, an occupant sitting in a room and losing heat by evaporation at the normal rate of 100 B.t.u. per hour. If the air temperature drops from 72F to 45F the rate of convective loss will rise from 120 B.t.u. per hr. to 290 B.t.u. per hr. and the gross heat loss by these two mechanisms alone would therefore be 410 B.t.u. per hr. Under such conditions the radiant system would establish a net radiant heat flow of 10 B.t.u. per hr. to the body with a resultant net rate of heat loss amounting to 400 B.t.u. per hr. and consequently equal to Q_p. With the net loss thus equalling the actual rate of production there would be no heat entering or leaving storage in the tissues and therefore the thermal equilibrium of the body would be maintained.

Thus the basis of radiant heating is to establish a net rather than a gross body heat loss. Theoretically, the energy requirements of a system which would accomplish this end should be extremely low since it is necessary only to provide heat to the body while the temperature of the air and of the structure are not altered. In practice, however, the need for low temperature large area panels as an aid to effective distribution and to prevent the local overheating which would accompany intense irradiation is responsible for a rather great indirect heating effect on both the air in the room and the materials of the structure. As a result of these secondary effects, the practical panel system acts partially as a convective heating plant and the air temperature in a space heated by this method is usually within 10F of that which would be maintained if 100% convection were used.

METHODS OF PANEL WARMING

Many methods of panel heating are now in use. Practically every working substance which finds application in convection heating plants has also been adopted for use with panels. Thus steam, hot and cold water, exit gases, air and electricity are all finding use as means of carrying energy to the surface of the panel.

Panels of the high temperature electrical or direct fired gas type are, in general, limited to units of the "package" or detachable type and find their widest field of application as auxiliary sources of heat or in establishing zones of local comfort. For high ceilinged structures, otherwise difficult to heat, or in large workrooms where the need for comfort heating is restricted to small and scattered areas, panels of this type serve an excellent purpose.

Steam is rarely used, at least abroad, as a panel heating medium except in units of the cast iron waterway type. Even then, its use is subject to question as the higher surface temperatures of a steam heated panel would seem to hinder the uniformity and distribution of heating effect which are basic characteristics of low temperature radiant warming. Vapor systems operating at temperatures comparable to those now common with hot water (100 — 110F) would hardly be feasible.

Low temperature panels using electric resistances for the heating elements have been introduced within the last few years and recently have gained considerably in favor. One method uses a thin dielectric fabric which is applied to the ceiling of the space to be heated; loading is less than 65 B.t.u. per square foot of heated surface per hour and the surface temperature is rarely allowed to exceed 95F. The fabric is usually installed between layers or plaster board and the thermal capacity of the system is such that continuous operation is recommended; control ordinarily is from an off-on switch actuated from a room thermostat. Since the operating temperatures found in these installations are practically the same as those common to the more widely used sinuous coil system, it follows that the characteristics and performance of the latter system can be used as a guide.

Water is the most commonly used medium for both heating and cooling with panels. Relatively high water temperatures are used in unit panels of the cast iron type while much lower temperatures prevail in the sinuous coil type of embedded panel. The latter method is the one which has been most widely used and it is with this type of panel that the subsequent section of this paper deals.

EMBEDDED COIL PANELS

Construction

The standard sinuous panel is fabricated from ½ in. pipe placed on 6 in. centers with welded 180° turns at each end. Length of the coil and number of returns for a given panel are subject to change dependent on the needs of the particular installation. Panels are prefabricated, all joints welded and the completed unit shop tested for leaks and to insure freedom from "internal" welds. When so constructed, there are two lineal feet of pipe per square foot of panel surface. In some cases ¾ in. pipe is used—in such cases the spacing is 8 in. rather than 6 in.

Present practice in England is to rate a sinuous panel at 100 B.t.u. per square foot when installed in the usual way and operated with water at the recommended temperature. Calculations will show that this rating is somewhat higher than would be expected, but the discrepancy is explained by heating contractors on the ground that the heat losses from a panel heated structure are less than if convective heating were used and that the above rating is suitable for designing panels in terms of the expected heat loss as calculated for a conventionally heated structure. The exact rating of the panel would, of course, be a function of the method of installation as well as of the size, spacing, and heat loss from the pipe coil.

The coil rating and structure heating requirement determine the fraction of the ceiling which must be equipped with coils. Knowing the necessary panel area the next step is to distribute the heating surface in such a way that reasonable uniformity of heating will be realized. This is accomplished by following the customary procedure for shape factor analysis, giving consideration to the distribution of occupants and the sources from which cold drafts may enter the enclosure. For small rooms a single centrally located panel will often suffice; for larger rooms a common solution is to provide the panel surface in the form of a strip running around the room a short distance in from the walls.

The method of installation varies widely with the type of construction. In concrete structures the coils are usually placed flat against the forms and concrete floors (second and above) poured directly over them. Shims are not used so when the forms are stripped dark stripes show the position of the coils above a thin film of laitance. Plaster is then applied to the underside of the concrete and the coils become invisible. In some cases keyed tiles are cast flush with the lower surface of the concrete (at intervals) to provide a binding for the plaster. This method of construction has been criticized because of the likelihood of peeling the film of concrete from the coils when the forms are removed.

A modified method for use in concrete structures is to shim the coils and embed them on the underside to a depth of not less than 3/16 in. In some cases the coils are affixed to the reinforcing and cast in the main body of the slab; this method obviates the need for shims and permits a smooth finish to the concrete which is then faced and remains unplastered.

The relative heat flow up and down through the slab will obviously be a function of the ratio of resistances on either side of the coil; shimming the coil increases the flow of heat to the floor above, but except in the top story this energy provides convective heating of the structure and therefore does not represent a loss.

The radiant heated Daily Mail building, one of eight London newspapers radiant heated with panels. The ninth is electrically radiant heated.

When panels are installed in wood joist construction or in suspended ceilings, it is customary to provide two inches of insulation over the entire ceiling, unheated as well as heated portions. Coils are secured to the joists or to a special supporting framework and the insulation is placed directly on top of them. A special cement plaster with added binding material is then worked up through the coils and packed against the insulation; continued layers of this material are applied until the coils are embedded. Metal lath is then placed under the coils and the same finish plaster is used as with other methods of coil placement. In cases where it is possible to apply the plaster from above, the lath is installed first and the coil plastered in; insulation is then pushed down into the plaster while it is still wet.

The surface of the panel is usually finished with a ⅜ in. floating coat, followed by a ⅛ in. setting coat to which is applied a canvas scrim of ⅛ in. mesh. The scrim is put on dry and trowelled in to a depth such that its pattern is just visible from the plasterer's scaffold. In applying the scrim the Panel Warming Association* recommend that it be extended for at least 12 in. beyond the area in which the panels are located and that the floating and setting coats be applied to all parts of the ceiling.

Heat can be applied to the finished panel ten days after plastering is completed, but for the first week the temperature of water to the panels should not exceed 75F. It can then be increased at a rate not exceeding 2F per day until a temperature of 100F is reached, at which temperature operation should continue until the structure is thoroughly dry.

Decoration of the plaster can be carried out in the usual way, but subject to the following restrictions:

1. Varnish finishes should not be used as there is a danger of discoloration due to temperature.
2. Metallic paints or gloss enamels should be avoided as their emissivities are low and the energy dissipation from the panel is thereby reduced.
3. Efflorescence must be completely removed from the plaster surface before finishing is applied.
4. Where possible, decoration should be postponed until one year after completion of the plastering and in no event should decoration be

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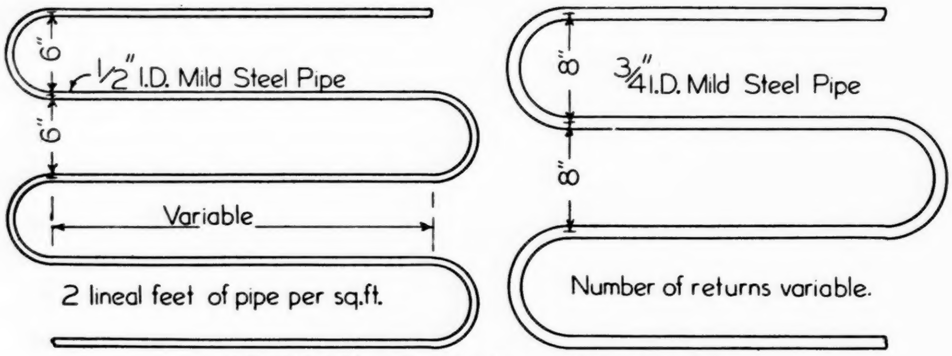
*The method of panel construction, plastering, etc., is patented by Richard Crittall & Sons (Gr. Br.). Wolff & Munzer, Inc., American licensees.



STANDARD PIPING OF SINUOUS COIL

started prior to three weeks of operation of the system.

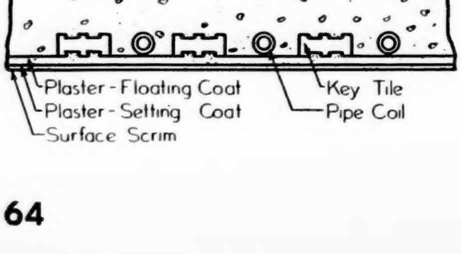
Maintenance

It is obvious that with a system in which the heating elements are permanently embedded in the structure, great care must be exercised in the selection and fabrication of materials. In present practice all coils are made at the factory and tested under 500 lb. per sq. in. water pressure prior to delivery at the site of installation. Field connections to mains and circulating branches are by welded joints except in those cases where the joint will remain accessible. Each section of the work is field tested for six hours under 200 lb. pressure before being embedded.

The most striking demonstration of the practicability of embedded panels from the corrosion and erosions standpoint is in the evidence presented by thirty years of successful operation of such a system in a ten-story English office building; over 50,000 oxy-acetylene welds were cast in concrete, yet no trouble from leaks has been reported and a recent investigation showed that the pipe is in excellent condition. The exterior of embedded coils is obviously not subject to any greater danger of corrosion than are bars of reinforcing; interior corrosion is limited by the small amount of oxygen which enters the system in the make up.

In 1938 the British Hospitals Association sponsored an investigation to determine the relative cost of upkeep and maintenance of panel and radiator systems. The panel system maintenance problems considered were: (1) Liability to cracking of the plaster; (2) Difficulty of locating and repairing leaks if they occur; (3) Damage caused by such leaks. The investigation covered forty-two hospitals. Excerpts from the final report are given below:

Panel cast reenforced concrete floor with key tiles to support ceiling.



"In the majority of buildings visited no cracks at all were visible—in certain instances cracks were discovered by close scrutiny, but these were confined to the skin of paint or distemper. . . .

"In all hospitals visited no repairs of any kind had been required to any embedded panels or steel piping, nor had any leaks occurred. It is considered that objections (2) and (3) have no foundation in actual experience—nor is there reason to anticipate failure through external or internal corrosion or obstruction from deposits in the pipes."

Operation

The principal operating difference between the sinuous coil panel system and other heating systems in which forced circulation hot water is used as the heating medium is that the surface of the panels is maintained at a temperature decidedly less than is used with radiators or other standard heating elements. Surface temperatures exceeding 120F are rarely permitted in panel work and it is considered better practice to design for a normal load surface temperature of less than 100F. This leads to a circulating water temperature much lower than that which would be used in a radiator system. Values of temperature as found in typical office building installations are: water to panels, 95F to 110F; temperature drop through the panels, 6F to 10F; panel surface, 83F to 98F.

Gravity circulation has been used in panel systems, but forced circulation is more common. The pressure drop through each panel is usually equivalent to approximately six ft. of water and a customary range of total head for the entire system is 15 — 20 ft. of water. Coils are ordinarily piped in parallel, although it is not uncommon to find up to four coils in the same panel connected in series.

Since sinuous coils are embedded in the mass of the structure it would naturally be expected that the heat capacity of a panel system would exceed that of convective systems. This consideration leads to a question as to the ability of such a system to respond to sudden changes in the weather. The only data on this problem based on a study of actual installations is that collected by the British Hospitals Association. From observations of the panel systems in 42 hospitals the Association reports:

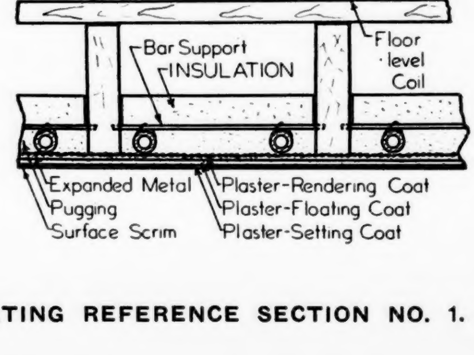
"In certain circumstances the time lag of the warming effect from embedded panels is greater than that of a radiator system. Inconvenience from time lag does not arise, however, when proper circulating temperatures in relation to the requirements are maintained and is only likely to occur seriously where the system is being operated at a temperature higher than that recommended."

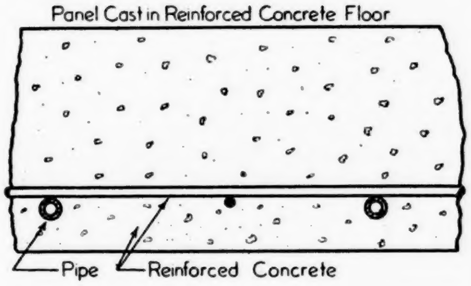
That the above statement is not unquestioned is suggested by the following quotation from a personal communication submitted by an engineer in charge of a large panel heated office building:

"The main defect of a panel system is the difficulty of lowering the temperature of the space heated at a sufficient rate to synchronize with the weather conditions. In this connection you will appreciate the mass of concrete forming the floors gets warmed up and even though the water temperature is reduced this heated mass tends to maintain the temperature at the higher level; with adequate ventilation . . . it is possible to maintain reasonably good comfort conditions."

The whole matter of heat capacity and time lag in panel systems is in need of attention both from the analytical and experimental standpoints. Lag will obviously be greatest at times when the outside temperature is rapidly rising or falling; in climates where this condition is common a careful analysis of the lag effect based on the storage characteristics of the particular structure under consideration should precede the choice of a heating plant.

The effect of high thermal capacity would materially increase first cost due to the necessity of providing an unusually large start up capacity of the heating plant, if it were not for the fact that panel systems are almost always designed for continuous rather than intermittent operation. On the face of it, this fact would suggest increased operating costs, but experience seems to show that continuous operation actually reduces the energy requirements in most cases. Theoretical analysis in terms of the transient flow equations would permit close determination of





the actual requirements for a particular structure and would serve to check on the above general observation.

Control

Control of a hot water panel system is accomplished by adjusting the temperature of water to coils in accordance with outside conditions and throttling the flow through individual sets of coils to meet the needs of particular rooms or sections of the building. The desired water temperature is a function of outside temperature and wind velocity; in large installations it is manually set twice daily, the proper setting being determined by the operator from a posted schedule for the particular building. Throttling control for individual rooms is either manual or is automatically carried out through use of an air thermostat. The control point for air temperature is frequently reduced 5F during the night.

That a radiant system should use air temperature as an index of comfort seems rather strange, but the control point is at a value less than that which would represent comfort conditions with a convective system. It should also be noted that the air temperature necessary for comfort will be a characteristic of each particular system and the proper thermostat setting for any given installation will therefore have to be determined through operating experience.

The control procedure outlined above is that which is most commonly used in European installations. Experience indicates that it does give reasonably satisfactory results, but theoretically the method is unsound in that the inside air temperature which would correspond to optimum comfort must vary with the outside temperature. One method of providing more rational control would be to use a combination radiation-convection sensitive device to actuate the energy input. One such device is now available.

Characteristics

From considerations of comfort and operating economy, the two most important characteristics of any low temperature radiant system are, respectively, lowered air temperature and reduced floor to ceiling temperature gradient.

Experience shows that in an average panel heated building, it is possible to obtain comfort with an inside air temperature approximately 5F lower than that which would be needed if a convective heating system were used. The exact reduction in temperature which is possible is obviously a characteristic of the particular installation and the above figure is given only as a general indication of what can be expected. Mr. Barker cites the experience of an American firm which established a branch in London and insisted that the panel heating system designed for their large drafting rooms be capable of maintaining a 75F air temperature. Under protest, they reduced the requirement to 70F, but found when the system was installed that any temperature in excess of 65F was unendurable.

Except in buildings having unusually high ventilation rates, the lower limit of air temperature is usually established by the secondary convective heating effect of the panels rather than by comfort requirements of the occupants. If it were possible to increase the fraction of panel heat dissipation represented by radiation, there is much evidence to indicate that air temperatures well below existing practice would be comfortably possible. As development of the panel system continues and suitable surfacing materials become available, there is every reason to believe that the inside air temperature will be reduced.

The possibility of attaining comfort in a room with lowered air temperature has been the subject of a number of investigations. With one exception all evidence indicates that there is neither physiological nor psychological objection to establishing a body heat balance at lowered air temperatures; use of radiant energy to make up the excess body heat loss is apparently as acceptable as the use of convection to reduce the total loss. The one exception is a report by Dr. Vernon based on work at the British Industrial Fatigue Laboratory which showed that (based on a limited series of experiments) it was possible to achieve comfort only when the air temperature was within 1F of that value which would be considered comfortable if the room were heated by convection.

In opposition to the conclusion of Dr. Vernon is the statement of Sir Leonard Hill:

"A much lower room temperature, e.g. 56F, is felt to be comfortable when the ceiling affords radiant warmth than when heated air is made the means of obtaining warmth."

An investigation of conditions at the sinuous coil heated British Embassy at Washington, D. C., led to the statement:

"The consensus of opinion was that substantially comfortable conditions were experienced in a room maintained at, or slightly below 60F,—that complete comfort was experienced in the rooms maintained at about 62F and that a sensation of serious overheating was experienced in the rooms at 66F."

L. W. Schad, discussing the work carried out at the Westinghouse Laboratory, comes to the following conclusions:

"Preliminary work showed quite definitely that not only can comfort conditions be obtained with 80F walls and 60F air, but also that such an environment is very invigorating. The cool air is quite acceptable when at the same time one feels comfortable."

Dr. C. A. Mills summarizes the results of limited experimental work at Cincinnati with the statement:

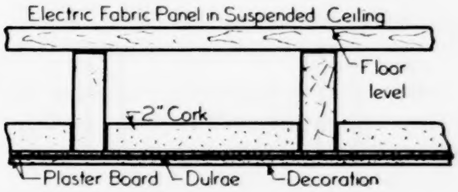
"The finding that body comfort and optimal animal growth can be secured in a (panel conditioned room), below the perspiration point, means that removal of practically all body heat through radiant channels is physiologically possible and apparently quite desirable."

Transmission losses in a panel heated structure are not necessarily reduced as a result of the lower air temperature because the outer walls are heated by direct radiation and may reach an equilibrium temperature higher than that of the corresponding convective system. Note that this value is not *necessarily* higher than that of the convective system since, with panels, the warmer ceiling and floor may raise the mean surface temperature even though the walls are less warm than would be the case if convection were used. Calculation is necessary before it is possible to state for any given structure whether the use of panels will increase, decrease, or leave unchanged the transmission losses.

The need for careful engineering is greater with panel design than it is for convection. Calculations necessary for precise determination of the relationships between the heated and unheated room surfaces, between the inside and outside air temperatures, are less adaptable to abbreviated methods or rule of thumb equations than are the transmission calculations for a convective heated structure.

The floor to ceiling temperature gradient is much less with a panel than with any other type of heating system. The Industrial Fatigue Research Board investigated gradients in a panel warmed room in which the panel was installed in the ceiling near a wall with windows. The difference in temperature between the floor and a point just below the ceiling was less than 2F.

A similar investigation by the British Hospitals Association showed that in rooms having ceiling heights of 25 ft. or more, ". . . the temperature registered at the floor level is only a few



degrees below that found two ft. from the ceiling."

Substantiation of the English observations of reduced temperature gradient comes from recent American tests of floor warmed heating systems. In tests at a small residence it was found that the temperature difference between floor and ceiling was 3F when the floor temperature was 69F, but dropped to .3F when the floor surface temperature was raised to 70.6F. Since in panel heated rooms the floor temperature usually exceeds the air temperature it would seem from the above results that this may be the factor, or one of the factors, responsible for the low temperature gradient that is characteristic of panel installations.

Installation

Sinuous coil panel construction has been so completely standardized that little is to be gained from prolonged consideration of existing installations. In most cases the arrangement and construction of the panels and the operation of the system follow a general method corresponding to that already discussed and in agreement with the recommendations of the Invisible Panel Warming Association. Illustrations of a few buildings so equipped should serve to indicate the type of structure in which the radiant system is being used.

The Daily Mail building is but one of eight London newspaper offices (including the Times) heated with panels; a ninth, the Provincial Building, is electrically heated, and represents a load of 46 kw.

The Strand Palace Hotel, Lyons Laboratories, and the Raleigh Cycle Company are typical business structures using sinuous panels.

While there are more European installations in residences than in any other class of building, schools, hospitals, and public buildings hold second, third and fourth places in popularity and among them account for over 500 multi-story European installations.

One of the best known London panel warmed buildings is the ten-story "Underground" Administration Building at 55 Broadway. Sinuous ceiling coils are used throughout this structure with auxiliary floor coils near the entrances at the street floor. The calculated maximum heat load on the system is in excess of 3,000,000 B.t.u. per hour. An interesting feature of this installation is the use of electric thermal storage; all energy is received during the night hours when the off-peak rate schedule is in effect and conversion takes place in immersion heated boilers which raise the temperature of water in two storage cylinders to 275F. The temperature of circulating water going to the coils is maintained at its proper value (usually around 100F but varying with the weather) by automatically replacing the necessary fraction of circulating water with high temperature fluid from storage.

