Chapter 13 Part 3
HEATING AND VENTILATING DESIGN

Within 30 years, Box's famous *Treatise on Heat* shows that a more reliable and quantitative method of pipe-sizing had become available. Box gives tables of pipe friction based on Prony's formula.[12] The method was, however, one of trial and error, in which the pipe size was adjusted so that the friction loss was equal to the available head when the water quantity was that necessary to make good the heat requirement of the building.

The situation remained unsatisfactory, however, and in 1889 Baldwin wrote:[4]

> "For small tubes under very small heads of water, and such as can be found in heating apparatus, there are no authentic data (on pipe friction). I depend on the tables of G. A. Ellis of Boston."

These tables gave typical data:

Flow		Resistance (in wg/10 ft length)		
US/gal/min	Imp. gal/min	1 in dia	2 in dia	3 in dia
2	1.6	0.4	0.012	-
10	8	8.53	0.28	0.04

Baldwin used Box's data for the equivalent length of bends.

Pipe sizes were often directly related to the area of heat emitting surface. Thus Babcock, in 1885 proposed the formula:

$$\text{pipe diameter (in)} = 0.1 \text{ radiating surface (ft}^2\text{)}$$

for steam pipes; while Wolff (Germany) gave tables connecting pipe diameter with area and heat emission. Hoffman and Raber used similar tables in 1913.

Jones, in 1904, stated that the sizes of the main pipes were based on experience, and related to the heating surface of the radiators. As an empirical rule, the area of 1-in i.d. pipe was taken to be sufficient for 100 ft^2 of radiation, though relatively more was allowed in the smaller sizes of pipe. Jones[33] recommended that somewhat larger mains were to be allowed for radiators than for the equivalent length of 4-in pipe, on account of the greater frictional loss in the former.

Picard (1897) used Levy's formula:

$$D = \frac{1}{3}\sqrt[5]{\frac{Q^2}{J}}$$

where D = pipe diameter
Q = volume of water circulating
J = pressure drop per metre

to size the pipes, and adopted Rietschel's method based on an index run.

Carpenter, in America, outlines the steps in sizing hot water pipes (1910):

(a) calculate the water quantity, from the heat output and temperature drop;
(b) calculate the theoretical velocity, without friction, from the available head;
(c) hence determine pipe diameter;
(d) allow for friction by using a pipe one size larger.

Rietschel's text of 1895 gives extensive tables of fluid flow in pipes. It seems that they were used in Germany and the German-speaking countries of Europe, but they do not appear to have been employed in either Britain or America. In a later edition, he gives the resistance of a straight rectangular duct of length l as:

$$R = f \cdot 2l \frac{(a + b)}{a\,b}$$

f = friction coefficient
a and b = sides of rectangle

and the total resistance of a duct with fittings as

$$Z = R + \Sigma\rho$$

ρ being the "local" resistance of a fitting, expressed in velocity head.

The head loss in m of air is then

$$h = \frac{v^2}{2g} \cdot \frac{Z}{1 + \alpha t}$$

αt being a density correction for temperature. For local resistances, his own tests of 1905 yielded

square elbow	(brickwork)	ρ = 1.5
round elbow	(brickwork)	1.0
grille	(brickwork)	0.3 to 2, depending on face area
square elbow	(metal)	1.1
round elbow	(metal)	0.25–0.07 depending on radius
easy bend	($r > 5d$)	0
branch		1.1
change of section		$f/f_1 - 1$ (f, f_1 areas)

Rietschel studied the resistance of radiators to water flow:

connections top, bottom, same end	10-section radiator	0.0143 in wg
	20-section	0.12
connections top, bottom, opposite ends	10-section	0.003
	20-section	0.026

The quantity of water circulated was such that the temperature difference across the radiator was 20°C in each case. Barker preferred to use a single figure, $\zeta = 2$, for any radiator.