The radiant heated Wandsworth Municipal offices in London, England.

Electric thermal storage systems similar to the one described above are increasing in popularity throughout England as a result of the favorable off-peak rates; large users can purchase off-peak energy for space heating at $0.004 per kw. hr. Already there are 200,000 kw. of such systems in use including the Hackney Town Hall, Birmingham Airport and a Synagogue in West London.

Thames House, Millbank, London, is a large completely panel warmed building using coke-fired boilers for direct water heating. Boiler capacity was originally calculated on the basis of a convective heating system and indicated a need for four boilers (including 15% allowance for "startup"). With panel heating installed in the building it has been found possible to operate in coldest weather with two boilers and, in consequence, one of the original units has been taken off the system. Operating experience in this building has shown that a gradual reduction in heat requirements occurs over the first two or three years after the system is installed. It is believed that this is due to slow drying of the structure.

Thames House is also noteworthy in that it is one of the three largest buildings in which the sinuous panels are used for cooling as well as heating. For summer operation water is cooled to 55F by means of an ammonia refrigerating unit and is then circulated through coils. A reduction in air temperature of from 6 to 7F is obtained, but the resultant effect in terms of comfort is much greater than this small air temperature reduction would indicate since body heat dissipation is increased by radiation to the cold panels. An obvious necessity with such a system is that the temperature of cold water to the panels be carefully controlled to insure a surface temperature higher than the dew point in the room; no difficulty has been experienced at Thames House from this cause. The second and third largest year-round panel conditioned buildings are in Paris — the Banque l'Union Parisienne and the Paternelle Insurance Company.

Special Applications

Aside from the direct comfort and operating advantages claimed for low temperature radiant heating, an additional factor which contributes to the popularity of this system is its adaptability to use with certain unusual arrangements of equipment. Thermal storage has already been mentioned. While there is nothing new in the use of off-peak energy for heating, this arrangement has been too costly to justify its use with heating systems of the conventional type. With panel heating, however, the temperature of the water going to the coils is so much lower than with radiators, that the cost of thermal storage is greatly reduced. Consider, for example, a building with hydrostatic head such that the storage temperature could be raised to 270F; the heat which could then be stored in each pound of water would be 270 — 100 = 170 B.t.u. per pound (assuming a design temperature of 100F for the circulating fluid) while a radiator system circulating at 180F would permit storage of only 270 — 180 = 90 B.t.u. per pound, or, to put it differently, the storage plant needed for a panel heated structure of this kind would be only 53% as large as that which would be necessary if hot water radiators were used. The advantage of panels in this respect increases as the available hydrostatic head decreases. For a residence or other one-story structure where the permissible storage temperature is 212F the storage cylinders needed with panels would be less than 1/3 the size of those needed with 180F water. Thus it is seen that the panel system affords an opportunity to take advantage of the operating savings made possible through the use of storage systems without the usually heavy burden of increased capital charges.

Of even greater significance is the possibility of using thermal storage to reduce the size of refrigerating plant necessary for panel cooling. If 100 tons of capacity were required during a twelve-hour period it is clear that a compressor of one-half this capacity could, in conjunction with a storage plant, satisfactorily handle the load.

The reduced fluid temperatures characteristic of panel heating make this system particularly attractive for use with a heat pump and suggest the probability of efficiencies well beyond those which are obtained with the usual system.

A limited number of reprints of this 32-page section on radiant heating (H & V Reference Section No. 1) available at 25 cents each, from HEATING AND VENTILATING, 148 Lafayette St., New York, N. Y.

How to Solve Anthracite Stoker Servicing Problems

By RALPH A. KRAUSS†

In last month's article, the writer classified service complaints as *emergency calls, poor efficiency* complaints, *unsatisfactory control* troubles, and *nuisance* problems. These will now be discussed one by one.

Emergency Calls. The majority of emergency calls are occasioned by failure of the coal to feed, usually caused by the worm becoming uncovered or by an obstruction of foreign material. Broadly speaking, this may be understood to include jams in the ash removal system due to failure to empty the cans, although safety releases are ordinarily employed to spill the excess on the floor and permit continued operation.

An excessive load caused by foreign material which moves with difficulty through the coal tube may trip the overload cut-out, stopping the motor. However, if the self-resetting type of cut-out is employed, the stoker will continue to operate at intervals, although the relay remains in the closed position. Ordinarily, however, foreign material will cause the shear-pin or safety link to break, so that the motor continues to run without feeding coal. The same conditions may be caused by hard clinker in the ash-removal system or by accidental interference of moving parts.

In any case, the proper procedure is to find and remove the interference and replace the shear pin if necessary. The problem of hard clinker will be discussed in a subsequent section.

Failure of controls or faulty electric wiring, while less frequently encountered, will also result in emergency calls. In this case the motor will not run because the line will be dead ahead of the overload cut-out. Tracing the circuit with a test-lamp will isolate the fault. Ordinarily, a faulty control instrument should be replaced and returned to the manufacturer.

Insufficient Heat. Although insufficient heat may be the cause of an emergency call, it is a condition which may best be treated separately.

A complaint of insufficient heat may often be due to causes over which the stoker has no control. Any number of faults of the heating system may prevent the heat generated by the stoker from reaching the rooms, in which case the limit controls take charge, resulting in intermittent operation even while the room thermostat may be continually unsatisfied. Other heating system faults which may lead to this complaint will be discussed later under *poor distribution.*

Constant manipulation of the thermostat may also be a cause. It is a rather prevalent practice, indulged in particularly by women, to reduce the thermostat setting at times to save fuel, and restore the control to normal when more heat is wanted. Naturally, the temperature does not immediately rise to the desired point, and in large houses of massive masonry construction the recovery may take one or two hours. Meanwhile, the stoker may be operating intermittently due to the action of the limit controls.

Sometimes the boiler may be so dirty that the efficiency is seriously impaired. Cleaning the flues once, or even twice, each year is recommended for domestic heating boilers, while for commercial boilers more frequent cleanings are usually necessary.

If the stoker is at fault it may be due to insufficient coal feed caused by too low a setting of the worm speed or by accidental starving of the feed system. On the other hand, it may be due to low efficiency from a number of possible causes.

Low Efficiency. This term should be considered mainly as a convenient one for grouping the major combustion complaints. Actually, it is prohibitively expensive to obtain a direct measurement of the efficiency of a small heating plant, although close approximations by the *heat-balance* method may sometimes be employed to advantage. This will be discussed in a subsequent paragraph.

A great many complaints of poor efficiency are based upon the user's inexpert appraisal of the ash. The major producers of anthracite today are marketing a product containing so little ash that a small quantity of unburned combustible is rendered conspicuous out of all proportion to its importance. Whether the customer wishes to operate an efficient heating plant or an ash factory is, after all, a free choice, providing he understands the difference. While it is often possible to secure an extremely well-burned ash and high efficiency, the fact remains that in most installations the maximum economy is obtained by the use of an air setting which yields a noticeable percentage of carbon in the ashpit. The wisest course is to explain that there are invisible as well as visible losses, and the true index of efficiency is in the coal bill.

Heat Balance. Because most of the losses may be measured or computed, and the others estimated with reasonable accuracy, a fair approximation of the efficiency of the plant may be obtained without resorting to power-plant methods. The loss due to the heat of the escaping gases in the greatest, and subject to the greatest variation and control and may therefore be investigated with the most profit, using only a flue-gas analyzer and stack thermometer.

The accompanying chart, Fig. 1, permits rapid computation of the heat loss in the dry flue gases. Its simplicity arises from the fact that the combustible matter of anthracite is over 92% carbon, so that variations in its composition have little effect upon the mathematical and chemical formulae involved.

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†Combustion Engineer, Anthracite Industries Laboratory, Primos, Pa.

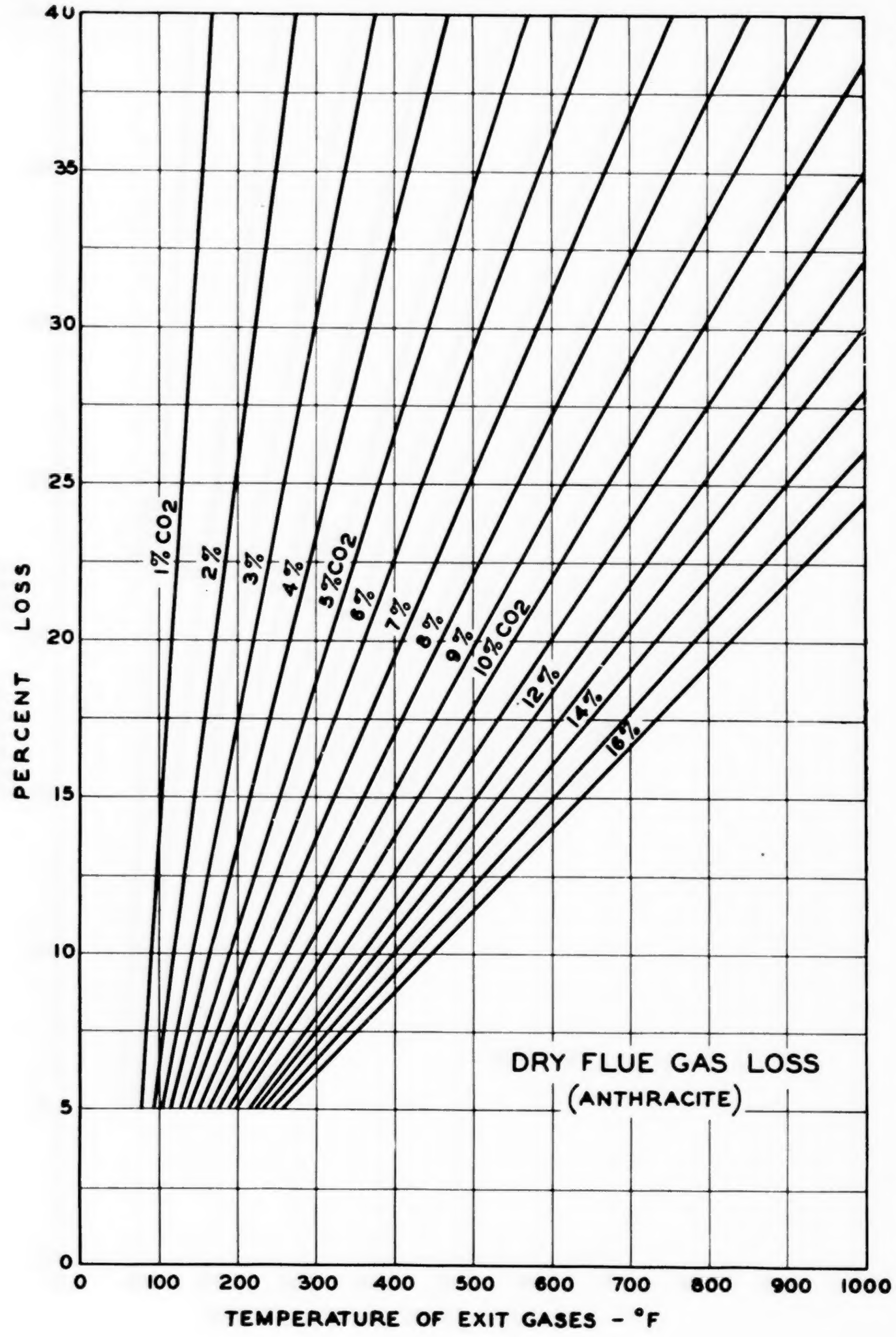


Fig. 1. To determine the dry flue gas loss move vertically from the temperature on the horizontal scale to the line representing per cent CO_2 as determined by analysis, thence, horizontally to the left scale and read the stack loss.

Since the exit gases from an underfeed anthracite stoker are normally free from even traces of combustible constituents, the CO_2 determination is sufficient, reducing the gas analysis and the heat balance to the simplest possible terms.

Fig. 2 gives the relation between the percentage of combustible in the ash and the actual loss in percentage of the total coal fed. Accurate determinations obviously require an analysis of the contents of the ash pit, a procedure seldom justified in domestic practice because of the expense. Visual observations are notoriously misleading and an expert may easily estimate with an error of 10% in either direction. The ash content of the coal, also necessary for this computation, may ordinarily be obtained from the shipper, but for most purposes an average figure will give reasonably accurate results. The Bureau of Mines has recently conducted a survey of the major sources of Pennsylvania Anthracite, reporting the average ash content of No. 1 buckwheat as 11.7% and of rice 12.4%.

Under average conditions the ash will not contain over 25% combustible, and the maximum for acceptable operation (according to the National Bureau of Standards) is 40%. It therefore becomes obvious that the ashpit loss will average about 4.5%, with a maximum, under heavily-loaded conditions, of about 9%.

Heat in the water-vapor in the flue gases varies with the percentage of hydrogen in the coal, the temperature of the flue gas, and several minor factors. Ordinarily, however, it remains between 2 and 3%, with an average of 2.5%. This figure may be safely used except where extreme accuracy is required, as in laboratory or power-plant practice.

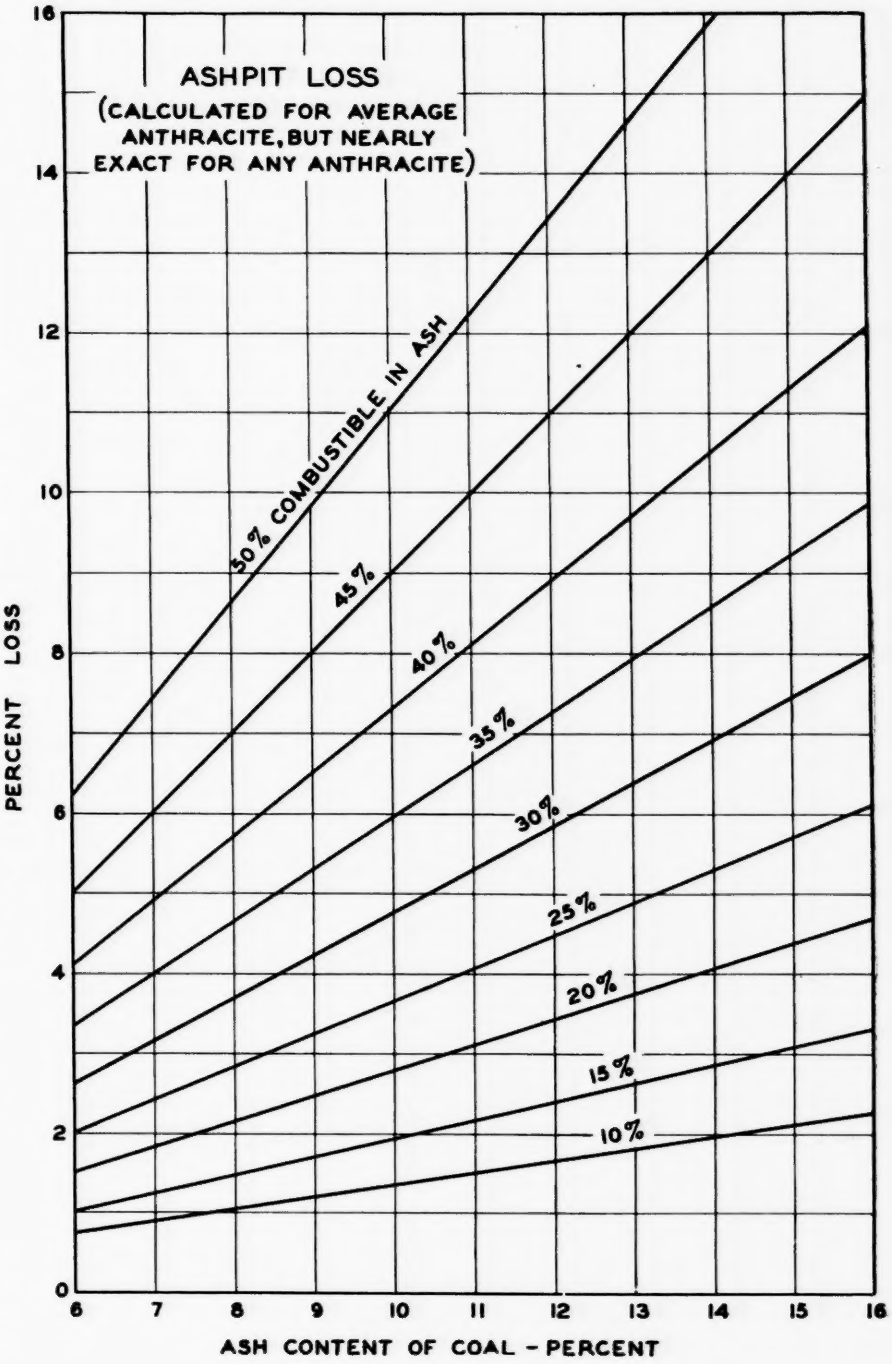
The usual heat balance also includes another term, *radiation and unaccounted-for.* In the case of an underfeed anthracite stoker, the *unaccounted-for* item is negligible because of the chemical simplicity of the fuel. When the efficiency is measured directly, however, it becomes the retreat of all errors both in measurement and calculation. *Radiation* is that portion of the heat which the boiler emits from its surfaces. Whether or not the boiler is insulated, this is useful heat and not a loss, so long as the boiler room is part of the house, and is not overheated.

The following is a typical heat balance:—

Loss to heat in dry flue gases (10% CO_2, 550F stack)	20.0%
Loss to ashpit (27% carbon in ash, 12% ash in coal)	5.0
Loss to moisture in flue gases (Estimated average)	2.5
Efficiency (by difference)	72.5

The efficiency determined in this manner is conserv-

Fig. 2. To determine the ashpit loss with anthracite, move vertically from the per cent ash in the coal, as determined by a chemist's analysis, to the curve representing the per cent combustion in the ash, also determined from analysis, thence, left to the scale showing the ashpit loss.



ative, because additional heat is given up from the flue pipe and from the chimney walls to the house, this heat having been charged off under dry flue gas loss. However, we are not so much interested in the exact efficiency as we are in finding whether the CO_2 is susceptible of improvement. Therefore we make the measurement as close to the boiler as possible to minimize the diluting effect of any inward leakage of cold air at the smokepipe joints.

Improving Efficiency. Readjustment of the air supply may sometimes become advisable because of such conditions as a change in the size or source of fuel, tampering with the setting, etc. If the fire is in good condition, free of holes and clinkers, proper adjustment is not difficult. If a flue-gas analyzer is not available, the air may be adjusted, slowly and at intervals, until the ring of dead ash at the edge of the retort is approximately one inch wide. Analyzing the flue gas, besides being a more dependable method, reveals the presence of excess air from other sources, such as leaks in the boiler setting.

Gas samples should be taken with the stack damper temporarily *cracked down* so that slightly positive pressures exist throughout the boiler. Ordinarily, at least 10% CO_2 should be obtained with the stoker in normal operation—that is, after running steadily for a half-hour. The higher the gas temperature, the more important it is to obtain a high CO_2, even at the expense of increased ashpit loss. However, if the boiler is very efficient or so effectively baffled that the stack temperature is below 500F, a lower CO_2 is permissible, although 8% should be regarded as the lower limit.

While efforts to increase the CO_2 should be halted short of the point where live coals are discharged into the ashpit, it is neither practical nor necessary to find the exact peak of the efficiency curve, which is quite broad and flat, due to the fact that as the ashpit loss increases, the flue gas loss decreases. Naturally, if there is sufficient draft, additional baffling is advantageous.

A second gas analysis, determined with the damper set normally (so as to obtain a slight negative draft in the firebox) will then show whether significant quantities of air are leaking in through cracks in the boiler setting, around fire doors, ashpit openings, etc. Such leaks may be found with a lighted candle or cigarette, and sealed with boiler putty, furnace cement, or asbestos plaster.

Dirty Fire. A ragged fire, full of clinkers, or a sluggish fire, constitutes a more complex situation. In either case the ashpit loss is probably excessive and the CO_2 low. Raggedness and clinkers may be due to unsuitable fuel, excessive air pressure, clogged tuyeres, and possibly other causes, while sluggishness can only be due to insufficient air. In general, it may be stated that unsatisfactory fuel bed conditions arise from one or more of the following:—

- Mechanical obstructions in the air supply system, preventing the proper quantity of air from being delivered and uniformly distributed in the retort.
- Coal having a low fusion temperature, so that it clinkers readily.
- Conditions in the coal or conveying system which causes excessive pulverization.
- Excessive air pressure.

To investigate all these possibilities requires removing the fire and inspecting the retort and tuyeres. However, there are good reasons for postponing this step as long as possible. It not only necessitates shutting down the plant for perhaps an hour, adding to the discomfort of insufficient heat, but in addition, the newly built fire will be clean and regular, and for a period of an hour or more will not exhibit the previous symptoms.

If the coal feed has been set higher than the burning capacity of the retort, efforts to secure proper combustion by increasing the air feed will only result in clinkers, blow-holes, and general raggedness. Sometimes under these conditions a pie-shaped sector may completely refuse to burn. Obviously the only remedy is to reduce the coal feed, and it is often found that even a slight decrease will permit the proper adjustments of air pressure to be made, with an increased efficiency which more than makes up for the decreased feed. However, if the stoker is badly overloaded, due to miscalculations or the addition of unanticipated load, the retort size must be increased, and this can sometimes be done by adding a dead-ring.

Sluggish fires, while usually due to obstructions in the air pipe or fouled tuyeres, may be caused by an *airbound* cellar condition. This is found when all the cellar windows and doors fit snugly, and the air for combustion (some 300 cubic feet for every pound of coal) must be pulled through the cracks at the expense of considerable energy.

This condition is usually difficult to diagnose because the mere act of opening a door to enter the cellar will temporarily relieve the partial vacuum, and if the door is left open while the investigation is being made, the symptoms will not appear until after the serviceman has left. In order to guard against an airbound cellar, the air supply should be ensured either by blocking a cellar window partly open, removing a pane of glass, or sawing an inch off the bottom of the door communicating with the kitchen. The latter has much to recommend it as it helps to ventilate the kitchen, avoids cold drafts in the basement, and supplies the stoker with warm rather than cold air, resulting in better combustion and a slight gain in efficiency.

Checking the Coal Feed. It is often desired to determine, over a period of several days, the actual number of running hours. This may be accomplished by a Telechron clock wired in parallel with the motor, or by a counter of the rotary or ratchet type, clamped in such a position as to be actuated by some moving part. Observation over a period of a few minutes will establish the normal speed as recorded by the counter, and the corresponding rate of a coal feed can be secured from the manufacturer, or determined by test at the dealer's premises.

Comparisons of the readings of either instrument with the average outdoor temperature during the test period will permit a close forecast of the total annual coal consumption, and will also show whether the coal feed is sufficient to carry the load in the most severe weather.

Heat Distribution. A stoker is seldom responsible for inequalities of heat in the various parts of the building; nevertheless, the stoker dealer is almost invariably held responsible. Whether the unequal temperatures are due to wind and sun conditions, inoperative valves, traps, or vents, unbalanced radiation, or improper piping, the correction should be at the customer's expense, and often more salesmanship than mechanical ability is required.

However, if the stoker (or even the baffle) is set out of center in a round warm-air furnace, more heat will be transmitted to the leaders on one side and the system must be rebalanced. Also, in a one-pipe steam system converted from hand firing to stoker (or oil burner, for that matter) some inequalities of distribution may be accentuated, particularly where there are long mains. In this case, a quick-vent air valve at the end of the main will overcome a great deal of the difficulty.

Nuisances. Today's anthracite stoker is so well constructed that it makes less noise in the living room than does the average electric refrigerator. Noise or vibration sufficient to constitute a nuisance is therefore usually caused by accident or careless installation. Even flexible conduit, when bent to a sharp curve, will transmit vibration, possibly to the floor joists above. Motors set in rubber cushions can set up an annoying disturbance if in contact with the casing of the stoker. Hard clinker may cause crunching noises in the ash removal system. These are usually easy to detect and correct, while squeaks and squeals have a way of appearing to be where they are not. Methodical lubrication in most cases takes care of these, however.

Dust can be blown about only when there is a hole through which it can issue and a pressure to cause it to be ejected. Neither should exist, and either or both may be corrected so far as the boiler is concerned. Holes or cracks in the coal-conveying system will leak dust, whether or not there is air pressure in the tube. Obviously the crack should be sealed.

Odors as well as dust may result from the bin or hopper being allowed to run low so that the end of the worm is uncovered. This is a pungent odor, quite different from the sharp, acrid, choking odor of flue gas, which indicates a positive pressure within the boiler and the need for opening the stack damper slightly wider.

Summary. The service department can be either an asset or a liability to the stoker dealer. This article suggests ways in which it can be made to pay real, if somewhat intangible, dividends in the shape of pleased customers, and how costs may be reduced by searching diligently for trouble before it makes itself evident.

Air Dilution in Industrial Ventilation

By W. C. L. HEMEON[†]

This is the concluding article of a series on dilution in connection with industrial ventilating. Parts 1, 2, and 3 have appeared in previous issues.

Effect of Room Dimensions

SPACIOUS arrangement of processes, high-posted rooms, etc., are frequently cited as safeguards against room air becoming over-contaminated. An examination of the facts relating to room volume as it affects air contamination is instructive.

Natural Ventilation—Rooms of large dimensions are likely to have more window and door area than smaller ones with, consequently, improved facilities for natural ventilation. This may be of advantage where processes in the room are susceptible to control by good natural ventilation either by reason of the small quantity of contaminant evaporated into the air, or of a liberal toxic limit for the contaminant in question. In many cases, obviously, the opposite is the case when these advantages would be slight, or negligible.

Build-up Rate—The rate at which concentrations of contaminant increase in a workroom before an equilibrium value is reached is influenced by the volume of the workroom as well as by certain other factors. Depending on circumstances such rate may or may not be of importance.

An analysis of the relations prevailing during the period of changing concentrations (for example during the early morning period immediately following the start of operations) will make this point clear.

Continuous Contamination—Consider a room ventilated by propeller fans in which a toxic substance is continuously evaporating at a known rate. The differential increase in the total quantity of contaminant in the room air, d (Vx), in time interval, dt, is equal to the rate of generation minus rate of escape in exhaust.

$V dx = G dt - Qx dt$, where

x = Concentration (fraction) of contaminant in air at time, t.

Q = rate of air flow through room (c.f.m.)

V = room volume (cu. ft.)

G = rate of generation of contaminant (c.f.m.)

when rearranged and integrated, we get—

$$\int_0^x \frac{dx}{G - Qx} = \frac{1}{V} \int_0^t dt$$

$$\ln \frac{G - Qx}{G} = - \frac{Q}{V} t \qquad \ldots\ldots\ldots\ldots (3)$$

This relationship enables us to calculate the build-up rate, starting with zero concentrations in a room, provided minimum data are available.

Taking the data of Example A[1], let us compute the time before concentrations of methanol will have reached 100 p.p.m., which is about two-thirds of the equilibrium value. (Note that, theoretically, the equilibrium concentration is attained only in infinite time.)

Data

x = 100 p.p.m.
Q = 8000 c.f.m.
V = 150,000 cu. ft. (150 × 50 × 20 ft.)
G = 1.35 c.f.m. (50% of 2.70 c.f.m.)

$$t = -\frac{150{,}000}{8{,}000}(2.303)\log \frac{1.35 - 8000\frac{100}{1{,}000{,}000}}{1.35}$$

$$t = -43.1 \log 0.407$$

$$t = 17 \text{ minutes}$$

In a similar manner, it is computed that concentrations will reach 140 p.p.m. in 33 minutes.

If with other conditions unchanged the spacing of operations were to result in doubling the floor area, then twice the previous time periods would be required to reach stated concentrations—34 minutes for 100 p.p.m., 66 minutes for 140 p.p.m. From these calculations, it is to be concluded that variations in room volume within practical limits are frequently without significant effect on ventilation requirements.

Discontinuous Contamination—The advantages of large room volume on the control of contaminants may be real in circumstances where there are important interruptions in those processes that foul the room air. The following hypothetical example is illustrative of such a situation.

Suppose identical processes in two different rooms, Numbers 1 and 2, the latter having twice the cubic volume of Number 1, rate of generation of contaminant, rate of ventilation, and schedule of operations with respect, particularly, to interruption periods during which no fouling of air occurs, are the same in both cases.

Schedule of operations is as follows: starting with pure air in the room, a preliminary period of normal operations ensues for 34 minutes during which contaminant is generated at a constant rate. Then follows a period of 26 minutes during which no fouling process occurs, then 21 minutes of normal operation. Successive similar periods of 26 minutes and 21 minutes occur. The exhaust fan is operated continuously.

Data

Q = 4,000 c.f.m.—rate of ventilation
G = 1.35 c.f.m.—rate of generation of vapor
V_1 = 150,000 cu. ft. (Room No. 1)
V_2 = 300,000 cu. ft. (Room No. 2)

[†] Engineer, Division of Occupational Hygiene, Massachusetts Dept. of Labor and Industries.

[1] See page 41, HEATING AND VENTILATING, November, 1940.

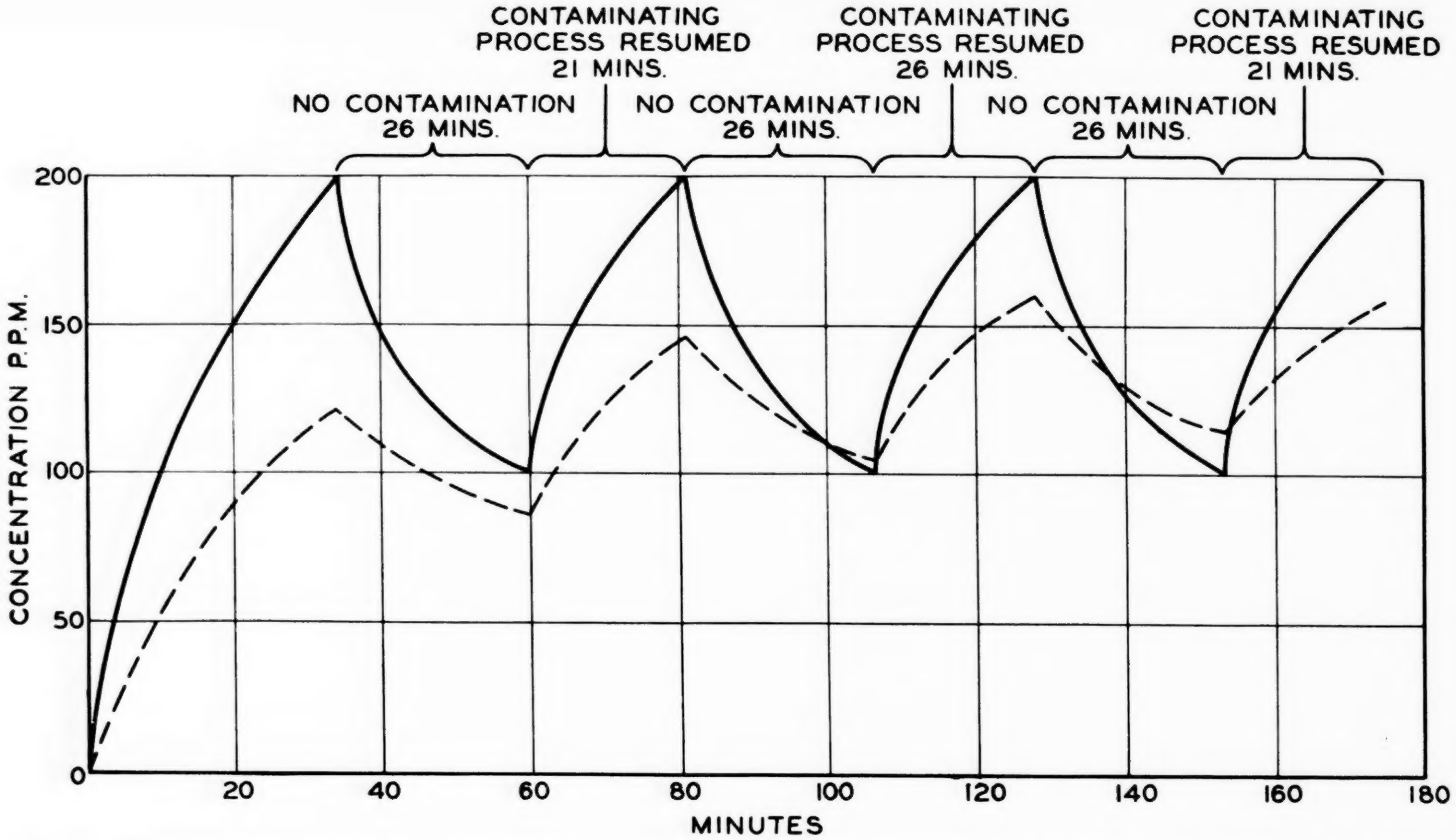


***Fig. 6.** Effect of room volume on concentrations of contaminant where contaminating process is not continuous. All conditions are identical in two cases, where solid line represents a room of twice the cubic volume of the other. From the practical viewpoint of the ventilation engineer, no account would be taken of such differences in most cases.*

What will concentrations be in each room throughout several cycles, assuming perfect and instantaneous mixing of air and vapor?

Solution: Equation (3) applies to the preliminary 34-minute period when starting concentrations are zero. For periods during which no vapor is generated, the following relation obtains:

$$V\,dx = Q\,x\,dt$$

$$\int_{x_0}^{x} \frac{dx}{x} = \frac{Q}{V}\int_{t_1}^{t_2} dt$$

$$2.303 \log \frac{x}{x_0} = \frac{Q}{V}(\triangle t) \quad \ldots\ldots\ldots (4)$$

x_0 and x being concentrations at beginning and end of the period, respectively.

For alternate periods which start with contaminated air, another form of (3) applies (derived similarly):

$$2.303 \log \frac{G - Qx}{G - Qx_0} = -\frac{Q}{V}(\triangle t) \quad \ldots\ldots\ldots (5)$$

Concentrations have been calculated by means of these equations for the several time periods, the results of which are shown in Fig. 6. The effect of larger room volume under these circumstances is clear. It will be noted that not only is there a lag in rate of increase of concentrations during active periods, but there is a corresponding lag in the rate at which concentrations decrease during the subsequent lull period.

It is apparent from the foregoing that where processes generate air-contaminating substances continuously, the only effect of larger room space is in a time lag at the beginning of the day (and following such shut-downs as at noon time) in the rate at which concentrations of contaminant build up. The effect of larger room volume is more likely to be noted where there are important interruptions in the air-fouling process.

Air Changes Rate—Rates of ventilation (as applied to systems of dilution ventilation) are often expressed in terms of numbers of air changes per minute or per hour. This terminology is in most cases meaningless and it is urged that it be avoided. It will be noted that the term Q/V, a factor expressing air changes per unit time, occurs in equations (3), (4) and (5) which deal with rate of change of concentrations, but that in (3) and (5), Q appears in other parts of the equation divorced from the factor V. Only in (4) does it appear that the rate of ventilation in terms of air changes per unit time is simply related to the rate of change in concentration. This unit should, therefore, be dispensed with in industrial ventilation usage in favor of a straightforward statement of air flow rate in terms of volume per unit time.

Effect of Vapor Density—Considerable misapprehension appears to exist as to the importance of vapor density in the design of ventilation systems. Vapor density is the weight of the vapor or gas (or mixtures with air) relative to the weight of an equal volume of air. When evaporation takes place at room temperature and atmospheric pressure, the maximum theoretical vapor density may be calculated from the molecular weight of the substance and its vapor pressure at the given temperature. Benzol (C_6H_6), for example, cannot form vapors at room temperature and atmospheric pressure with a density greater than about 1.17 that of air, because of a limited vapor pressure. Carbon dioxide with a vapor pressure exceeding atmospheric produces vapors about 1½ times that of air, the same as vapors of carbon tetrachloride. On the other hand, trichlorethylene which is used in degreasing machines at its boiling point (87 C.) forms vapors *at this temperature* that are 3½ times the density of air *at room temperature,* whereas at room temperature its vapor density cannot exceed 1 1/3 that of air.

In the use of such solvents in practice where evaporation takes place in the open air, mixtures even more dilute than those cited above occur due to slow rate of evaporation relative to velocities of convection currents. Densities of most concentrated vapor-air mixtures greater than 1.01 to 1.02 of air are unusual and even densities of this order are not to be expected except at very short distances from the source.

Floor Ventilation—The belief that vapors of substances of high molecular weight sink rapidly to the floor or other adjacent low point and accumulate there in high concentrations sometimes results in the practice of exhausting air near the floor level in the belief that the vapors are thus more effectively removed. If the quantitative principles of local exhaust or of dilution are applied to the problem, there is no objection to this practice. Serious error is likely to result, however, if an insufficient flow of air is provided to control the vapors by either of the two above-mentioned mechanisms. When large quantities of air must be withdrawn for effective dilution, it may be desirable to exhaust it at points near the floor for the sake of minimizing heating difficulties in cold weather, but, on the other hand, if exhaust inlets located at higher levels can be made to entrain appreciable quantities of vapor by local exhaust and thus reduce amounts to be diluted, this is the better arrangement.

That solvent vapors do not usually collect near the floor in markedly higher concentrations than at breathing level is demonstrated by a series of observations made over a period of time in a number of different industrial establishments and described in Table 5.

TABLE 5.—AVERAGE CONCENTRATIONS OF VAPORS (30-45-MINUTE SAMPLES) IN THE ATMOSPHERE OF VARIOUS INDUSTRIAL ESTABLISHMENTS AT FLOOR LEVEL AND AT BREATHING LEVEL

	CONCENTRATIONS FOUND, P.P.M.	
VAPOR	BREATHING LEVEL	FLOOR LEVEL
Toluol	340	855
Toluol, ethyl acetate, ethyl alcohol	870	1260
Carbon disulfide ...	35	45
Ethylene dichloride..	165	205
Benzol	215	290
Benzol	55	80
Benzol	90	145

Although it will be noted that concentrations are slightly greater at floor level than at the breathing level, the differences are insignificant in most cases. It is furthermore of importance to note that in none of these cases was there any ventilation in the form of floor level exhaust which would decrease the noted differentials.

In one plant where ventilation of this type had been installed, concentrations of vapor were measured in an exhaust duct drawing air from a point near the floor, and in the general air, at breathing level. There was little difference, as follows:

In exhaust duct	235 p.p.m.
At breathing level	215 p.p.m.

The purpose of this discussion is to emphasize the fact that (1) exhausting near the floor should be considered as no more than a system of ventilation by dilution in which supply air entering the room is the phenomenon effecting control, and (2) if a duct system is to be used and the exhaust inlets can be located in the vicinity and at the level of the sources of contamination and thereby capture at least some of the material at its source, this is a better arrangement.

Down-draft ventilation, in which an exhaust inlet is located below the source of contamination, and close to it, as by means of a grating in a work-bench or in the floor, is an excellent arrangement in most cases but it is impossible to treat it as other than one of the two systems of ventilation (usually local exhaust in the form of a flanged exhaust hood in a face-up position).

It is a well-recognized fact that in the heating and air-conditioning field, slight difference in densities of air streams due to temperature differentials are of tremendous importance in the design of air distributing systems. For example, if air supplied to an auditorium through a simple opening at the ceiling be 10F cooler than the main body of air (1.02 times heavier), an increase in velocity of 200-300 feet per minute or greater would be readily attained by the column of cooler descending air when it reaches floor level. Why, then, are such density differentials not of equal significance in the case of solvent vapors in industrial processes?

Theoretical Velocities—The answer is to be found in the great difference in quantities of denser gas involved and in factors brought out in the following analysis. The downward velocity attained by a vapor-air mixture of greater density than the surrounding air is a function of the relative weight of the column of denser mixture which in turn is dependent on the density differential and on the height of the column. Specifically,

$$h = \frac{H}{5.2}(P_1 - P_2) \qquad \ldots\ldots\ldots\ldots (6)$$

where h = pressure head attained, inches of water,
H = height of unbroken column of gas, feet,
P_1 = average density of gas in column, lbs. per cu. ft.,
P_2 = density of surrounding air (0.075 lbs. per cu. ft.), and
5.2 = lb. per sq. ft. per inch of water column.

A vapor-air mixture flowing from a continuously replenished source in a column of height, H, would theoretically attain a velocity corresponding to the calculated pressure head, h. If there were no mixing with air to modify its density in passage downward the distance represented by H, we could take the initial density of the mixture to represent that in the entire column. Actually, this is impossible since entrainment of air is inevitable in such a dynamic system. However, the assumption is convenient in enabling computation of the theoretical velocity in the following situation.

Suppose a vapor-air mixture initially 1.02 times heavier than air, and that nearly still-air conditions permit a vertical column of the mixture to stream downward a distance of 1 foot before being broken up by convection currents. The theoretical velocity head is calculated from equation (6)

$$h = \frac{1}{5.2}(0.075)(1.02 - 1.00)$$
$$= 0.00029 \text{ inches of water.}$$

The velocity corresponding to this head is

$$\text{Velocity} = 4000\sqrt{h}$$
$$= 68 \text{ f.p.m.}$$

As small as this velocity is (natural convection currents are commonly as high as 50 f.p.m.), it is too great because it rests on two invalid assumptions. First, air entrainment would result in dilution of the stream in its downward passage, thereby decreasing the density differential. Second, and more important, are the horizontal convection currents that break the continuity of the vertical stream and thus decrease the effective value of H.

Relative Momenta—The momenta of convection currents that play havoc with the tenuous streams of solvent vapors from an industrial process, are obviously insignificant when compared with that of the great streams of cool air, several inches to a foot or more in diameter, encountered in air-conditioning. An analogy exists in the comparison between rising smoke from a cigarette in a quiet room, and the large quantities of heated air that rise laden with dust from a hot casting during "shaking out" in a foundry. The slightest horizontal air motion breaks up the rising stream from the cigarette tip, whereas nothing short of a strong blast of air can appreciably disturb the rising air currents from a large heated casting.

It is therefore not surprising that the difference in concentrations of vapors at floor level and at breathing level shown in Table 5 are not great, for with the small differences in density of vapor-air mixtures and relatively small quantities of toxic vapors commonly escaping in industrial processes, normal convection currents are able to disrupt effectively the downward tendency and cause mixing with the surrounding air before the downward velocity can attain a significant magnitude. It need hardly be pointed out that once mixture of vapor with air takes place, there is no possibility of subsequent segregation of vapor from air and the effective density is that of the vapor-air mixture.