In spite of Rietschel's work, Jones, writing in 1904, made no use of it; and Carpenter writing in 1910, quotes two formulae which relate the head and velocity of water flowing in pipes, and which were said to be generally accepted at the time. One of these, due to Eytelwein, reads:

$$v = 50 \sqrt{\frac{d\,h}{l + 50\,d}}$$

where v is the velocity, ft/s
d is the diameter, ft
l is the length, ft
h is the head, ft of water.

For very long pipes, where the length is much greater than the diameter, the formula becomes:

$$v = 50 \sqrt{dh/l}$$

or

$$h = v^2\, l\, /50\, d$$

Hawkesley's expression, also quoted by Carpenter, differs only in the use of the factor 1/48 in place of 1/50.

A. H. Barker[7] seems to have been more determined than most of his contemporaries to rest his practice as far as possible on fundamentals. Accordingly, his text of 1912 deals with fluid flow at some length, and he draws on Weisbach's and Rietschel's work. He appreciated that the friction data obtained by these engineers was deficient, in that they were specific to the pipes and ducts tested. He felt it convenient, in the theory, to express the various resistances in a fluid circuit in terms of the velocity head, but in actual computation he deemed it preferable to use instead the concept of the equivalent length of straight pipe, since in practice the velocity may vary from point to point. The velocity head concept arose since "by a scientific accident, rather than from any fundamental reason" the loss of energy in pipes and fittings was found to be proportional to v^2. At very low velocities, where the flow is laminar, Poiseuille's law holds, and Barker thought it to be appropriate to the velocities encountered in gravity hot water systems. He commented too that neither the Weisbach nor the Poiseuille formulae take the pipe roughness into account, and cannot be applied to any but smooth pipes. He was unfortunately not in a position to substitute anything better, and perforce had to use Weisbach's data on which Rietschel's tables were based. (The same comment would apply to the formulae recommended by Carpenter.) Yet Hoffman and Raber (1913) were able to present a friction chart (taken from Church's *Hydraulic Motors*) identical in form to modern charts (Fig. 13.7).

In 1926 Preston noted that although many formulae had been published which related to the flow of water generally, there was an apparent lack of research into the flow of hot water under the conditions usually obtained in accelerated circulation.[49] He believed the most appropriate formula was that of Harding and Willard (*cf.* Rietschel - NSB/BMR):

$$H = \frac{0.32\ V^{1.86}}{10\left(\frac{D}{12}\right)^{1.25}}$$

where H = loss in feet per 100 ft.
D = pipe diameter, in
V = velocity of water, ft/sec.

Preston also pointed out the peculiarities of the "one-pipe" heating system:

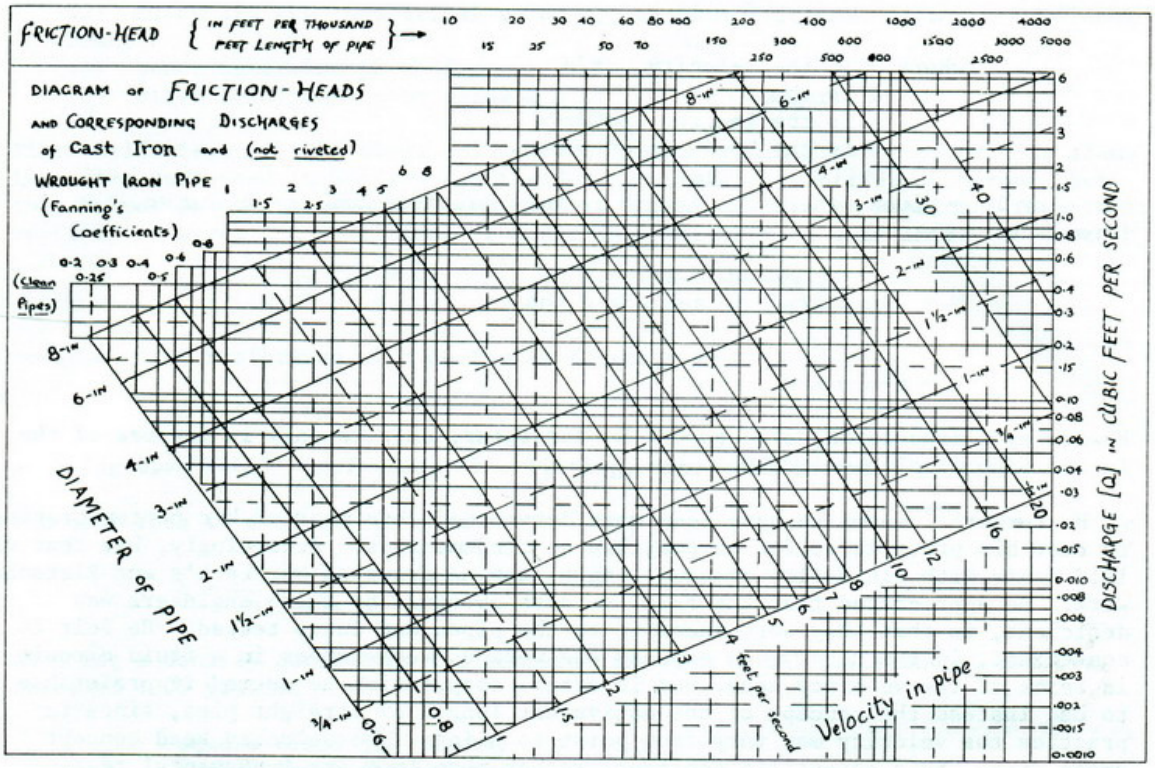

Fig. 13.7. Friction chart (1913).

> "... it must not be forgotten that the radiators depend chiefly on gravity, and the connections to same should be of ample size. For instance, it is not unusual to have, say, a ½ in main and ¾ in connections off same to the radiators."

In 1927, Pullen[50] remarked:

> "The friction head for pipes and fittings is a point on which no two authorities seem to agree exactly, no doubt due to the variations of size of the internal diameters of commercial piping with which their tests have been carried out.* Box's formula... has proved thoroughly satisfactory in practice."

From the practical point of view, it seems preferable to follow Box, Barker and Carpenter and express local resistances in terms of equivalent length of straight pipe. This practice has been adopted by CIBS** in Britain, and with slight modifications by ASHRAE*** in America.

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*Some problems of the time were apparently caused by foundry sand which had been left in radiators by the manufacturers. Barker had another concern. He believed that in different pipes of the same nominal bore from the same factory the friction varied by as much as 25%; he even suggested that every length of pipe ought to undergo a friction test before it was sent out!

**Chartered Institution of Building Services, England (formerly IHVE).

***American Society of Heating, Refrigeration and Air Conditioning Engineers.

A practical problem which has not yet been satisfactorily solved concerns the flow of fluids through a succession of closely-spaced fittings — though Rietschel did attempt to treat this by giving percentage increases to the resistance of closely spaced sharp elbows. All experimental values for the resistance of a fitting are determined with the entering flow either laminar or fully developed turbulence; in both cases, the velocity profile is axi-symmetric. Now, downstream of a fitting, the flow will not in general be fully developed, nor the profile symmetrical. It is easy to see therefore that the resistance offered by a fitting when the upstream flow is "non-standard" will be different from (and probably larger than) that obtained in laboratory tests on single fittings.

Dye (1901) gives a table for sizing the steam pipes in low-pressure systems. For a one-pipe system:

ft^2 radiation	Diameter of main (in)
200	1½
200-400	2
400-650	2½
1250-1600	4

(100 ft^2 radiation requires $\pi/4$ in^2 of pipe area) and for a two-pipe layout:

ft^2 radiation	Diameter (in) steam main	Diameter (in) condensate
100-140	1	¾
125-200	1¼	1
200-300	1½	1¼
2500-4000	6	4

(140 ft^2 radiation requires $\pi/4$ in^2 of steam main).