Slots in Benches—These principles apply as well to the case of benches or work tables equipped with exhaust slots along their perimeter where vapors originate in the central portion of the bench at a relatively great distance from the exhaust openings. Such an arrangement can be treated only as a system either of local exhaust or dilution-ventilation or a combination of the two, and if lesser volumes of air are exhausted than are required on such a basis on the supposition that use can be made of greater density of vapors, unsatisfactory control is likely to result. Examples of this fault were observed during a study of ventilation in fur coat cleaning establishments, and a description of it appears[8] elsewhere. The most important disturbing factor was shown to be the convection currents of air at the source of vapors which are accentuated by movements of workers' hands, material or mechanical equipment. Under conditions of nearly complete stagnation of air, the situation would be different, but this is very rare since such ventilation arrangements are virtually always selected to avoid interference with necessary manipulations on the table. When the exigencies of a process permit no other arrangement, the air requirement should be computed on a basis of dilution unless it appears that the local exhaust mechanism will induce controlling air velocities where necessary with a smaller ventilation rate.

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[8] Hemeon, W. C. L., "Downdraft Ventilation," *Safety Engineering*, April, 1940.

Anthracite Sizes

The accompanying table shows the standard Anthracite Sizing Specifications approved and adopted by the Anthracite Emergency Committee January 8.

When slate content on broken to pea inclusive is less than these standards, bone content may be correspondingly increased, but slate content specified here shall not be exceeded in any event and the total maximum impurities shall not exceed those specified.

STANDARD ANTHRACITE SIZING SPECIFICATIONS

Designation	Test Mesh, In.		Round Mesh			Maximum Impurities	
	Through	Over	Oversize Maximum	Undersize			
				Max.	Min.	Slate	Bone
Broken	4-3/8	3-1/4	—	15%	7½%	1½%	2%
Egg	3-1/4	2-7/16	5%	15%	7½%	1½%	2%
Stove	2-7/16	1-5/8	7½%	12½%	7½%	2%	3%
Nut	1-5/8	13/16	7½%	10%	5%	3%	4%
Pea	13/16	9/16	10%	15%	7½%	4%	5%
Buckwheat	9/16	5/16	10%	15%	7½%	12% Ash	
Rice	5/16	3/16	10%	15%	7½%	13% Ash	
Barley	3/16	3/32	10%	20%	10%		

MIXING OF HOT AND COLD WATERS—1

For many purposes the hot water which is finally used is obtained by mixing hot water with cold water either by passing through a mixing device or by mixing at the point of use. Since, in most operations requiring hot water, the total amount of water is fixed, the amount of hot and cold water required depends on the temperature of both the hot water supply and the cold water supply. This point is discussed on HEATING & VENTILATING's Reference Data Nos. 191-192, and on that sheet also the amounts of mixed water necessary and the temperatures at which the water is used are tabulated.

In making advance estimates to insure an adequate supply of hot water, a logical procedure would call for an analysis of the uses to which the water would be put. Once this is known and a reasonable table of frequency and demand can be set up, the amount of either hot or cold water required can be readily determined. This can be done by calculation in each case or it can be simplified by preparing a table showing the amounts of both hot and cold water at a given temperature required to produce a given amount of mixed water at some intermediate temperature. The table on the other side of this sheet has been prepared to cover the usual range of temperatures at which hot water is supplied, the temperature at which cold water is available, and the temperatures at which the mixed waters are most frequently used.

As mentioned in HEATING & VENTILATING's Reference Data Nos. 191-192, the maximum temperature at which the mixed hot water is used generally does not exceed 120F. Reference to Table 1 on the other side of this sheet shows that when the mixed water temperature is at 120F the number of gallons of hot water which must be added to a gallon of cold water to produce this temperature can vary all the way from 0.22 of a gallon up to 2 gallons. This illustrates very well the tremendous effect of the temperature of the hot and cold water supplies on the demand for hot water.

Table 1 also illustrates the effect of the variation in the number of gallons of hot water required for a given operation between winter and summer conditions. To make this clear consider the following example.

Example: 10 gallons of mixed warm water at 120F is required for a given operation. A hot water supply tank is kept at a temperature of 140F throughout the year while the temperature of the cold water from the city mains is 90F in summer, 50F in winter. How much greater demand is there on the hot water supply to secure the 10 gallons of water in winter than in summer?

SOLUTION: Enter the column headed "Temperature of Mixed Warm Water" and locate the 120F figure. Note that the 140F hot water supply is located at the bottom of the sheet in the center section. Move opposite the 120F of the column marked "Temperature of Mixed Warm Water"; under "140F Hot Water Supply" and under "50F Cold Water Supply," find the figure 3.5. Similarly locate the figure under the column showing the temperature of cold water 90F and here find 1.5. This means that in summer, 1.5 gallons of the 140F hot water must be mixed with 1 gallon of the cold water, and the result will be 2½ gallons of 120F warm water. In winter with 50F cold water, 3½ gallons of hot water are required to produce 4½ gallons of warm water at 120F.

If the Table on this sheet is used in connection with the Table given in HEATING & VENTILATING's Reference Data Nos. 191-192, the quantities of hot water and of cold water to supply warm water for any of the common operations can be readily computed.

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H. & V.'s REFERENCE DATA—194

MIXING OF HOT AND COLD WATERS—1

TABLE 1.—GALLONS OF HOT WATER OF VARIOUS TEMPERATURES WHICH MUST BE ADDED TO ONE GALLON OF COLD WATER TO PRODUCE A MIXED WARM WATER AT A SPECIFIED TEMPERATURE

Temp. of Mixed Warm Water, (T_w)	Temperature of Cold Water Supply, F. (T_c)																				
	40	50	60	70	80	90	100	40	50	60	70	80	90	100	40	50	60	70	80	90	100
	210F Hot Water Supply (T_h)							200F Hot Water Supply (T_h)							190F Hot Water Supply (T_h)						
200	16	15	14	13	12	11	10	—	—	—	—	—	—	—	—	—	—	—	—	—	—
190	7.5	7.0	6.5	6.0	5.5	5.0	4.5	15	14	13	12	11	10	9	—	—	—	—	—	—	—
180	4.67	4.33	4.0	3.67	3.33	3.0	2.67	7.0	6.5	6.0	5.5	5.0	4.5	4.0	14	13	12	11	10	9	8
170	3.25	3.0	2.75	2.50	2.25	2.0	1.75	4.33	4.0	3.67	3.33	3.0	2.67	2.33	6.5	6.0	5.5	5.0	4.5	4.0	3.5
160	2.4	2.2	2.0	1.8	1.6	1.4	1.2	3.0	2.75	2.5	2.25	2.0	1.75	1.50	4.0	3.67	3.33	3.0	2.67	2.33	2.0
150	1.83	1.67	1.5	1.33	1.17	1.0	0.83	2.2	2.0	1.8	1.6	1.4	1.2	1.0	2.75	2.5	2.25	2.0	1.75	1.5	1.25
140	1.43	1.29	1.14	1.0	0.86	0.71	0.57	1.67	1.5	1.33	1.17	1.0	0.83	0.67	2.0	1.8	1.6	1.4	1.2	1.0	0.8
130	1.125	1.0	0.87	0.75	0.625	0.50	0.375	1.29	1.14	1.0	0.86	0.71	0.57	0.43	1.5	1.33	1.17	1.0	0.83	0.67	0.5
120	0.89	0.78	0.67	0.55	0.44	0.33	0.22	1.0	0.87	0.75	0.625	0.50	0.375	0.25	1.14	1.0	0.86	0.71	0.57	0.43	0.29
110	0.70	0.60	0.50	0.40	0.30	0.20	0.10	0.78	0.67	0.55	0.44	0.33	0.22	0.11	0.87	0.75	0.625	0.50	0.375	0.25	0.125
100	0.55	0.45	0.36	0.27	0.18	0.11	—	0.60	0.50	0.40	0.30	0.20	0.10	—	0.67	0.55	0.44	0.33	0.22	0.11	—
	180F Hot Water Supply (T_h)							170F Hot Water Supply (T_h)							160F Hot Water Supply (T_h)						
180	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
170	13	12	11	10	9	8	7	—	—	—	—	—	—	—	—	—	—	—	—	—	—
160	6.0	5.5	5.0	4.5	4.0	3.5	3.0	12	11	10	9	8	7	6	—	—	—	—	—	—	—
150	3.67	3.33	3.0	2.67	2.33	2.0	1.67	5.5	5.0	4.5	4.0	3.5	3.0	2.5	11	10	9	8	7	6	5
140	2.5	2.25	2.0	1.75	1.50	1.25	1.0	3.33	3.0	2.67	2.33	2.0	1.67	1.33	5.0	4.5	4.0	3.5	3.0	2.5	2.0
130	1.8	1.6	1.4	1.2	1.0	0.8	0.6	2.25	2.0	1.75	1.50	1.25	1.0	0.75	3.0	2.67	2.33	2.0	1.67	1.33	1.0
120	1.33	1.17	1.0	0.83	0.67	0.50	0.33	1.6	1.4	1.2	1.0	0.8	0.6	0.4	2.0	1.75	1.50	1.25	1.0	0.75	0.50
110	1.0	0.86	0.71	0.57	0.43	0.29	0.14	1.17	1.0	0.83	0.67	0.50	0.33	0.17	1.4	1.2	1.0	0.8	0.6	0.4	0.2
100	0.75	0.625	0.50	0.375	0.25	0.125	—	0.86	0.71	0.57	0.43	0.29	0.14	—	1.0	0.83	0.67	0.50	0.33	0.17	—
	150F Hot Water Supply (T_h)							140F Hot Water Supply (T_h)							130F Hot Water Supply (T_h)						
150	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
140	10	9	8	7	6	5	4	—	—	—	—	—	—	—	—	—	—	—	—	—	—
130	4.5	4.0	3.5	3.0	2.5	2.0	1.5	9	8	7	6	5	4	3	—	—	—	—	—	—	—
120	2.67	2.33	2.0	1.67	1.33	1.0	0.67	4.0	3.5	3.0	2.5	2.0	1.5	1.0	8	7	6	5	4	3	2
110	1.75	1.50	1.25	1.0	0.75	0.50	0.25	2.33	2.0	1.67	1.33	1.0	0.67	0.33	3.5	3.0	2.5	2.0	1.5	1.0	0.5
100	1.2	1.0	0.8	0.6	0.4	0.2	—	1.5	1.25	1.0	0.75	0.50	0.25	—	2.0	1.67	1.33	1.0	0.67	0.33	—

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NEWS OF THE MONTH

Fuel and Heater Codes Reviewed at 47th ASHVE Convention; Walter Fleisher Elected President, Eastwood 1st V.P.

KANSAS CITY—Well over 400 members and guests registered at the 47th Annual Meeting of the American Society of Heating and Ventilating Engineers held at the Hotel Muehlebach here Jan. 27-29.

Walter L. Fleisher of New York was elected president for 1941. Other officers elected were E. O. Eastwood of Seattle, first vice-president; J. H. Walker, Detroit, second vice-president; and M. F. Blankin Philadelphia, treasurer.

Prof. A. P. Kratz, Urbana; W. A. Russell, Kansas City; L. P. Saunders, Lockport, and C. Tasker, Toronto, were elected three-year members of the Council.

Members elected for three years to the Committee on Research were C. M. Ashley, Syracuse; Prof. M. K. Fahnestock, Urbana; F. C. McIntosh, Pittsburgh; H. K. McCain, Atlanta; and T. H. Urdahl, Washington.

D. W. Tomlinson, vice-president in charge of engineering, Transcontinental & Western Air, Inc., presented a paper on Comfort in High Altitude Flying in which he reviewed the developments in the past decade in the heating and ventilating of airplanes. In addition, the paper featured the heating and ventilating system in the Stratoliners which consist of five separate, although intimately related systems, as follows: (1) Fresh air system, (2) Spent air system, (3) Warm air system, (4) Steam system, and (5) Ground air conditioning system.

Boiler Testing Codes was the subject of a paper by J. F. McIntire, vice-president of the United States Radiator Corporation. Mr. McIntire reviewed the work of the Institute of Boiler and Radiator Manufacturers in testing cast iron boilers and suggested the acceptance of the I-B-R Testing and Rating Code by the ASHVE.

F. L. Meyer, vice-president of The Meyer Furnace Co., presented a discussion of Warm-Air Furnace Codes in which he referred to the work done by the National Warm Air Heating and Air Conditioning Association in setting up codes for both gravity and forced air systems and called attention to the willingness of the Association to cooperate with other groups in any future work which might involve studies of stoker-fired furnaces, furnaces having exceedingly large amounts of heating surfaces, and studies of the effects of casing baffling.

Codes in the Stoker Industry were illustrated and described by E. C. Webb, engineering service manager, Iron Fireman Mfg. Co. He pointed out that the ASHVE Performance Test Code for Steam Heating Solid Fuel Boilers led to the preparation of a similar code for automatic stokers which was adopted in 1938 by the ASHVE and by the Stoker Manufacturers Association. Since, however, boilers and stokers are used together Mr. Webb mentioned that it has been suggested that attention be given to reporting the heat balance to show more clearly the individual performance of the two units.

The speaker also referred to the Stoker Manufacturers Association Stoker Rating Code and the Minimum Setting Heights recommendations of the Stoker Manufacturers Association in cooperation with other groups. He mentioned the work done by the Anthracite Industries Laboratory in connection with anthracite burners.

In conclusion Mr. Webb suggested that the ASHVE give consideration to a review of its Standard Code for Testing Stoker-Fired Steam-Heating Boilers with possible revisions of the outline for reporting the heat balance to show the performance of the stoker and boiler individually.

Friction-Heads in Standard Six-Inch Pipe, a research paper, was presented by retiring president F. E. Giesecke and J. S. Hopper, both of A. and M. College of Texas. The object of the research was to determine friction heads for water of ordinary temperature in 6-in. black pipe, in both welded and screwed elbows, and in screwed couplings and welded joints of 6 in. pipe, with a secondary objective of determining the feasibility of using welded elbows to meter the flow of liquids in pipe.

Walter L. Fleisher, newly elected president of the American Society of Heating and Ventilating Engineers.

The authors conclusions were:

1. The friction-heads in couplings and in well-fabricated welds are very small and may be neglected in friction-head calculation;

2. The friction-head in one foot of a new and clean standard black 6-in. pipeline with a normal number of joints or couplings for water having a temperature of about 95F is $h = 0.00834v^{1.78}$ in. of 70F water, where v is in feet per second;

3. The transition period from laminated flow to turbulent flow in 6-in. pipe seems to extend up to a velocity of about 3 ft. per second. Additional studies would be made to determine the loss of head for velocities in the transition range;

4. The friction-head in one 6-in. screwed elbow is $h = 0.1101v^{1.873}$ in. of 70F water, where v is in feet per second, when the temperature of the water flowing through the elbow is about 95F, and when the lengths of the connecting pipe lines are measured to the intersection of their axes. For the same conditions, the friction-head in one welded elbow is $h = 0.02127v^{2.275}$ in. of 70F water;

5. A welded elbow can be used as a flow meter by measuring the difference in the static pressures between the concave and the convex sides and at the center of the elbow. This measurement is preferably made by means of an inclined manometer; and

6. Additional research should be conducted to determine the effect of water temperature and also the effect of incrustation resulting from corrosion on the friction-head in pipes and fittings.

In a paper entitled "Radiation as a Factor in the Sensation of Warmth" F. C. Houghten, S. B. Gunst and J. Suciu, Jr., reported on tests in two rooms, one heated by a warm air heating system and the other by a hot water radiator plant.

Observations were made of the sensations of draft, coolness and warmth for the two types of heating. It was found that in the room heated by radiators a lower effective temperature could be maintained because of the warming effect of the direct radiation from the radiators. The average effective temperature for comfort in a room heated by radiators was 68.6F, while in the room heated by warm air the effective temperature for comfort was 69.4F. On the other hand, the heat loss for the radiator heated room was found to be about 8% higher than for the convected heat room.

The last part of this paper discussed the effect of mean radiant temperature on the feeling of comfort.

E. J. Rodee, engineer of the John B. Pierce Laboratory of Hygiene presented a paper summarizing "Operating Results of a Residence Radiant Wall Heating System," which dealt with the system in a house in Connecticut using convector type units in the base of stud spaces for heating the walls. The system was installed without increasing the wall thickness and without adding the normal amount of insulation. The conclusion was that the costs of operation compared favorably with standing radiators. The author pointed out that the temperature difference between floor and ceiling was materially less with the radiant system than in the case of free standing radiators. No significant saving of fuel was noted during the hours when the sun was shining.

"Comfort Requirements for Low Humidity Air Conditioning," was the title of a research paper presented by F. C. Houghten, H. T. Olson, and S. B. Gunst, all of the ASHVE Research Laboratory in Pittsburgh. The study indicated that a possibly lower effective temperature requirement of 1F at 30% relative humidity was possible when compared with the humidity at 70%. However, as the authors pointed out, the difference was barely significant and the relative lower optimum effective temperature indicated could have been attributed to the prevalence of relatively cold weather conditions during the past summer when the test was conducted.

The result of research sponsored by the ASHVE in cooperation with the College of Medicine at the University of Illinois was reported in a paper on "The Influence of Physiological Research on Comfort Requirements," by Doctors R. W. Keeton, F. K. Hick, and M. M. Montgomery, and Nathaniel Glickman, all of the University of Illinois. Generally speaking, this paper was primarily a review of the literature on the subject, including previous papers by the authors.

W. L. Fleisher and his son W. L. Fleisher, Jr., presented a paper entitled, "Comfort and Health and Temperature — a Mathematical Solution," pointing out that in the area covered by the comfort zone, when plotted on the conventional psychrometric chart, the effective temperature lines run practically parallel to the constant specific volume lines. The fact that the effective temperature lines are essentially parallel to the constant volume lines may be due to experimental errors in the determining effective temperature lines, but probably this is not the case, the authors believe. They have ingeniously tied in this relationship with theories of kinetic energy of gases in order to relate comfort to the physical matter or molecules which constitute the atmosphere.

Reporting the work in the Research Residence in the University of Illinois in the summer of 1940 Professors A. P. Kratz and S. Konzo, and E. L. Broderick, all of the University of Illinois, presented a paper entitled "Summer Cooling in the Research Residence with a Gas-Fired Dehydration Cooling Unit." Previous studies covered the use of ice and a mechanical condensing unit for cooling the research residence. The present study included the performance and operating characteristics of a gas-fired dehydration cooling unit.

The authors concluded that:

(1) The method of operation by which the cooling plant was started when the indoor effective temperature rose to 74F after a period of supplementary cooling with outdoor air at night, was more satisfactory than the one by which the cooling plant was started when the indoor dry-bulb temperature rose to 81F, in that it eliminated the period of discomfort occurring just previous to the start of the cooling unit;

(2) By the use of the dehydrating unit it was possible to maintain comparatively uniform relative humidity, and consequently uniform comfort conditions, irrespective as to whether the plant operated continuously or intermittently;

(3) Use of insulation in the sidewalls and ceiling effected a saving of approximately 33% in the sensible heat load;

(4) Lack of agreement between the calculated design moisture load and the actual moisture load indicated that moisture permeating the structure may be independent of the amount of infiltration or ventilation air, and suggests that some revision may be advisable in the commonly accepted method for calculating moisture load.

"A Laboratory Method for Cyclic Heat Measurements on Walls and Roofs," was proposed in a paper of that name by E. R. Queer, Assistant Professor of Engineering Research, and F. G. Hechler, Director, Engineering Experiment Station, both of The Pennsylvania State College. The method was developed in connection with work at Penn State for the Navy.

S. F. Nicoll, Air Conditioning Application Engineer, York Ice Machinery Corp., was the author of a paper entitled "Reheating by Means of Refrigerant Compressor Discharge Gas." Typical examples of a number of ways of regulating the capacities of air conditioning equipment and of alternating the relative proportions of sensible and latent heat removal were illustrated in the paper. Compressor discharge gas reheat may be used to neutralize the cooling effect when dehumidification alone is required, and may be used completely or partially to off-set sensible cooling.

At the Research Laboratory of the ASHVE in Pittsburgh, two cubicles for studying sun effect were built during 1939 on a turntable on a roof so that the cubicles could be made to face any horizontal direction. Use has been made of this arrangement to compare "Heat Gain Through Western Windows With and Without Shading." Information on this was presented by F. C. Houghten, and David Shore, of the ASHVE Research Laboratory.

AT THE 47TH ANNUAL MEETING OF THE ASHVE

(Left to right) Edward M. Jolly, The Marley Co., secretary, Kansas City chapter; John M. Arthur, Jr., Kansas City Power and Light Co., chairman of the committee on arrangements; Gustave Nottberg, U. S. Engineering Co., vice president, Kansas City chapter; William L. Cassell, consulting engineer, president, Kansas City chapter; Dr. F. E. Giesecke, A & M College of Texas, retiring National president; Walter L. Fleisher, New York consulting engineer and new national president of the ASHVE; Roger P. Campbell, E. K. Campbell Co., treasurer, Kansas City chapter, and E. O. Eastwood, University of Washington and new first vice president, ASHVE.

Braatz Addresses A.C. Council

BUFFALO, N. Y.—One of the largest gatherings of air conditioning men here in recent years attended the February meeting of the Air Conditioning Council of Western New York in the University Club. Nearly 100 representatives of all branches of the trade were at the dinner meeting.

Richard H. Mollenberg, president of the Council, presided. He brought up the matter of pending city legislation which would require the licensing of all stationary engineers. Mr. Mollenberg explained that the Air Conditioning Council, co-operating with other interested associations, had been successful in having the legislation tabled last year but that it is turning up again at this time.

Walter P. Davis, executive secretary of the Council, reported that the Buffalo Chamber of Commerce has arranged a series of meetings to rally interests opposed to the bill and that an alternative bill will be drawn up by these groups. Mr. Mollenberg appointed a committee consisting of William A. Meiter, Roswell Farnham, Joseph Davis and Walter Davis to represent the Air Conditioning Council in the fight to defeat this legislation which he described as detrimental to the interests of the air conditioning industry.

Guest speaker of the evening was Chester J. Braatz, sales manager of the Barber-Colman Company, Rockford, Ill.

Mr. Braatz discussed "Air Distribution." He said the distribution of air is a comparatively new subject, and is one that can do a great deal to compensate for other deficiencies in an air conditioning system. "To the man on the street, the air distribution really constitutes the air conditioning system," he added.

He outlined research work that led up to air distribution with emphasis on the variables that determine the performance of grills and registers of the side-wall and ceiling type. He recommended a procedure for studying variables, studying in the laboratory and presentation of data collected into a form useful in the solution of problems.

NBFU Celebrates Its 75th Year

NEW YORK—Commemorating the seventy-fifth anniversary of its founding in 1866, the National Board of Fire Underwriters this year will conduct an intensive campaign to stimulate country-wide recognition of the increased need for fire prevention as a contribution to the National Defense program. The anniversary program will reach its climax at a two-day meeting at the Waldorf-Astoria Hotel in New York on May 27-28, when leaders in many fields of endeavor will emphasize the part that the National Board of Fire Underwriters has played in progress.

Phillips Now President of NDHA

PITTSBURGH—Following the resignation of F. L. Witsell as president, L. S. Phillips, who has been first vice president, has been advanced to the office of president of the National District Heating Association.

Mr. Phillips is manager of Commercial Relations of the New York Steam Corporation. He is a native of Pennsylvania, attended Rayen High School in Youngstown, Ohio and was with the Tata Iron and Steel Company when the United States entered the World War. He returned home by way of China and Japan in March, 1918 and in September of that year left for France in the United States Engineering Corps. He was graduated from Syracuse University in February, 1924 cum laude.

L. S. Phillips, new president of the National District Heating Association.

Mr. Witsell, who became president of the National District Heating Association in 1940, was, at that time, with The Toledo Edison Company. Since then he has been made president and general manager of The Pueblo Gas and Fuel Company, Pueblo, Colorado, and for that reason has resigned as president of the Association.

John M. Arthur, Jr., Kansas City Power and Light Co., Kansas City, Mo., has been advanced to first vice president of the Association and Raymond M. Nee, Boston Edison Company, has been made second vice president.

Rock Wool Ass'n Elects Officers

NEW YORK—Mineral wool sales for the third quarter of 1940 registered a 35% gain in dollar volume over the same period of 1939, according to Wharton Clay, secretary of National Mineral Wool Association.

Officers of the association elected for the coming year are: President, M. L. Kuykendall of Insulating Industries, Inc., Detroit; vice president, E. R. Stevens, Baldwin Hill Co., Trenton; treasurer, E. I. Williams, Riverton Lime & Stone Co., Riverton, Va.

Disney Studio Visited by Engineers

BURBANK, CALIF.—A trip through the Walt Disney Studios in Burbank for an inspection of the air conditioning plant that serves the home of Donald Duck, Mickey Mouse and their associates, featured the February 13 meeting of Los Angeles Chapter ASHVE.

William Garity, chief engineer of the Disney Studios, explained the engineering details as the group passed from one department and from one building to another on the 50 acre studio lot. Of particular interest were the zoning features of the Animation Building, in which 156 individually controlled zones illustrated how minutely and individually the air is controlled for specific needs. The heating engineers were also escorted through the four other larger buildings for which air conditioning is handled from a central refrigeration plant, and five smaller buildings which are equipped with individual conditioning plants. The pneumatic control system, the visitors were shown, is supplied with compressed air from a 25 h.p. compressor which also supplies the compressed air needed for other purposes in the studio.

Of interest, too, was the central plant which consists of 12 individual units, each with its own 50 h.p. refrigeration machine. Total cooling effect, including well water, is 1,400 tons.

Plans Proceed for West Coast Show

SAN FRANCISCO—The Pacific Coast region will preview this year many noteworthy advances in indoor climate control for human comfort and industrial protection when the heating, ventilating and air conditioning industry convenes here for a series of engineering meetings in conjunction with the Pacific Heating & Air Conditioning Exposition, to be held in the Exposition Auditorium, June 16-20.

Reports received by the International Exposition Company, which has been conducting similar expositions biennially for a number of years, show that a substantial proportion of manufacturers are making product innovations at this time and are planning to participate in the Pacific Air Conditioning Exposition, many already having made reservations. About three-fourths of the exhibition space offered has already been booked, according to Charles F. Roth, president of the Exposition Company and manager of the Exposition.

The forthcoming Exposition, under the auspices of the ASHVE, will be an intermediate in the biennial sequence of heating and ventilating expositions started in 1930, and is to be held in conjunction with the summer meeting of that Society. The Heating, Piping and Air Conditioning Contractors National Association will also meet during the same period.

Minneapolis-Honeywell Launches Drive to Sell Automatic Heat; Campaign Will Aid Dealers in Competition from Outside Industries

MINNEAPOLIS—A mass drive to sell the broad idea of automatic heating to the public, behind which will be its entire 1941 advertising appropriation, has been launched by Minneapolis-Honeywell Regulator Company. The campaign will feature automatic heat irrespective of fuel, with the company's own part in the picture largely subordinated. Automatic heat, a survey showed, is not a saturated market by any means, but unfortunately has lacked a unified sales campaign which can only be conducted by an organization or group interested in the broad picture and not in any one fuel.

With impressive numbers of oil burners, gas burners, boilers and furnaces and stoker units being sold each year, some of the more pessimistic among those in the heating industry have begun to feel that the automatic heat market is rapidly approaching the saturation point. Some surveys have indicated that this is not the case, but most such surveys have been limited in scope.

Consequently, last December Minneapolis-Honeywell employed a nationally known research firm to study this question in the principal heating markets. The homes covered by the survey were those valued up to $10,000 but having not over eight rooms, and owner occupied. The basic facts brought out by the survey were:

1. Over 54% of homes, valued up to $10,000, *do not have* automatic heating.
2. Over 97% of these home owners think automatic heating is "Too Expensive" or, they are satisfied to "get along" without it.
3. Over one-third of these home owners do not realize that easy terms (FHA or others) are available for automatic heating—and the rest of them have not been impressed or influenced by easy terms.
4. There is a tremendous potential market for replacement sales. (Among oil burners alone, one estimate is that over 377,000 installations are ten or more years old.)

All of this can be still further boiled down into a brief statement that, although automatic heating is definitely a contribution to better living, millions of American homes are still without it.

Acknowledging the truth of this statement, the management of Minneapolis-Honeywell has come to the conclusion that automatic heating, as such (not individual units), has neither been sold hard enough nor widely enough, nor have all factors in the industry faced squarely its competition.

This competition does not primarily arise within the industry but springs from other devices such as the automobile, radio, electric refrigerator, and washing machine. All of these have been in direct competition for the customer's dollar, and in order to get it have been very widely merchandised and nationally advertised during the period when the automatic heating industry was making rapid strides in the improvement of its own products.

These improvements, however, have not been backed by the same merchandising pressure nor the same millions of dollars in advertising money as their competitors, for obvious reasons which are no fault of the automatic heating industry. The automatic heat industry is made up of many manufacturers and many diverse interests, so that obviously it would be extremely difficult for it to organize to promote automatic heating in general.

MINNEAPOLIS-HONEYWELL EXECUTIVES STUDY CAMPAIGN

Executives of Minneapolis-Honeywell Regulator Company look over the summary of the company's advertising campaign to sell the broad idea of automatic heating to the public. (Left to right, standing) George B. Benton, advertising manager; J. W. Pauling, vice-president, and C. B. Sweatt, vice-president in charge of sales; (Seated) H. W. Sweatt, president and general manager.

However, since Minneapolis-Honeywell serves the entire industry, it is in an ideal position to do this much needed job. Since M-H controls are used with all types of fuel and all types of heating systems, the company has no axe to grind and no prejudice in any single direction.

As a result of the survey, Minneapolis-Honeywell decided that it would devote its entire advertising appropriation in 1941 to a mass drive to sell automatic heat to the American public. The campaign, which will embrace six national magazines, nation-wide posting in selected centers on billboards, advertising in over thirty business papers, with dealer helps embracing direct mail, newspaper advertisements and mats, spot radio campaigns and window and counter display material, will start this month and continue throughout the year.

The important objectives which the campaign will stress are: (1) The tremendous improvement in all types of automatic heating in recent years; (2) Low cost of installing automatic heating; and (3) Easy terms available under FHA and through other sources for financing its purchase, making it possible for any home owner now to enjoy its luxury and economy.

The campaign will be entirely financed by Minneapolis-Honeywell.

Monroe Speaks in Pittsburgh

PITTSBURGH—The Pittsburgh chapter of the ASHVE met February 10 with R. W. Monroe, Motor Sales Division, Westinghouse Electric & Mfg. Company, as the speaker of the evening. The subject of discussion was "Electric Motors and Control Equipment." Illustrated slides accompanied the lecture which dealt with the general history concerning motor development, the torque characteristics of the many types of industrial motors, with special emphasis on motors used on fans, pumps and compressors.

Quiet operating motors are being ordered by the industry for which there is an extra charge of 15%. These motors are machined for precision clearances with close tolerances on bearings, parts and with balanced rotors.

More Defense Housing Sought

WASHINGTON — An emergency program calling on Congress for two supplementary appropriations totaling $156,750,000, to provide homes for workers in defense areas, and for an amendment to the National Housing Act designed to expedite participation by private industry in the government's defense housing effort, was announced Feb. 5 by C. F. Palmer, Defense Housing Coordinator.

One proposed appropriation, amounting to $150,000,000, would be devoted to construction in sixty defense areas of 37,000 dwelling units for families of defense workers and enlisted personnel. This fund would be at the disposal of the President for allocation.

Buildings suitable for single men as well as families would be built. Mr. Palmer said that many workers in some production centers and other defense areas were young unmarried men, 17 to 25 years of age, and that for them quarters along the lines of a college dormitory would be provided.

The other appropriation, of $6,750,000, would be used to provide temporary shelters for short-term use in areas where the demand was urgent and immediate action was required.

To deal with such situations, Mr. Palmer said, mobile housing units, of an improved trailer type, would be constructed, and unseaworthy passenger vessels would be reconditioned and used as "floating hotels" at Southern and West Coast ports. He estimated that 20,000 persons would be provided for by these "flying squadrons," including 5000 single men, about 3000 of whom would be housed in the proposed "floating hotels." He said the Maritime Commission had made a survey of available vessels and now was preparing for such use one ship having 500 staterooms, in each of which two men would live comfortably for the duration of a port job.

The proposed National Housing Act amendment, according to Mr. Palmer, would set up a separate housing insurance fund of $10,000,000, which would be used to underwrite $100,000,000 in mortgages on one to four-family dwelling units in defense areas. Under this amendment loans to builders would be insured up to 90% of the Federal Housing Administrator's appraised value of the property where the value of such dwellings ranged from $4000 on a single family residence to $10,500 on a four-family apartment building. Such houses could be sold with no down payments or could be offered for rent, he said.

At present mortgages up to 90% of the appraised value can be insured only in cases where the builder is the owner occupant and is able to provide 10% equity in cash.

Houghten Honored by Alma Mater

OLIVET, MICH.—Ferry C. Houghten, Pittsburgh, director of the ASHVE Research Laboratory, and a graduate of Olivet College, class of 1913, was awarded the honorary degree of Doctor of Science by his alma mater at its annual founders day convocation here February 25.

The citation for the honorary degree emphasized the scientific scholarship, leadership, and significant contributions of the recipient in the field of science particularly in the area of heating, ventilating and health problems of the home.

Speaking on the topic "The Challenge for Social Guidance in our Education," Dr. Houghten in the convocation address declared that the work of the social scientist dealing with human behavior has not kept pace with the work of the men of science.

"During the past decade grave questions involving such social agencies as governments, economic systems, and social orders find us in a state of turmoil, insecurity, and at times in dire want, literally smothered in material things resulting from our technology and mass production," he declared.

The speaker advocated a broadening of the educational background of the engineer to include a greater appreciation of the social aspects, and a parallel broadening of the educational background of the social scientist to include some aspects of pure and applied science and to give an appreciation for the applications of the engineer in the development of material things and services.

Following his graduation from Olivet College in 1913, Dr. Houghten received a teaching fellowship in Physics at the University of Washington, where he received a master of science degree in 1915. In 1918 he accepted a position as physicist with the Bureau of Mines. In his connection with the Bureau's Pittsburgh experiment station, he had charge of the general physical laboratory.

In 1920 Dean John R. Allen, first director of the ASHVE laboratory, appointed Mr. Houghten to head up important phases of research in the Society's laboratory. This work continued until 1924 during which time Mr. Houghten was instrumental in developing the effective temperature chart, studies in the flow of steam in pipes, infiltration, and heat transfer through building materials, which resulted in a number of technical papers being presented before the Society. In 1924 he became national secretary of the ASHVE, a position he held for approximately two years when he was returned to the laboratory as director.

Ferry C. Houghten

Under Mr. Houghten's directorship the Laboratory has continued to serve as one of the most important branches of the Society's activities, carrying on research in many phases of the general subject of heating, ventilating, and air conditioning. As a result of his work at the Laboratory he has been author or co-author of over 80 papers presented before the Society.

During this time an important phase of the Laboratory work dealt with the relation of man and his physiological and comfort reactions to his atmospheric environment. Many articles resulting from this important phase of research appeared in such medical journals as *The American Journal of Physiology*, *The American Journal of Hygiene*, *Industrial Medicine*, and *Journal of the American Medical Association*.

Mr. Houghten has also served on important committees of the American Society for Testing Materials and the National Research Council. He holds a commission as lieutenant commander in the U. S. Naval Reserve attached to the Bureau of Construction and Repair as a specialist in engineering.

Gas Heating Up in New Jersey

NEWARK, N. J.—Continued gains in gas heating installations in New Jersey are revealed by the 32nd annual report of Public Service Corporation of New Jersey, covering the year 1940.

"Gas sales for building heating again registered a substantial gain over previous years," the report commented. "They were 3,851,887,500 cubic feet, an increase of 30.84% over 1939. In 1940 building heated installations increased 2,712, constituting the largest increase in any one year. There were 13,332 installations on the mains at the end of 1940."

Increased air conditioning installations also were noted by the report, which revealed that summer cooling and ventilating installations during 1940 totalled 302 as compared with 254 in the preceding year.

Midwest Stoker Men Elect Burns

CHICAGO—The Midwest Stoker Association at its annual meeting January 28 elected Mount Burns, manager of the Chicago retail stoker division of Link-Belt as president for 1941 to succeed L. G. Briggs.

Other officers, re-elected, were E. M. May, Chicago branch manager of the Combustioneer Div., Steel Products Engineering Co., vice president, and E. W. Jones, office manager in Chicago for the Iron Fireman Mfg. Co., secretary-treasurer. Marc G. Bluth was reappointed manager of the Association, which is a retail organization of Chicago sales agencies.

N.Y.C.R.R. Conditions 120 Cars

ALBANY, N. Y.—New York Central Railroad officials testified recently before the Public Service Commission that "rapid progress" had been made in remedying conditions which brought complaints concerning the lack of cleanliness of both Pullman and coach cars. In connection with the hearing, A. H. Wright, vice-president and general manager, said that before 1941 the road had air-conditioned 155 diners and 263 coaches at a cost of $3,618,000 and had under way a program to air-condition ten diners and 110 coaches at a cost of $1,160,000.

Kirsten Addresses ASHVE Chapter

SEATTLE—Prof. Frederick E. Kirsten, University of Washington, was the chief speaker on "Research in the Problem of Dust Collection" at a meeting of the ASHVE Pacific Northwest chapter here recently. He explored the subject of dust collection and dust control from every angle.

Petersen Elected by Fan Group

ST. LOUIS—The Propeller Fan Manufacturers Association held its annual meeting January 9 at the Hotel Statler here. The following were elected to office for the coming year: E. W. Petersen, American Blower Corp., president; A. R. Stephan, DeBothezat Ventilating Equipment Division, vice-president; and V. C. Shetler, Detroit, Secretary-treasurer.

It was reported at the meeting that architects, engineers, utility companies and consumers are insisting that all fans bear the Certified Rating Label, and that the public is becoming more and more conscious of the necessity of good ventilation.

Members are cooperating to the fullest extent to aid in the national defense program and are increasing production to keep pace with the increased demand for ventilating fans. It appears that 1941 will be the greatest production year in the history of the fan industry.

Would License A.C. Contractors

ALBANY, N. Y.—Contractors engaged in construction, installation, alteration, maintenance and repair of air conditioning systems would be licensed under terms of a bill introduced in the State Legislature here by Assemblyman Robert J. Crews, Brooklyn Republican.

Similar to licensing bills unsuccessfully sought here last year and in other former sessions, the measure also provides for inspection of air conditioning systems by local boards subject to rules and regulations adopted by a five-member state board to be appointed by the Governor.

Air Hygiene Expands Activities

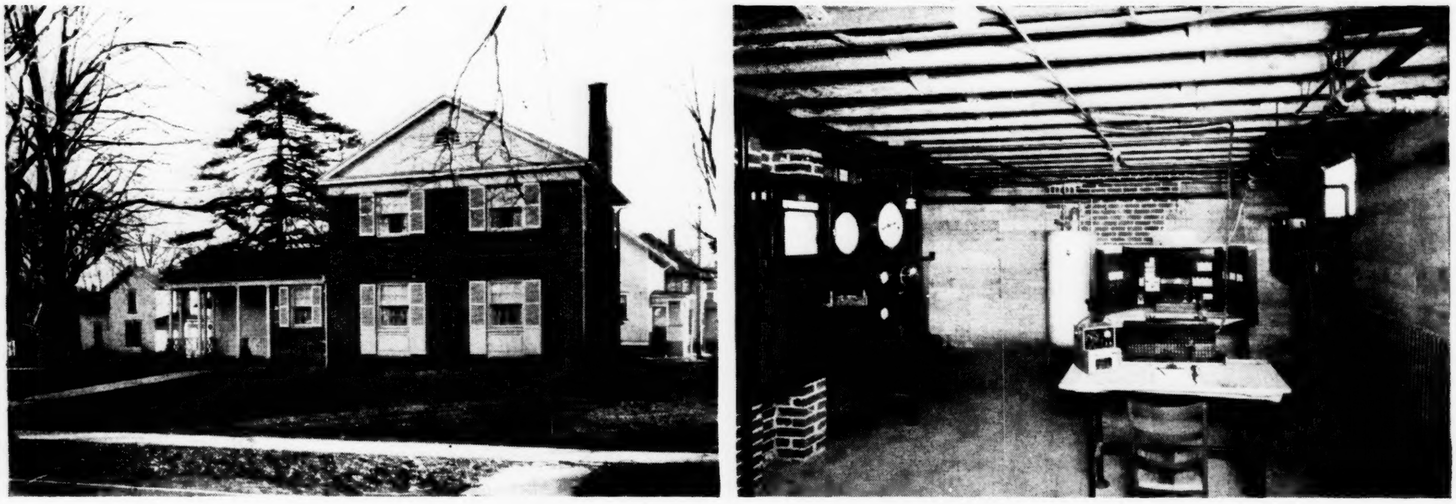
PITTSBURGH—Dr. H. B. Meller, managing director, announces the election of V. P. Ahearn of Washington as secretary of the board of trustees of Air Hygiene Foundation, and of Theodore C. Waters of Baltimore to the board as general counsel. John F. McMahon was promoted to executive secretary. The Foundation, located at Mellon Institute, is a non-profit organization of industrial concerns for the conservation of employee health.

In making this announcement Dr. Meller declared that the arming of America, demanding new production highs, has served to spur health protection in the industries as never before. He warned that "increased production increases occupational health hazards and demands increased precautions.

Dr. Meller reported a marked increase in the industrial hygiene surveys which the Foundation makes at the plants of its member companies to combat possible health hazards. He added that the medical and engineering researches which the Foundation supports at Harvard, Saranac (N. Y.) Laboratory, and University of Pennsylvania are being elaborated to embrace industrial health projects of immediate practical benefit to the industries in the national emergency.

Naval Ventilation Described

SEATTLE—Capt. H. A. Garrison, senior officer of the 13th Naval District at Seattle, speaking on "The General Aspects of Ventilation on Shipboard," interested a large audience of engineers in the auditorium of Guggenheim Hall on the campus of the University of Washington, late in January.



THE I-B-R RESEARCH HOME AT URBANA, ILL.

At the left is an exterior view of the I-B-R Research Home which is financed by 13 manufacturers of cast iron boilers and radiators who are members of the Institute of Boiler and Radiator Manufacturers. (See HEATING AND VENTILATING, *November and December, 1940). At the right is a view of the basement. The instrument panel at the left contains a ten-point temperature recorder which will be used to record wall gradient temperatures, a stack temperature recorder, a CO_2 recorder, and instruments for indicating burner and circulator operating time and power consumption, etc. The table in the center foreground contains the panel and instrument used in reading the moisture points located in all parts of the house. The lead wires from the moisture measuring stations are cabled together and can be seen running from the panel to the ceiling and over to the wall. The switchboard immediately beyond the moisture panel is the thermocouple switchboard at which point all the thermocouples in the house are read.*

Johnson Elected by A.C. Bureau

Boston—At the eighth annual business meeting of the Air Conditioning Bureau, January 30, the following officers were elected for 1941: President, E. Daniel Johnson; vice president, Forrest V. Paige; secretary, Earl G. Carrier; treasurer, Daniel Ricker; directors to serve three years each: E. V. Wetmore and Frederic L. Oliver.