He gave empirical tables of radiating surface required for each 1000 ft^3 of room volume (shades of Tredgold!) and these were the same for both low- and high-pressure steam. For example:

Living rooms	63-65°F	10 ft^2/1000 ft^3
Schools, offices	60°	8-9 ft^2
Light factory work	52-55°	5-7 ft^2

For the boiler, there was (again according to Dye) "no rule by which the power of a boiler can be judged correctly". He suggests 1 ft^2 heating surface in the boiler for each 17 ft^2 of radiating surface in the room, or alternatively 1 ft^2 grate area for 200-300 ft^2 radiating surface. Since boiler ratings were then often expressed in horsepower, Dye suggests 1 boiler horsepower to 150 ft^2 radiating surface; 13½ ft^2 boiler heating surface (direct + indirect) gave a power of one horsepower.

Duct sizing calculations were, if anything, less rational than those for water pipes, although Box and Rietschel had given appropriate formulae. Dye, for instance, gave a series of empirical rules for gravity warm-air systems in residences:

Warm-air duct to ground floor rooms	50 in^2 per 1000 ft^3 room vol.
first floor rooms	40
second floor rooms	32
Air inlets to rooms	clear area not less than $\frac{1}{4}$ duct area
Fresh air intake	clear area = $\frac{3}{4}$ total area of warm air inlets
Grating of intake, if cotton-wool filter or wire gauze used	area = twice intake area

All horizontal runs of warm air duct to ground floor systems to rise at least 1 in/ft if less than 15 ft long; $1\frac{1}{4}$ in/ft if more than 15 ft

Ventilation shafts or outlet ducts to roof from ground floor rooms	32 in^2 per 1000 ft^3 room vol.
first floor rooms	40
second floor rooms	50

Monroe (1902) supposed that data on air friction were entirely empirical, and proposed to use half the theoretical velocity to allow for friction (a procedure identical with Tredgold's eighty years before).[42] This may have been due to ignorance of Weisbach's work, for Rietschel made use of it, as did Barker 10 years later. Yet in 1922, the ASHVE Guide was referring to empirical charts based on experiment at the University of Illinois, when discussing duct sizes for gravity warm-air systems. These results were expressed, rather like Dye's, as x mm^2 per kW. There is of course the probability that a deliberate attempt had been made to eliminate computation as much as possible, and to reduce design to a rule-of-thumb procedure.

Present-day guides give charts and/or tables based on the accurate formulae. The routine of duct-sizing was considerably simplified by the introduction of mechanical aids — notably the "Trane Ductulator" in 1950, and the "Carrier Duct calculator" of 1973; the latter device also enabled ductwork to be sized by the static regain method (in addition to the usual equal friction or less common velocity reduction methods).

13.12 SUMMER CONDITIONS

Whereas winter heat losses from a continuously warmed room may reasonably be regarded as steady, heat gains in summer are essentially periodic, being due to sunshine. Only the average value can be deduced from steady state equations. Early attempts to consider the periodic flow (e.g. by Mackay and Wright 1942-4) led to graphs for decrement factor and time lag which were too cumbersome for general use.[38][39] These were replaced (for both peak and hourly loads) by "equivalent temperature difference" — empirically determined values to allow for solar absorption, decrement and lag, of outside walls and roofs. The Carrier Manual went further than this, and applied storage factors for the properties of the whole room (possibly found from computer).

Danter was able to simplify decrement and lag data, by demonstrating the dependence on thickness and the near-independence of density. The subsequent development of admittance theory made the calculation of peak loads, from both internal and external sources, very simple.

Mackay and Wright's most notable contribution was the concept of "sol-air" temperature:

$$t_{sol} = t_a + \alpha I R_{so}$$

an idea which has been universally adopted for use for summer calculations (Fig. 13.8).