Guest speaker was Holcombe J. Brown, president of the Engineering Societies of New England. In his talk, Mr. Brown admonished air conditioning men not to oversell air conditioning, as done in a number of cases in the country. The air conditioning industry has enough to offer without promising too much. He brought up the question of gas attacks should this country be invaded, and noted that most fresh air intakes are located near the ground, exposing them to indrawing of gas from bombs to the air conditioning system, endangering occupants of the building. He suggested that air conditioning engineers, heating and ventilating engineers make a survey of air conditioned buildings to see if something could not be done to remedy this situation, so that an air conditioning system would not have to be shut down in case of a gas attack.

Fuller Gives Talks on Motors

Boston—As a result of the interest aroused in his simple demonstration of the fundamentals of motor operation before a recent meeting of the Worcester Chapter of Refrigeration Service Engineers, R. A. Fuller, General Electric motor engineer, repeated the lecture before the Boston chapter of the same society on February 10 and before the Boston chapter of the American Society of Refrigerating Engineers February 21.

Adopting the premise that the principles of direct-current, single-phase, and polyphase electric motors can be easily understood, Mr. Fuller uses a laboratory demonstration of electric motor fundamentals employing charts and simple operating devices to show how the laws of magnetism are utilized to make motor operation possible.

Phila. Oil Burner Show Sold Out

New York—All space has been taken in the National Oil Burner Progress Exhibition, to be held at Philadelphia March 17-22, under auspices of Oil Burner Institute.

The 1941 Exhibition will far surpass in size, in the number of exhibits and in the number of exhibitors, any of the long line of great industry shows sponsored by OBI. The space taken will occupy 183 booths by 102 paid exhibitors.

Engineers' Function in Wartime

Los Angeles—Basil E. Rice, director of co-ordination of the Major Disaster Emergency Council of Los Angeles, outlined the part refrigeration and heating engineers will play in the comprehensive county-wide plan that has been evolved by the Council in the event of a major disaster occurring in Los Angeles or its environs, at a joint meeting of the Los Angeles chapters of the National Association of Practical Refrigeration Engineers and the ASRE February 12.

All elements in Los Angeles city and county, Rice explained, will function as an organized army to perform the duties that have been mapped for them by the Major Disaster Emergency Council. Refrigeration and air conditioning engineers will be called to duty by the Refrigeration Industry Committee for handling matters involving broken ammonia lines and the like; determining where broken gas or ammonia lines are located; type of dangerous gases released; and how quickly lines can be repaired or shut off.

The function of heating engineers in times of disaster have also been planned in minute detail, Mr. Rice declared. Heating engineers will serve under the sub-committee of the heating industry, whose duties will pertain to the examination for possible damage of boiler plants, steam and hot water lines. They will also serve as consultants and squad leaders in performing repairs or shut-offs on heating plants that threaten life and property.

Oil Burners Dry Out Earth Dam

Seattle — A most unusual heating project is being carried out by the Hurley Engineering Co. of this city. It involves the installation of a battery of oil burners for heating 425,000 cu. yd. of dirt at Mud Mountain Dam. The core of the dam will be of earth which, before it is packed with a rock filling, must be reduced to 10% moisture content. To accomplish this three oil burners, which will burn 900 gal. of fuel oil per hr., are being furnished to dry out the earth.

Oil Burner Group Changes Name

Buffalo, N. Y. — The Buffalo Oil Burner Association has changed its name to the Oil Heating Association of Western New York. This action was taken by the association's board of directors following the February meeting.

The change in name was made because many oil burner men in Niagara Falls, Tonawanda, Dunkirk, and Jamestown wished to join the Buffalo group.

Guest speaker at the meeting was Frank Weber, Hoffman Specialty Company, who spoke on "Why and Where to Use Traps."

U.S. Gov't Seeks Building Experts

Washington — An examination has been announced by the U. S. Civil Service Commission to secure superintendents of general construction for work in the national defense program. Broad and responsible experience is required, and qualified persons are urged to file their applications at the Commission's Washington office where they will be rated up to December 31, 1941.

There are several grades of positions with salaries ranging from $3200 to $5600 a year, less a 3½% retirement deduction. In general, the duties involve the direction of foremen, laborers, and mechanics on large construction projects. Appointees will inspect materials and workmanship to see that they conform to specifications and will organize men and materials for efficient construction operations.

Competitors must have had progressive experience in the field of general construction. Part of this experience must have been as superintendent on large projects involving excavation, reinforced concrete, steel, wood, and masonry, and supervision of three or more foremen of different building or construction trades. Engineering courses completed at a college or technical institute may be substituted for part of the general experience.

Further information and application forms may be obtained from the Secretary of the Board of U. S. Civil Service Examiners at any first- or second-class post office, or from the U. S. Civil Service Commission, Washington, D. C.

Republic to A.C. Blast Furnace

Birmingham, Ala.—Air conditioning has so conclusively proven its value in blast furnace procedure, that a contract has just been awarded by the Republic Steel Corporation to the Carrier Corporation for the conditioning of the fourth iron smelter in this district.

A. R. Acheson

Syracuse, N. Y.—Professor Albert R. Acheson, head of the Department of Mechanical Engineering in the College of Applied Science at Syracuse University, and a consultant in heating, ventilation and steam generation, and identified with the perfection of The Associated Press wire-photo, died of a heart attack February 25. He was 58.

Born in Riverton, New Zealand, he graduated from Canterbury College, New Zealand University.

He became associated with the Westinghouse Airbrake Company at Addington, New Zealand, in 1904 and 1905 and later was with the Wellington & Manawatu Railway Company.

Besides his widow, Professor Acheson leaves two sons, two daughters, two brothers, and three sisters.

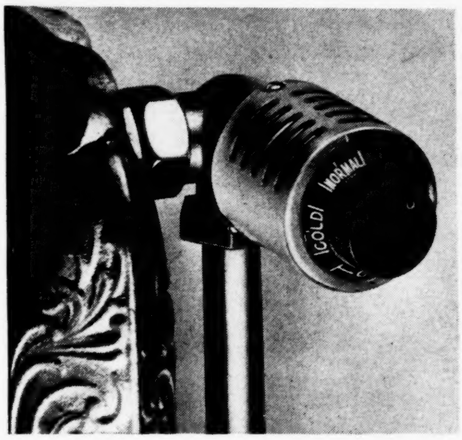
Number of Degree-Days for January, 1941

HEATING & VENTILATING continues its thirteenth year of publishing degree-day data for various large cities. Forty-four cities have been added to the list of those previously published. A total of 116 cities are listed below.

	Degree-Days Jan., 1940	Degree-Days Jan., 1941	Deg. Days Cumulative, 1940-41
Albany, N. Y.	1549	1447	4147
Alpena, Mich.	—	1371	4181
Atlanta, Ga.	1109	658	1696
Atlantic City, N. J.	—	951	2614
Baker, Ore.	—	1014	3652
Baltimore, Md.	1208	940	2468
Binghamton, N. Y.	—	1282	3821
Birmingham, Ala.	1068	569	1438
Bismarck, N. D.	—	1655	4823
Boise, Idaho	—	982	3270
Boston, Mass.	1303	1228	3406
Buffalo, N. Y.	1454	1258	3667
Burlington, Vt.	1654	1601	4620
Cairo, Ill.	—	815	2152
Canton, N. Y.	—	1637	4777
Charles City, Iowa	—	1384	4107
Charlotte, N. C.	—	699	1804
Chattanooga, Tenn.	—	744	2115
Cheyenne, Wyo.	1435	1078	3728
Chicago, Ill.	1511	1145	3265
Cincinnati, Ohio	1426	996	2700
Cleveland, Ohio	1410	1097	3136
Columbia, Mo.	—	964	2696
Columbus, Ohio	1466	1036	2911
Concord, N. H.	—	1384	4119
Concordia, Kan.	—	1091	3068
Davenport, Iowa	—	1178	3287
Dayton, Ohio	—	1075	2979
Denver, Colo.	1264	950	3033
Des Moines, Iowa	1694	1223	3438
Detroit, Mich.	1431	1204	3568
Dodge City, Kan.	1497	973	2696
Dubuque, Iowa	—	1244	3641
Duluth, Minn.	1764	1602	4900
Eastport, Me.	—	1427	4388
Elkins, W. Va.	—	1051	3096
El Paso, Tex.	696	567	1499
Erie, Pa.	1427	1134	3301
Escanaba, Mich.	—	1381	4342
Evansville, Ind.	1411	927	2561
Fort Smith, Ark.	—	676	1795
Fort Wayne, Ind.	1564	1167	3347
Fort Worth, Tex.	935	523	1401
Fresno, Calif.	—	—	—
Grand Rapids, Mich.	1395	1203	3564
Green Bay, Wis.	1585	1370	4161
Greensboro, N. C.	—	839	2272
Harrisburg, Pa.	1336	1097	3110
Hartford, Conn.	1416	1282	3674
Helena, Mont.	—	1319	4340
Huron, S. D.	—	1515	4271
Indianapolis, Ind.	1495	1046	2859
Ithaca, N. Y.	1448	1272	3803
Kansas City, Mo.	1616	1013	2738
Keokuk, Iowa	—	1107	3012
Knoxville, Tenn.	1189	756	2019
La Crosse, Wis.	1675	1353	4043
Lander, Wyo.	—	1372	4457
Lansing, Mich.	1453	1246	3834
Lincoln, Neb.	1736	1182	3296
Little Rock, Ark.	1115	642	1649
Los Angeles, Calif.	166	226	505
Louisville, Ky.	1382	896	2353
Lynchburg, Va.	—	848	2273
Madison, Wis.	1675	1332	3951
Marquette, Mich.	—	1367	4280
Memphis, Tenn.	1176	673	1672
Milwaukee, Wis.	1530	1225	3627
Minneapolis, Minn.	1816	1478	4391
Nantucket, Mass.	—	1062	3016
Nashville, Tenn.	1242	758	1985
New Haven, Conn.	1304	1184	3303
New Orleans, La.	683	290	639
New York, N. Y.	1239	1089	2936
Norfolk, Va.	1054	760	1857
Northfield, Vt.	—	1623	4857
North Platte, Neb.	—	1153	3416
Oklahoma City, Okla.	1245	733	2061
Omaha, Neb.	1761	1211	3402
Oswego, N. Y.	—	1313	3839
Parkersburg, W. Va.	—	978	2700
Peoria, Ill.	1602	1136	3201
Philadelphia, Pa.	1227	1030	2740
Pittsburgh, Pa.	1381	1045	2882
Pocatello, Idaho	—	1196	3841
Portland, Me.	1421	—	—
Portland, Ore.	708	648	2176
Providence, R. I.	1313	1195	3330
Pueblo, Colo.	—	997	3186
Raleigh, N. C.	—	748	1902
Reading, Pa.	1287	1088	3067
Reno, Nev.	941	840	2907
Richmond, Va.	1172	871	2277
Rochester, N. Y.	1444	1309	3838
Roseburg, Ore.	—	606	2073
St. Joseph, Mo.	—	1077	2941
St. Louis, Mo.	1479	920	2452
Salt Lake City, Utah	1021	1004	3062
Sandusky, Ohio	—	1125	3250
San Francisco, Calif.	383	340	1023
Sault Ste. Marie, Mich.	—	1567	4716
Scranton, Pa.	1387	1227	3616
Seattle, Wash.	608	593	2082
Sioux City, Iowa	—	—	—
Spokane, Wash.	1089	1023	3430
Springfield, Ill.	1565	1057	2854
Springfield, Mo.	—	877	1872
Syracuse, N. Y.	1483	1316	3897
Tacoma, Wash.	—	638	2303
Terre Haute, Ind.	—	1019	2782
Toledo, Ohio	1479	1168	3388
Trenton, N. J.	1285	1098	3031
Utica, N. Y.	1620	1421	4134
Washington, D. C.	1244	936	2489
Wichita, Kan.	1512	948	2647
Yakima, Wash.	—	988	3187

(—) Indicates data not available.

NEW EQUIPMENT



Thermotrol control for radiators.

Sterling Radiator Control Valves

NAME—Sterling Thermotrol, model No. 120.

PURPOSE—For the individual control of both steam or hot water radiators.

FEATURES—These controls have been re-designed so that they now measure 3 x 6 inches overall and are finished in a satin nickel with black plastic fittings. The control is fully self-contained and has a built-in volatile type thermostat that opens and closes the valve to hold the temperature at the desired setting. The construction is such that no outside source of energy is required and the adjusting knob can be set at any desired temperature up to 80F. It can also be locked at any desired temperature and is also free to close completely by hand.

MADE BY—*Sterling, Inc., 3732 North Holton St., Milwaukee, Wis.*

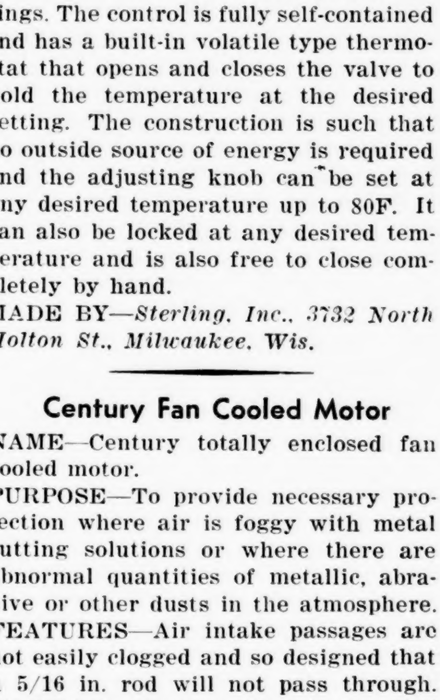
Century Fan Cooled Motor

NAME—Century totally enclosed fan cooled motor.

PURPOSE—To provide necessary protection where air is foggy with metal cutting solutions or where there are abnormal quantities of metallic, abrasive or other dusts in the atmosphere.

FEATURES—Air intake passages are not easily clogged and so designed that a 5/16 in. rod will not pass through.

MADE BY—*Century Electric Co., 1806 Pine St., St. Louis, Mo.*



Century fan cooled motor.

Mercoid Visaflame

NAME—Mercoid Visaflame.

PURPOSE—Mercury switch designed to operate in conjunction with the company's safety control panels as a power, ignition and flame failure shut-off for all types of oil burners.

OPERATION—Device is a gas-filled hermetically-sealed glass bulb containing a mercury pool at the bottom with one immersed electrode. The other electrode contains two opposing bimetal coils arranged to compensate each other for ambient temperatures. Inner or smaller coil located directly in front of a concave mirror so that when light rays from flame strike the mirror they focus upon this coil. Transmutation of the light energy into heat causes the bimetal coil to rotate and the mounted electrode is moved into the mercury to complete the circuit. The circuit remains closed in the presence of light from the flame, open in its absence.

FEATURES—Combustion chamber incandescence said to have no appreciable effect upon device, which is not to be confused with a photo-cell. Advantages are said to lie in independence of stack conditions for operation of device.

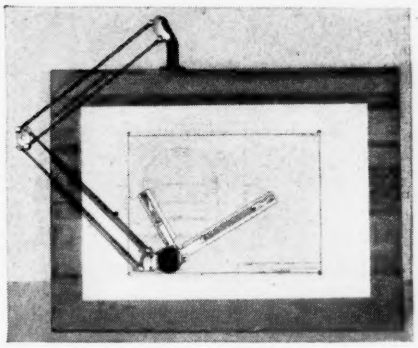
MADE BY—*The Mercoid Corporation, 4201 Belmont Ave., Chicago.*

Fan Housing

NAME—Venturi-type fan housing.

FEATURES—A pressed steel fan housing with Venturi section drawn down in one operation. Available as a standard item. Designed in a wide range of sizes and gauges for exhaust and ventilating fans and unit heaters. Available to manufacturers and also to contractors as a standard assembly unit.

MADE BY—*The Commercial Shearing & Stamping Co., Youngstown, Ohio.*



Low priced drafting machine.

Drafto Drafting Machine

NAME—Master-Drafto, Model No. 60.

PURPOSE—For drafting.

FEATURES—This is said to be a low priced drafting machine and is designed to take a maximum size sheet of 34 x 36 inches. The arms are constructed of seamless steel tubing fitted with solid bearings while the scale blades are designed so that any scale, either boxwood or aluminum, can be inserted. The protractor device is graduated in two degrees and is said to be set accurately for one-half degree readings by use of a graduated vernier. While the protractor can be locked at any degree, it is fitted with a latching spring to lock the scales at 0, 30, 45, 60 and 90 degrees on either side.

MADE BY—*The Drafto Company, 159 Walnut St., Cochranton, Pa.*

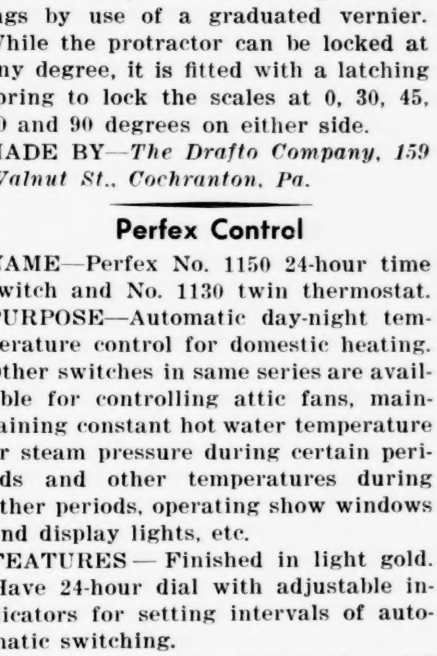
Perfex Control

NAME—Perfex No. 1150 24-hour time switch and No. 1130 twin thermostat.

PURPOSE—Automatic day-night temperature control for domestic heating. Other switches in same series are available for controlling attic fans, maintaining constant hot water temperature or steam pressure during certain periods and other temperatures during other periods, operating show windows and display lights, etc.

FEATURES — Finished in light gold. Have 24-hour dial with adjustable indicators for setting intervals of automatic switching.

MADE BY — *Perfex Corporation, 415 West Oklahoma Place, Milwaukee, Wis.*



Perfex time switch and twin thermostat.

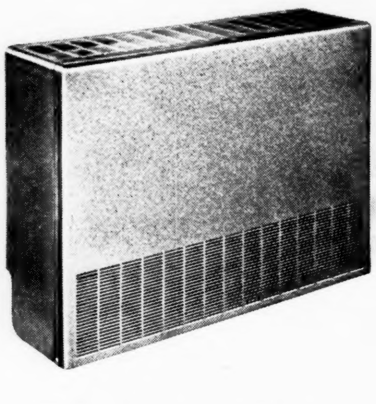
One of the line of Trane fans.

Trane Fans

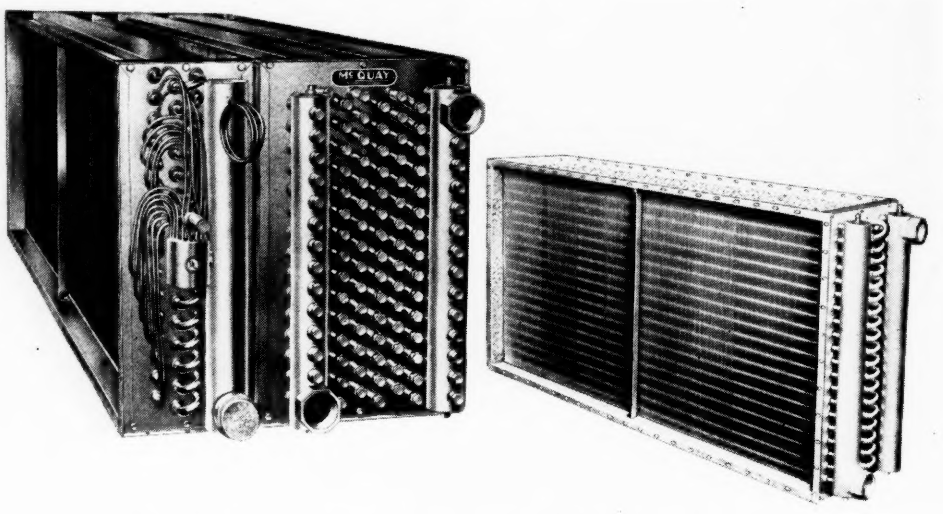
NAME—A line of Trane blower fans, now offered for sale separately.
FEATURES—Available in two types, FC of the forward curved multi-blade type and BI of the backward curved non-overloading type. Former developed to conserve space, lessen power requirements, eliminate noise, and deliver full capacity at lowest possible speed. Latter have twelve backwardly inclined blades. Included in the line are direct connected units for general utility duty, supply and exhaust. Both FC and BI fans available in this construction which consists of integral motor brackets fan housing, single width wheel mounted on shaft.
SIZES AND CAPACITIES—Type FC in single and double widths with wheel diameters from 1½ in. to 60 in. BI type with wheel diameters from 15 in. to 66 in.
LITERATURE AVAILABLE—Bulletin DS-348. Fifty-six-page bulletin giving complete engineering data, sample specifications, and a friction chart.
MADE BY—*The Trane Company, La-Crosse, Wis.*

Cabinet Unit Heater

NAME—Modine cabinet unit heater.
PURPOSE—For heating offices, lobbies, stores, and similar spaces.
FEATURES—The unit combines blowers, motor and heating coil and can be used with either steam or hot water systems. All piping and electrical connections are made within a cabinet enclosure. Cold air is drawn into the unit through the inlet grille at the base, is heated and then distributed into the room through the top grille. Five types of heaters are available; floor, wall, inverted wall, ceiling, and recessed. Wall cabinet is designed for installations where recessing is not practical and where easy cleaning beneath the floor is demanded. Inverted wall cabinet is designed for use in installations where it must be out of the way. Ceiling cabinet is used where requirements demand the installation of a unit overhead or out of the way, as in basements.
CAPACITIES—Three capacities ranging from 105 to 450 sq. ft.
LITERATURE—Catalog No. 841.
MADE BY—*Modine Manufacturing Co., Racine, Wis.*



Modine floor type unit heater.