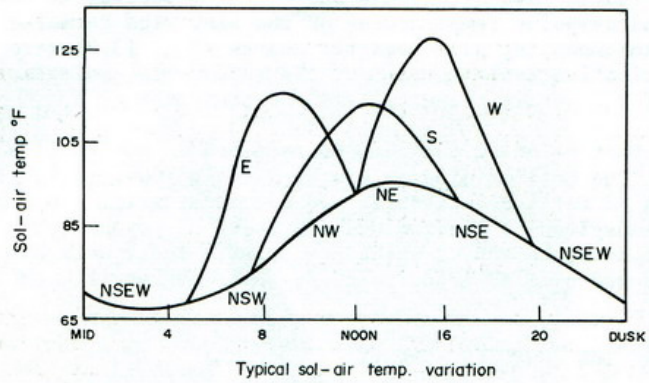


Fig. 13.8. Typical sol-air temperature variation.

Coblenz (US National Bureau of Standards) had shown the value of a white coating for the exclusion of solar heat striking a roof (i.e. a reduction of α in the sol-air temperature). A large part of the summer problem is solar gain through glazed areas. Ollett (1929) complained that he knew of no data on the transmission of sunshine through a skylight.[45] About this time, Beckett (BRS) was concerning himself with the transmission of light and heat through glass, and with methods of solar heat exclusion. There has been continuous refinement of transmission data for glass, and within recent years, the development of absorbing and reflecting glasses, to minimise the gains.

13.13 PSYCHROMETRIC DATA

The basic data for air conditioning calculations are found in tables of the physical properties of moist air — the psychrometric tables. Boyle (1659) established the relation between air temperature and density, and in 1800 John Dalton formulated the laws governing the vapour pressure of water in air. James Apjohn (an Irish Chemist) propounded the theory of adiabatic absorption of water by air in 1835-6, though he was unable to verify it. This was that the wet bulb temperature was related to the total heat (enthalpy) of moist air.

The first attempt to tabulate psychrometric data was made in 1847 by the English meteorologist James Glaisher, who computed reliable tables of the stationary wet- and dry-bulb temperatures. It is, no doubt, these tables which were used for the Cotton Cloth Factories Act of 1889, and which were praised by Wilson. They were followed by Ferrel's empirical formulae (1886) on which the US Weather Bureau psychrometric tables were based. In 1900, Professor Marrin produced new tables, though these too seem to have been based on empirical formulae.

Box[12] gave tables of the properties of moist air. Carpenter used these tables in 1910, but he also gave humidity data from the US Department of Agriculture. Poynting and Thomson ("Heat")[48] refer to Hazen's tables, published in Washington, and which are based on a simplified version of Apjohn's formula.

According to Ingels,[32] Willis Carrier was dissatisfied with the basis of the US Weather Bureau tables, and when in 1906 he wrote a catalogue for the Buffalo Forge Company, he included a new hygrometric chart. This was later refined, and in 1911 he published a paper entitled "Rational Psychrometric Formulae" which brought him international fame.[15] It dealt with sensible, latent and total heat, enunciated the theory of adiabatic saturation, and showed the relationship between the dry bulb, wet bulb and dewpoint temperatures of the air. The formulae presented by Carrier and his accompanying psychrometric charts (Fig. 13.9) were rapidly to become the authoritative basis for all of the fundamental calculations necessary for the design of air conditioning systems and equipment.

A psychrometric chart relating moisture content and RH was published by Marr in Germany in 1915. The Mollier diagram was devised in 1904, (Z.V.D.I.) and has been used as the basis of refrigerant tables and charts as well. Goff and Gratch (University of Pennsylvania) re-examined the bases of psychrometry in 1945, and produced the most accurate tables which now exist. Their work has been adopted by ASHRAE in America and by CIBS in Britain for their hygrometric tables.

13.14 THE COMPUTER REVOLUTION

The calculations required for the design of building services have grown enormously in complexity within recent years. When it was possible to consider only the maximum steady-state heat loss, and the steady flow of fluids in pipes, and when engineers could deal with each service separately, little other than simple arithmetic processes and the use of tables was necessary.