(Left) McQuay combination water cooling and direct expansion coil; (right) water cooling and heating coil.

Vapor Barrier Blankets

NAME—Rock wool sealed blankets.
PURPOSE—Heat insulation with vapor barrier on one side to resist vapor and condensation.
FEATURES—The blankets have strong tacking flanges which are said to be tailored to exact fit to insure a job that will stay in place and give maximum temperature control. Blankets are applied with the vapor barrier to the room side and the crepe surface to the sheathing. Advantages claimed include permanence of application, vapor sealing, fewer joints, more area covered per carton, ease of application, and small warehouse space.
SIZES AND CAPACITIES—Available in thick, medium, and thin blankets of 3, 2 and 1 in. thickness respectively and with areas of 15 in. x 41 1/3 ft.; 15 in. x 62 ft.; and 15 in. x 103½ ft. respectively.
MADE BY—*The Philip Carey Company, Lockland, Cincinnati, Ohio.*

McQuay Air Conditioning Coils

NAME—Water coils, removable plug type water coils, and combination water and direct expansion coils.
PURPOSE—For water cooling and heating or water cooling and refrigeration in central fan systems and industrial applications.
FEATURES—Line of water coils recommended for cooling when sufficient cold water is available. In some installations it is possible to utilize coils for both heating and cooling using steam or water for heating. If water conditions require tube cleaning the RP type with non-ferrous removable plugs at both ends of all tubes are recommended. Combination coils designed for use with refrigerants; these are equipped with McQuay distributing headers or thermostatic expansion valves with integral distributing arrangements. With this type coil for cooling separate heating coils are necessary. Features of this coil include a large area of contact due to the wide fin collar design; permanent contact pressure secured by the hydraulic expansion method of expanding the tubes into the fins; and the quality of contact between collar and tube is due to polishing the inside surface of the fin collar during the spinning. Coils are made of copper, aluminum, brass, admiralty metal, etc., to meet requirements.
SIZES AND CAPACITIES—Over 10,000 standard size combinations available from 1½ in. x 1½ in. to 60 in. x 120 in. and may be had in any number of rows deep.
LITERATURE AVAILABLE—Catalog No. 289.
MADE BY—*McQuay, Inc., 1619 Broadway, N.E., Minneapolis, Minn.*

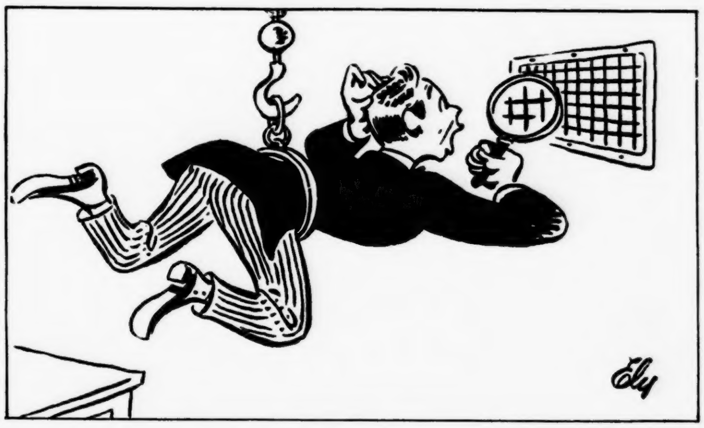
Air Filtering Facts—for forward-looking engineers

1. You're going to have the Treasurer eating out of your hand! Just wait 'till he gets a squint at the *low* low cost of Dust-Stop Air Filters in the air-conditioning system! They're about 1¢ per CFM to install complete, and only 1/10¢ per CFM to replace!

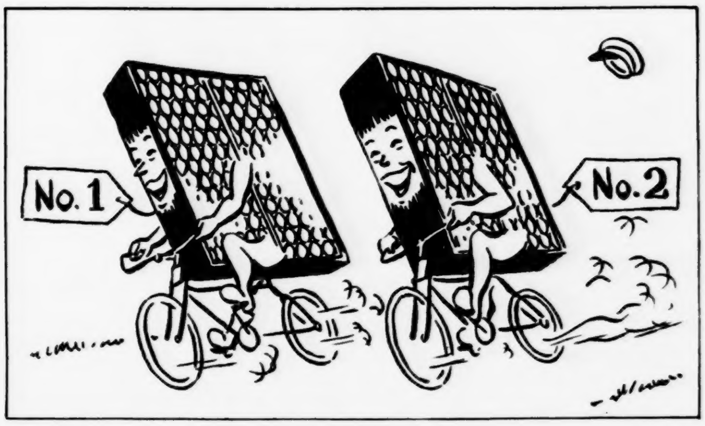
2. You don't hire a mechanic to open a pack of cigarettes . . . you don't have to hire one to change Dust-Stops! No skilled work required. What's more, they eliminate cleaning, charging, draining, etc. Nor do you have to keep any spares on hand.

3. People will say you're smart as a whip, when you specify Dust-Stops. They're ***efficient***, stopping virtually all "nuisance" dusts. Capacity: 2 CFM per square inch of area at 300 FPM. Average resistance, new (in inches of water gauge): .065 for 1-inch; .125 for 2-inch Dust-Stops.

4. And the only kind of fire you might have to worry about is the kind above. Dust-Stop's glass-fiber filtering element and adhesive ***will not support combustion!*** The special patented adhesive, incidentally, is available only on Dust-Stop filters.



5. You can look and look and you won't find any adhesive droplets on your walls or inside your ducts. Dust-Stops just won't bleed! So—of course there's no problem there, either. But don't just read this advertisement and do nothing about it . . .



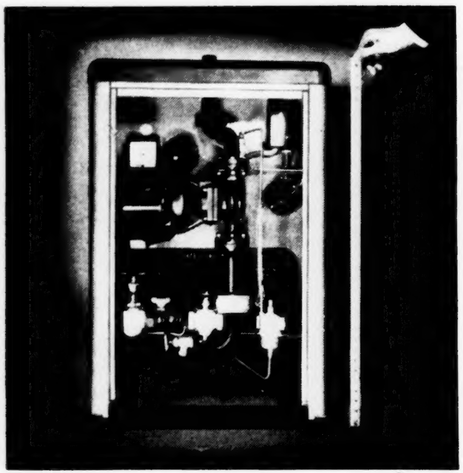
6. Order replaceable Dust-Stop Air Filters in a hurry! They come in two sizes, No. 1 (1-inch) and No. 2 (2-inch). Your air-conditioning manufacturer, jobber, or dealer has them. Or write: Owens-Corning Fiberglas Corporation, Toledo, Ohio.

FIBERGLAS* DUSTOP* AIR FILTERS

*T.M. Reg. U. S. Pat. Off.

Made by Owens-Corning Fiberglas Corporation, Toledo, Ohio

DIRECT CANADIAN INQUIRIES TO FIBERGLAS CANADA, LIMITED, OSHAWA, ONTARIO



Gas fired packaged heating unit.

Crane Basmore Package Unit

NAME—Crane Basmor 2WG de Luxe gas-fired package heating unit.

FEATURES—Package unit includes cast iron sectional gas-fired boiler, circulator, relay, room thermostat and pressure relief valve for hot water heating systems. With the exception of the thermostat all parts are connected and mounted in position inside the green baked enamel cabinet completely assembled.

SIZES AND CAPACITIES — Three sizes ranging from 415 to 739 sq ft. of 150 B.t.u. cast iron radiator surface.

MADE BY—*Bastian-Morley Co., Inc., LaPorte, Ind.*

Air Conditioner

NAME—Wizardaire Conditioner.

PURPOSE—A window unit for summer air conditioning and year-round ventilation.

FEATURES—Unit incorporates a filter, hermetically sealed refrigeration system, and fans all encased in an attractive cabinet. The cabinet cover is designed so that it can be easily removed. Filter is of the throw-a-way type and can be easily changed.

SIZES AND CAPACITIES—Three sizes with capacities ranging from 4100 B.t.u. to 5350 B.t.u. per hour.

MADE BY—*Certified Products Company, 2014 North 14th St., Toledo, Ohio.*

Wizardaire conditioner.

Sterling No. 150 Thermotrols

NAME—Sterling 150-A and 150-AH Thermotrols.

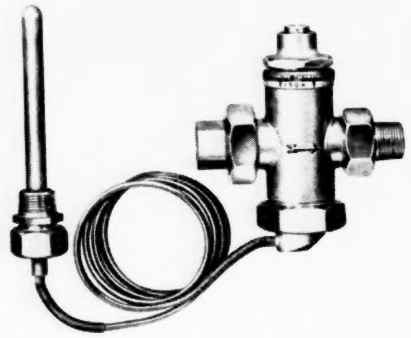
PURPOSE—Direct-acting thermostatic control for tank and process applications to regulate flow of steam or water.

FEATURES—A sensitivity of 1½F plus or minus on most applications is claimed. Economically priced.

SIZES AND CAPACITIES—No. 150-A, for 0 to 30 lb. pressures; No. 150-AH for 0 to 125 lb. pressures, both in ½, ¾ and 1 in. sizes.

LITERATURE—Bulletin 402.

MADE BY—*Sterling, Inc., 3732 N. Holton St., Milwaukee, Wis.*



Thermotrol for water storage tanks and similar applications.

Emco Gas Regulator

NAME — Emco Appliance Regulator No. 094-02.

PURPOSE—To provide accurate pressure control on gas appliances having a low gas consumption.

FEATURES—Modern appearance and of compact size. Device can be held in the palm of hand.

LITERATURE—Bulletin 1067.

MADE BY—*Pittsburgh Equitable Meter Company, 400 N. Lexington Ave., Pittsburgh, Pa.*

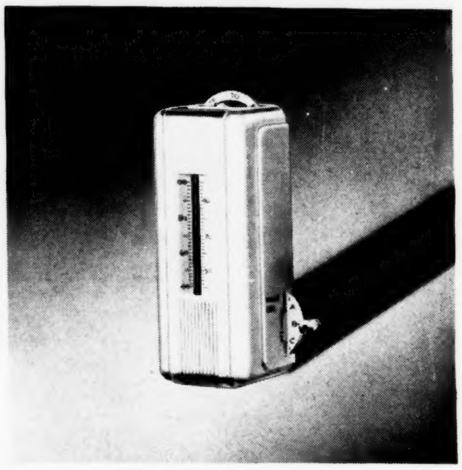
Ric-wiL Insulated Pipe Units

NAME—Ric-wiL insulated pipe units.

PURPOSE—Method of connecting the company's underground steam conduits.

FEATURES—Armco iron conduit, used in the company's lines, is rotated into position to match the helical corrugation of adjacent sections. For the lap joint ⅛ to ¼ in. lap of metal is obtained by springing the edge on one of the conduits. After welding the joint is protected against deterioration by ¼ in. thick asphalt blanket applied with a torch, making a bond with the metal. Asphalt is wrapped tightly with asbestos felt jacket to protect asphalt coating. A joint with thin flanges welded to the conduit and welded at the outer edge can also be supplied.

MADE BY—*The Ric-wiL Company, Union Commerce Bldg., Cleveland, O.*



Perfex Magic room thermostat.

Perfex No. 155 Thermostat

NAME — Perfex No. 155 Magic Dial Room Thermostat.

PURPOSE—Room thermostat available for distribution through heating unit manufacturers.

FEATURES—A heater type thermostat with sensitive and powerful bimetal element. Heater element is attached directly to the bimetal element so that the coil supplies heat to the bimetal by conduction, said to result in quicker response and regular recycling. Recycling characteristics can be adjusted by an external lever so as to "tune" the thermostat to the heating system, heating unit or type of fuel for most economical and satisfactory operation, said to result in close control of room temperatures. Styled and finished in light gold.

MADE BY—*Perfex Corporation, Milwaukee, Wisc.*

Barber-Colman Impressor

NAME—Barber-Colman Impressor.

PURPOSE—A hardness tester.

FEATURES — Intended for aluminum alloys or similar soft metals, plastics, hard rubber and the like.

MADE BY — *Barber-Colman Company, Rockford, Ill.*

Impressor, a hardness tester.

PERSONALS & PERSONNEL

John H. Ashbaugh, Westinghouse Electric & Manufacturing Company, formerly engineering manager at East Springfield Works, has been appointed assistant manager of manufacturing and engineering for the merchandising division. He will maintain headquarters in East Springfield but will be responsible for work at both Mansfield and East Springfield.

John G. Chaffe has been appointed district manager of the Philadelphia office of *Tube-Turns, Inc.*, Louisville, Ky., manufacturers of welding fittings. Mr. Chaffe's office will cover a surrounding area that extends through Wilmington, Baltimore, and Washington, D. C. The Tube-Turns Philadelphia office was simultaneously moved from the Lafayette Building to the Broad Street Station Building.

R. C. Cross was recently appointed chief engineer of the heating division of *Schwitzer-Cummins Company*, Indianapolis. Mr. Cross was for the past five years with Battelle Memorial Institute, engaged in the study and testing of bituminous coals, combustion, and stoker investigations. Prior to that he was with the *United States Bureau of Mines* as assistant fuel engineer and the *H. B. Smith Company*, in charge of boiler testing and also in charge of stokers and boilers.

Elliott Harrington

General Electric Co., Air Conditioning and Commercial Refrigertion Dept., Bloomfield, N. J., has made a series of organization changes. All sales activities are now consolidated under the supervision of *Elliott Harrington*, with *E. B. McClelland* as assistant sales manager. Responsibility for product sales has been segregated into three sections, two of which are new in the department organization. The automatic heating sales section under *L. H. Hobson*, formerly a zone manager, will supervise all oil heating and gas heating equipment sales. The cooling equipment sales section, under *C. M. Rowland*, also formerly a zone manager, will supervise sale of water coolers, room coolers, store coolers, refrigerators, air circulators, and ventilators. The third, the contractor and industrial sales section, responsible for the sale of all condensing units, evaporators, evaporative condensers, and central plant equipment, remains in charge of *S. Martin, Jr. Frank H. Faust*, secretary of the department's management committee has been appointed commercial engineer. *Harold T. Hulett* and *A. E. Pierce* have been transferred to other branches of the company.

Manning, Maxwell & Moore, Inc., Bridgeport, Conn., has extensively expanded its factory to keep pace with increased demands for its products. A sixth floor has been added to the company's Bridgeport plant and about 50,000 sq. ft. of space leased across the street. *The Hancock Valve Division* of this firm, located in Boston, has secured an additional four-story factory that will be devoted exclusively to the manufacture of Hancock steel valves.

J. A. Miller has been appointed advertising and sales promotion manager of the *Standard Oil Company of New Jersey* and affiliated *Esso Marketers*. Mr. Miller, who is 37, has been assistant manager since 1937. He succeeds James A. Donan who died recently.

J. E. Myers has joined the Metropolitan New York sales force of *Spencer Heater Div., Aviation Mfg. Corp.* Mr. Myers will operate out of Spencer's New York Branch, 101 Park Avenue.

Plandaire, Inc., Pittsburgh, has appointed *Roscoe C. Rider, Jr.*, as advertising manager. Mr. Rider worked for Standard Sanitary Manufacturing Company for a number of years and resigned from the Detroit sales promotion office in 1936. He later attended the Harvard School of Business Administration. Until recently he was employed by the Acheson Manufacturing Company of Rankin, Pa.

Plandaire, Inc., Pittsburgh, has appointed W. A. Witheridge Co., 746 South Fourth Ave., Saginaw, Mich., as sales representative in Northeastern Michigan; John Lloyd & Sons, 949 Hamilton St., Allentown, Pa., and Bennett Bldg., Wilkes-Barre, Pa., as representatives for Northeastern Pennsylvania; Hero D. Bratt, 228 Ottawa Ave., N.W., Grand Rapids, Mich., as representative for the Western Michigan territory; Harry E. Clark, 3209 Oakdale Ave., P. O. Box 370, Houston, Tex., as eastern Texas representative; Karl J. Hart, 903 No. State St., Jackson, Miss., as representative for the State of Mississippi; A. L. Haskins, 1624 Monroe Ave., Rochester, N. Y., as representative for the territory directly in the vicinity of Rochester; Electrol of Iowa, 533 Seventh St., Des Moines, Iowa, is the newly-appointed agent for Iowa; Percy L. Luck, 1936 General Taylor St., New Orleans, La., as sales representative in Southeastern Louisiana; and George Warren Stetson, 141 Milk St., Boston, Mass., as representative for Eastern Massachusetts.

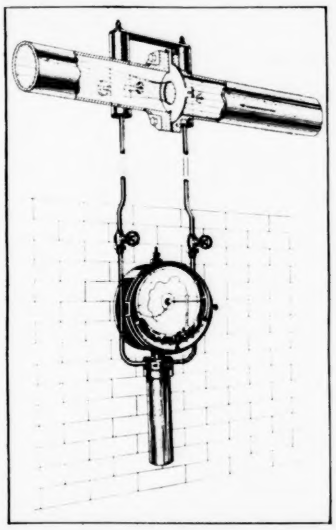
The Research Products Corporation, Madison, Wis., has appointed *H. W. Nussbaum*, 1837 W. Gramercy Place, San Antonio, Tex., as representative for that state.

Two new members have been added to the staff of the research and testing laboratory of *The Torrington Mfg. Co.*, Torrington, Conn., which is under the supervision of *Prof. W. L. Upson*. *E. Bryan Williams* has joined the laboratory staff as a research engineer. Mr. Williams had been, since 1928, chief traffic engineer with the St. Louis Public Service Company, devoting considerable of his time to research and design. He is a graduate of Washington University in St. Louis. *Philip M. Nigro*, until recently a professor of mathematics at Loyola College, Montreal, has also joined the staff as a research engineer. Mr. Nigro is a graduate of Holy Cross College, Worcester, Mass., in chemistry.

E. Bryan Williams (left) and Philip Nigro, research engineers, in the laboratory of The Torrington Manufacturing Co., conducting a special NAFM test.

The ADSCO FLOW METER, Orifice Type

ACCURATELY METERS STEAM, WATER, COMPRESSED AIR OR GAS.
SIMPLE IN CONSTRUCTION - - EASILY INSTALLED



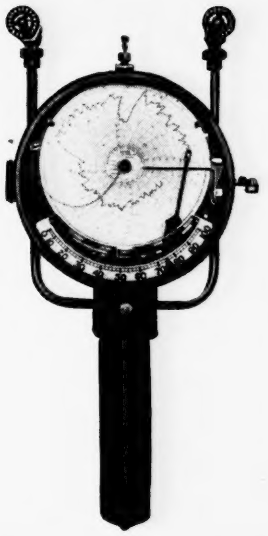
Free Floating, Frictionless Meter Mechanism transmits flow to direct-reading, evenly-divided chart

The ADSCO Flow Meter has a record of exceptional accuracy at all rates of flow. It is highly sensitive to fluctuations in flow but cannot be damaged by sudden overloads or reverse flows. Simple and rugged in construction—easily installed and maintained—requires no frequent inspection or adjustments.

Available with recording chart, indicating scale and integrator counter to totalize the flow or in other combinations of these three devices.

Correspondence is invited regarding your metering problems so that ADSCO may make recommendations based on over 25 years experience in the manufacture of meters.

Write for Bulletin No. 35-83V

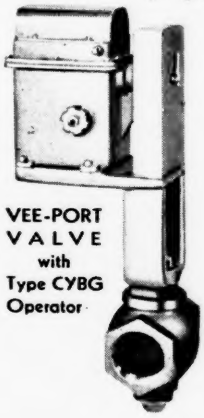


AMERICAN DISTRICT STEAM COMPANY
NORTH TONAWANDA, N.Y.
IN BUSINESS OVER SIXTY YEARS

WITH BARBER B-C COLMAN *ELECTRIC* CONTROLS

P·R·O·P·E·R E·N·G·I·N·E·E·R·I·N·G

SPELLS ▶ *DEPENDABLE PERFORMANCE*
▶ *LONG LIFE*
▶ *MINIMUM MAINTENANCE*



VEE-PORT VALVE with Type CYBG Operator

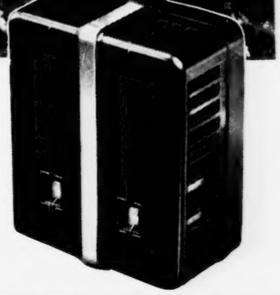
● An outstanding feature of Barber-Colman ELECTRIC Temperature and Humidity Control Systems is ENGINEERING SERVICE. The installation is carefully planned in every detail and is then backed with a positive *guarantee of satisfactory operation*. Quality materials, high manufacturing standards, and attention to accuracy both in construction and performance, assure that Barber-Colman ELECTRIC Controls will provide long and dependable service, with maintenance costs held to a minimum.