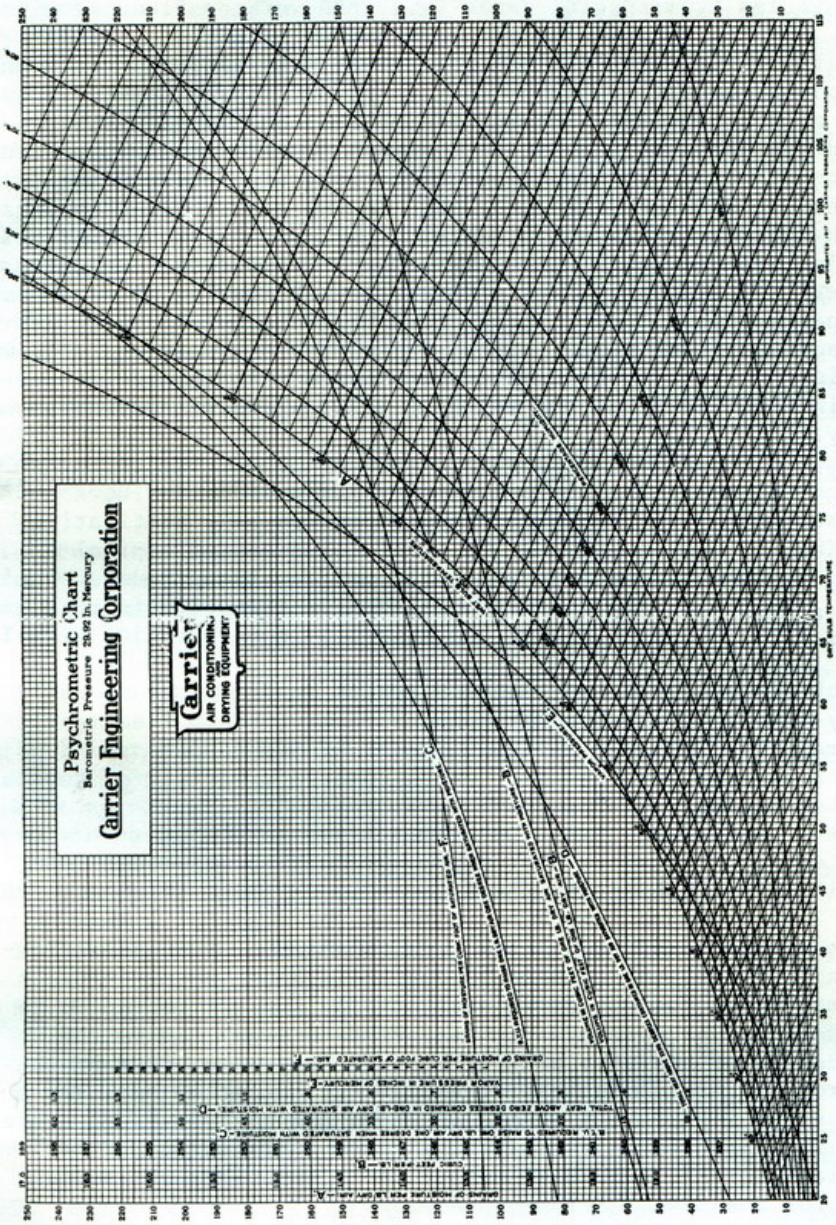
Changes in building construction (the advent of light structures), increasing variety of methods of heating, including air conditioning, each with differing characteristics (of which warm-air heating and floor-warming are extreme examples), the wider use of controls, and the economic forces all combined to render the steady state approach inadequate, and to require closer estimates of annual energy use.

The mathematical theory which was necessary for the solution of non-steady heat flow problems had been available since Fourier's classic work, about 1820. The practical application of the theory to the heating of buildings and to system analysis waited upon Heaviside's operational calculus, the Laplace transform and matrix algebra. There was also very limited knowledge of the thermal properties of building materials, of ventilation rates and of meteorological data.

But even when all the data and mathematical techniques were available, any solution of the probems was a tedious business.

Initially, recourse was had to mechanical means (Nessi and Nisolle[44]) of solving equations. About 1948, a graphical solution, based on an analogy with electrical conduction, and which could be applied to multi-layer structures, was provided by Marmet.[40]
The relaxation method of Southwell, and the Hardy-Cross finite difference method were also used in research studies.