Write for Descriptive Bulletins

BARBER-COLMAN COMPANY PLANT AT ROCKFORD, ILLINOIS

Established 1902

The basic aim of Barber-Colman Company for over 38 years has been sound engineering, and the manufacture of dependable products, designed primarily for accurate performance and long life.



MICROTHERM
An outstandingly sensitive and responsive control instrument, originated and developed by Barber-Colman Company.

1224 ROCK ST. **BARBER-COLMAN COMPANY** ROCKFORD, ILL.

George W. Akers of the *George W. Akers Company*, Detroit, Mich., Lieutenant-Commander in the United States Naval Reserve, has been ordered to active duty in the Navy.

American Machine and Metals, Inc., DeBothezat Ventilating Equipment Div., East Moline, Ill., has appointed *Alonzo M. Harp* as assistant sales manager of the DeBothezat Division. Mr. Harp has served in the manufacturing, engineering and sales departments of the DeBothezat organization since 1929. For the past three years, he has been sales manager of the central division. He will be located at headquarters in East Moline.

Paul E. Warburg, for six years district wholesale manager of the air conditioning division, American Radiator and Standard Sanitary Corp., has joined the *Dorex Division, W. B. Connor Engineering Corp.*, manufacturer of odor control equipment.

Roy H. Warmee

Roy H. Warmee has been appointed sales promotion manager of *Minneapolis-Honeywell Regulator Company*, with headquarters in Minneapolis. Mr. Warmee was formerly sales manager of the Philadelphia Coke Company, Philadelphia. In 1937 he won the National Howard G. Ford award for outstanding achievement in sales management and at the present time he is national chairman of the 1940 award.

WITH THE MANUFACTURERS

Airtemp Div., Chrysler Corp., Dayton, Ohio, has prepared a comprehensive program of advertising and sales promotion for 1941. The plan was presented in detail to the 48 district managers of the division at a meeting in Dayton in February. Highlight of the convention was a series of skits written and acted by *Arthur Suit*, district manager at Washington, and *Jack Duer*, district manager at Cincinnati. In their talks to the field organization *Earl Marr*, general sales manager and *Ward Barnett*, merchandising manager, made it clear that the division had a big job laid out for 1941—first in completing national distribution and second in aiding Airtemp direct dealers to get their share of the increasing market for air conditioning equipment. *D. W. Russell*, president of Airtemp, pointed out that in six years several millions of dollars had been spent by the Chrysler Corporation to bring Airtemp to its present position.

Anemostat Corp. of America, New York, announces that its high velocity air diffusers will be made available to buyers of industrial and commercial unit heaters through arrangements that have been made with a number of manufacturers. The diffusers have been available with Trane projection heaters for the past year, but under the new sales plan they will be supplied with certain types of heaters manufactured by Young Radiator Corp.; Modine Mfg. Co.; McCord Radiator Co.; Niagara Blower Corp.; and the Webster-Nesbit Co. The company has appointed *Control Engineering Company*, headed by *A. F. Erickson*, as representative in Kansas City area. This company succeeds the Leffel Company as Anemostat representative in that area.

Autovent Fan and Blower Co., Chicago, is celebrating its 25th anniversary this year. The company was organized early in 1916.

Buffalo Forge Co., Buffalo, N. Y., will build the ventilating equipment for several new United States battleships. It has just received a $103,233 order from the Navy Department for such equipment, including fans, ducts and other accessories. Filling of the order will be spread over several years.

Carrier Corporation, Syracuse, has sold its Allentown, Pa., plant to American Armament Corporation.

Davies Air Filter Corp., New York City, formerly at 390 Fourth Avenue, have announced the removal of their office and plant to new and larger quarters at 118-120 East 25th Street, New York City.

The Detroit Air Conditioning Service, 1474 Holden at Trumbull, Detroit, Mich., is manufacturing a line of air conditioning instruments, including the Detroit grillometer, air flow meter, draft and filter gauges, Pitot tubes, and other air conditioning instruments.

The Elgo Shutter and Manufacturing Co., Detroit, Mich., has acquired and moved into a new plant at 6970 W. Jefferson Ave., giving the company four times as much floor space as the plant previously occupied. The company manufactures ventilating specialties, including shutters and dampers.

Modine Mfg. Company, Racine, Wis., on February 4 was host to 78 plumbers and steam fitters from Northern Wisconsin, Michigan, Minnesota, North Dakota and Montana. The group, under the sponsorship of the *Duluth Plumbing Supplies Co.*, Duluth, arrived in Racine aboard four special pullman cars. The visitors were welcomed to the city by *A. G. Dixon*, sales manager and *Owen E. Desmond*, assistant sales manager. After an inspection of the plant and its manufacturing processes, the plumbers and steam fitters saw displays of Modine products and participated in a discussion of the unit heaters led by Mr. Dixon.

Sarco Company, Inc., 183 Madison Ave., N. Y., has moved its headquarters to 475 Fifth Ave., N. Y., where the company will occupy the entire 24th floor.

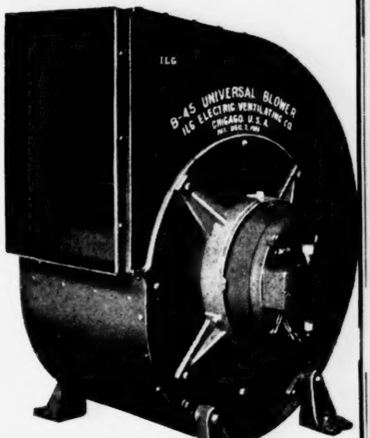
Sterling, Inc., Milwaukee, has appointed *Allonier Engineering Company* as sales representative in the Cincinnati, Ohio, territory. Prior to this appointment, Oscar Allonier represented the Dunham Co. for a great many years.

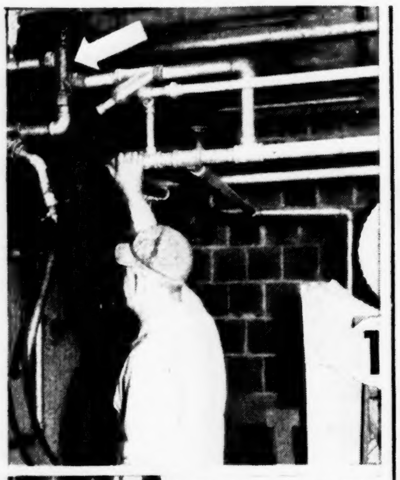
The Torrington Mfg. Co., Torrington, Conn., announces its purchase of patents covering design and methods of producing pressed steel blower wheels from *American Blower Corporation*. Torrington has for several years been manufacturing these fans under license from the American Blower.

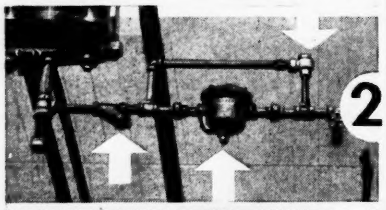
Universal Cooler Corporation has moved its factory and offices from Detroit to Marion, Ohio, where it is now occupying the extensively remodelled former plant of the Susquehanna Silk Mills of that city.

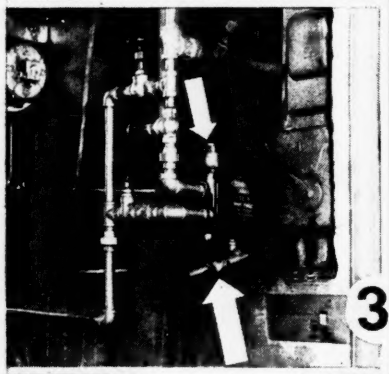
Water Cooling Corporation, 71 Nassau St., New York, has been formed by *A. B. Tappen*, former president of the Cooling Tower Company. The company specializes in complete installations of water conservation equipment including mechanical draft cooling towers, atmospheric cooling towers, spray nozzle cooling systems and roof cooling systems.

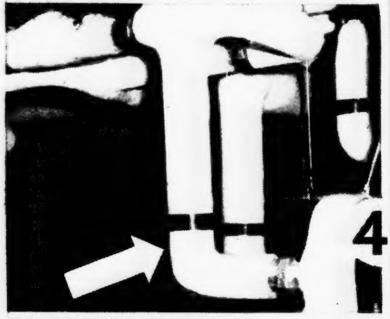
Worthington Pump and Machinery Corp., Harrison, N. J., has announced that *M. P. Robinson*, proprietor of the Robinson Filter Co., has recently taken charge of their newly organized water purification equipment division.

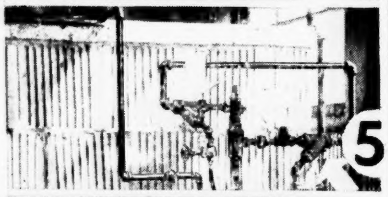












NEW TRADE LITERATURE

Air Distribution. A 4-page standard size colored folder entitled "Air Conditioning—Success or Failure?" and showing recent important installations of Anemostat air diffusers. Anemostat Corporation of America, 10 E. 39th St., New York.

Boilers. A 64-page catalog on National coal, stoker, oil and gas boilers—both steam and cast iron. Ratings and data on radiators, convectors, enclosures, and hot water heaters are also included. The National Radiator Co., 221 Central Ave., Johnstown, Pa.

Building Equipment. An 8 x 10⅝ in., 28-page bulletin No. WP-1099-B27, illustrating and describing Carbondale equipment for buildings and institutions, and including air conditioning units, refrigeration compressors, compressor-condensing units, condensers, coolers, deaerators, feedwater heaters, jacket water coolers, pumps, refrigerating units, room coolers, turbines, etc. Each product is illustrated and accompanied by, in most cases, size and capacity data and a brief summary of features, construction, capacities and advantages. Carbondale Division, Worthington Pump and Machinery Corporation, Harrison, N. J.

Coils. Standard size 40-page bulletin in color containing descriptions of the company's line of water coils, direct expansion coils, and combination coils, with complete descriptive and illustrative material relating to them. A feature of the catalog is the complete set of data and graphs relating to the various coils, accompanied by complete instructions and examples for using these data in selecting coils for air conditioning. A copy of the Bulkeley chart is included with each catalog. McQuay, Inc., 1619 Broadway, N. E., Minneapolis.

Condensate Pumps. An 8 x 10½ in., 4-page bulletin No. W-325-B1, illustrating Worthington single-stage condensate pumps, type JZ. Worthington Pump and Machinery Corp., Harrison, N. J.

Dehumidifiers. A 24-page catalog, entitled "Lectrodryer—Dry Air and Gases," in color, of the company's line of equipment for dehydrating air and other gases. Includes a psychrometric chart in color with absolute humidity plotted against dewpoint temperature. Pittsburgh Lectrodryer Corporation, P. O. Box 1766, Pittsburgh, Pa.

Fans. A dealer sales manual of the complete line of Wagner fans available in 1941, entitled bulletin FU-41. Standard size, 20 pages. Wagner Electric Corporation, 6400 Plymouth Ave., St. Louis, Mo.

Odor Control. A 4-page standard size bulletin, No. 111, entitled "What Dorex Odor Adsorbers Mean to Air Conditioning." Cost comparisons are presented showing the cost of introducing outside air into an air conditioning system for both winter and summer as compared with the cost of recirculating the air and using Dorex odor adsorbers. The example presented shows somewhat over 100% annual return on the Dorex installations. Dorex Division, W. B. Connor Engineering Corp., 114 East 32nd St., New York.

Shaft Seals. A 20-page standard size illustrated bulletin No. 825 covering Sylphon shaft seals. Illustrates many types of shaft seals whose purpose is to prevent leakage of gas or liquid around a rotating shaft where the shaft extends into a vessel or enclosure. Bulletin includes specifications, dimensions, operating characteristics and instructions. Fulton Sylphon Company, Knoxville, Tenn.

Spray Painting. A standard size 44-page catalog No. DH covering the 1941 line of DeVilbiss spray painting equipment, including exhaust chambers and spray booths and other ventilating equipment for use in connection with spray painting. THE DEVILBISS COMPANY, Toledo, Ohio.

Stainless Steel. A comprehensive report covering all phases of fabrication of Allegheny stainless steels. Standard size, 28 pages. ALLEGHENY LUDLUM STEEL CORPORATION, Pittsburgh, Pa.

Refrigeration Compressors. A 6-page standard size bulletin C-1100-B11, on the Worthington-Carbondale two-cylinder type refrigeration compressors. Gives information on the design, construction, and dimensions of sizes 5 in. x 5 in., and smaller. CARBONDALE DIVISION, WORTHINGTON PUMP AND MACHINERY CORPORATION, Harrison, N. J.

Unit Heaters. A standard size, 12-page catalog U41 entitled, "Reznor Gas Fired Suspended Unit Heaters." Includes descriptions of fan type, blower type and duct type unit heaters and is fully illustrated with drawings and photographs which show the important construction features of the various types of units. Unit capacities range from 55,000 B.t.u. to 200,000 B.t.u. per hour input. REZNOR MANUFACTURING COMPANY, 219 James St., Mercer, Pa.

Water Conservation. A standard size 8-page bulletin in color describing the atmospheric cooling towers, mechanical draft cooling towers, spray nozzle cooling systems and roof cooling systems of the company together with proposed specifications for these equipments. WATER COOLING CORPORATION, 71 Nassau St., New York.

Water Conditioning. A 48-page catalog, Publ. No. 3000, on hot process water softeners for removal of hardness, silica, and other scale-forming material from boiler feed and industrial process waters. Contains a nine-page appendix of feed-water chemistry, in addition to heat balances, flow diagrams, two-color construction drawings, and engineering tables and charts. COCHRANE CORPORATION, Philadelphia, Pa.

COMING EVENTS

MARCH 17-22, 1941. National Oil Burner Progress Exhibition of oil heating and air conditioning equipment. Commercial Museum, Philadelphia. Convention headquarters, Benjamin Franklin Hotel. Further information available through C. F. Curtin, Oil Burner Institute, 30 Rockefeller Plaza, New York.

JUNE 10-13, 1941. The 32nd Annual Meeting of the National District Heating Association. To be held at the William Penn Hotel, Pittsburgh, Pa.

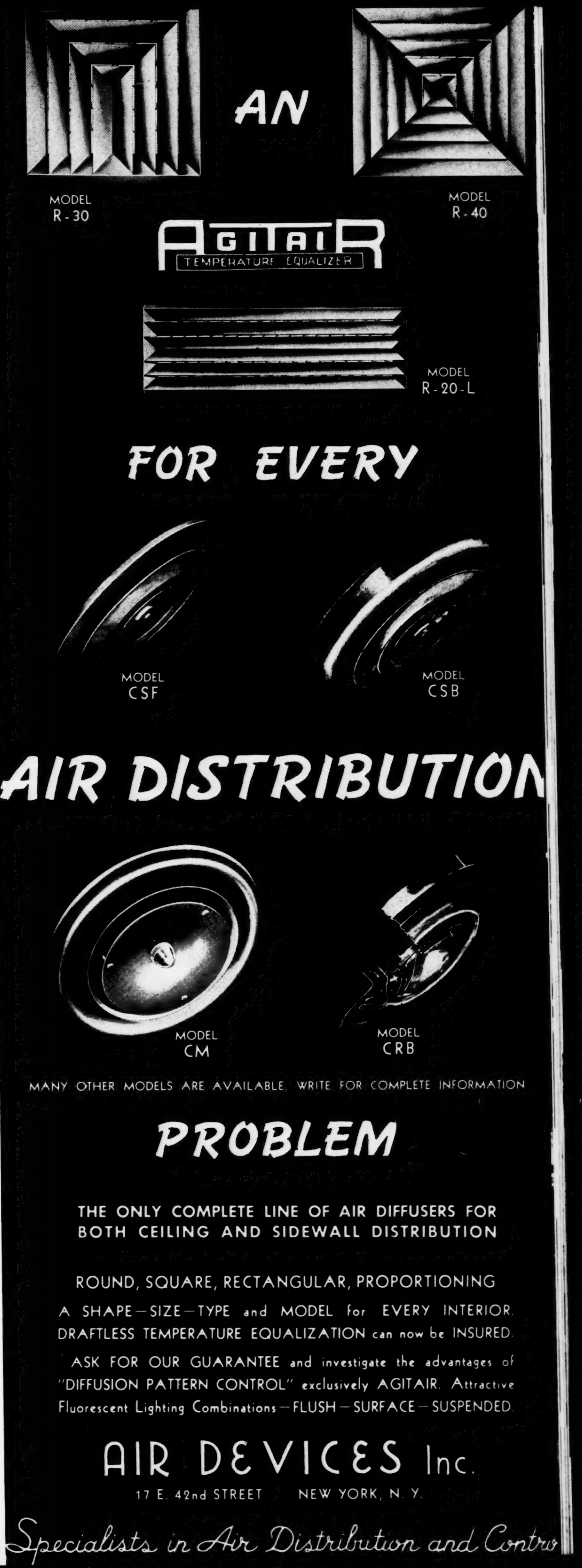
JUNE 16-20, 1941. The Pacific Heating and Air Conditioning Exposition, Exposition Auditorium, San Francisco, Cal. Managed by the International Exposition Company, Grand Central Palace, New York.

JUNE 16-20, 1941. Meeting of the Heating, Piping and Air Conditioning National Association. To be held at St. Francis Hotel, San Francisco, Cal.

JUNE 16-20, 1941. Summer Meeting of the American Society of Heating and Ventilating Engineers. To be held at Palace Hotel, San Francisco, Cal.

JUNE 23-27, 1941. Annual meeting and exhibit of the American Society for Testing Materials. Palmer House, Chicago, Ill. For information, write Society at 260 So. Broad St., Philadelphia, Pa.

OCTOBER 14-17. 70th annual meeting of the American Public Health Association to be held at Convention Hall, Atlantic City, N. J. Headquarters, Hotel Traymore. For information, write to American Public Health Association, 1790 Broadway, New York, N. Y.



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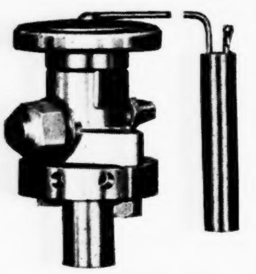

Range of Sizes:

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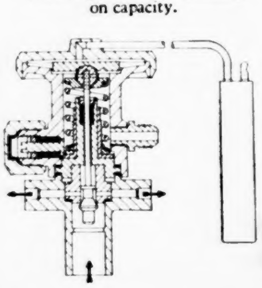
Two to 36 outlets, depending on capacity.



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Due to the patented design and construction, Alco Multi-Outlet Thermo Valves provide better distribution of refrigerant than is possible with any of the ordinary distributors or tube clusters now in use.

Distribution takes place within the valve body, at the point of expansion, and before separation of gas and liquid occurs. With Alco Multi-Outlet Thermo Valves, the flow of liquid refrigerant to each of the evaporator circuits is equal and direct and distribution is not affected by load changes.

The purchasers of Alco Multi-Outlet Thermo Valves are assured of full protection against the infringement of any existing application patents.

The controlled distribution in the Alco Multi-Outlet Thermo Valve eliminates distribution problems and three years of actual field experience has shown many cases of increased coil efficiencies up to 20 per cent.

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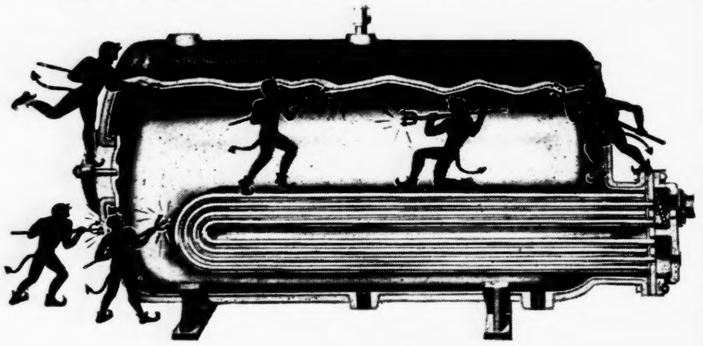
Demon Rust Can't Attack

Patterson Everdur or Copper-Lined Hot Water Heaters



These heaters can't possibly rust because water never touches anything but copper. Without rust, there is no red water and practically no wear.

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Installation is extremely simple and inexpensive.
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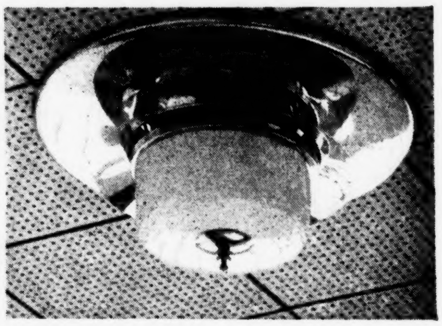

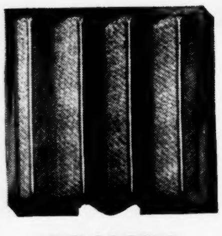

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Where Somers Hair Spun Glass Filters are installed as original or replacement equipment there's an end to the cost. Somers Filters are washable, odorless, non-absorptive, do not rot, and are practically indestructible.

Dust, dirt, pollen and detritus in the air stream effectively removed with minimum back pressure. No adhesives required.

Somers Filters may be cleaned easily by using the stream from an ordinary water hose (at city pressures) pinched to obtain a spray effect.

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HEATING AND VENTILATING

The Air Conditioning Monthly