None of these methods was suitable for use as a routine design procedure. They were, however, valuable in enabling quasi-empirical tables to be prepared (e.g. for intermittent heating allowances).



All References to Temperature are in Degrees Fahrenheit

Dry Bulb Temperatures are represented by vertical lines with values indicated on lower edge of chart.
Wet Bulb Temperatures are represented by oblique straight lines with values indicated on the curved line marked "Wet Bulb Temperature."
Dew-Point Temperatures are represented by horizontal lines and their values indicated on the curved line marked "Wet Bulb Temperature."
Percentages of Relative Humidity are represented by converging curved lines with values indicated between the oblique straight lines for 63° and 64° wet bulb temperature.
Any two of the above properties may be found if the other two are known. First, find the point of intersection of the lines representing the given properties, and then follow through this point, the lines representing the unknown properties, and the values of the latter can be read from their respective scales.
We will be glad to mail a larger reproduction of this Chart to any Engineer, upon request.

16 17

Fig. 13.9. Carrier Psychrometric Chart (1917).

At about the same time, analogue models (Beuken, Paschkis *et al.*) were developed for steady-state situations, and applied to heat-loss problems. Resistance-capacity networks were also employed to some extent in research. An analogue computer for ventilation networks in mines was constructed by Scott *ca.* 1953 — the noteworthy feature being the use of electric lamps instead of simple resistors to give a power-law relationship between pressure and velocity. As with the mathematical tools, these analogues were unsuited for design use, but were used to derive empirical formulae and tables from special cases (e.g. BRE work on floor warming).

The invention of the differential analyser and its development into the electronic computer enabled the complex equations to be quickly solved. In our case, it is the ability to solve a number of simultaneous differential equations, with complicated initial and boundary conditions (rather than a capability of handling otherwise intractable equations) which is significant. The high cost of the first computers restricted them to research laboratories. As the cost fell, they became more widespread, though few were in use in industrial design offices. Nevertheless, engineering graduates learned to program in their university courses and gained experience on the university computers. There began the publication of a stream of programs which would enable routine calculations of heat loss or fluid networks to be carried out in a fraction of the time required for manual computation. The way was open for optimisation — the repetition of a calculation with a range of parameters, with the final selection of one solution giving, for example, minimum cost, or minimum energy.

An iterative mathematical solution for fluid networks was developed by Howard[31] in 1957 while still a student at the National College for Heating, Ventilating, Refrigeration and Fan Engineering in London. Computer solutions of typical problems were successfully achieved, and Howard indicated how an optimum economic layout could be found. The first British commercial application of this type of program was in 1968, by M. B. Swain of Rosser and Russell. in cooperation with ICT.

A comprehensive range of air conditioning load and system design programs was introduced in the early 1960's by the Swedish company Fläkt (SF Air Treatment). In 1965, the American magazine *Heating, Piping, Air Conditioning* sponsored a major conference in Chicago on the use of computers, which covered a variety of topics including calculating heating and cooling loads, system simulation, energy studies, duct sizing and equipment selection. Over the next few years, many manufacturers have developed computerised equipment selection programs for refrigeration machines, heating and cooling coils and air handling plant (notably Carrier, Trane and Fläkt).

The art is perhaps most advanced in North America and in France, where a nation-wide computer network now exists, having been in operation since about 1970. A designer anywhere in France can feed specific data into a terminal connected to a central computer, and receive, as output, the appropriate design. Up to the present time, many thousands of radiator heating systems have been sized by the network. The French programs are compatible, that is, the output of one, say a heat-loss program, can serve immediately as the input to the next (e.g. pipe-sizing). The ultimate objective is a program which starts with the architect's conception, works through the services design, and whose output is an ordering list and a critical path analysis for the installation process.

A more recent innovation is the interactive display unit, in which a designer can adjust a parameter and immediately see, on a screen, the effect of the alteration.

The use of computers is not confined to engineering problems. Indeed, in general industry, this may well be a minor role. Apart from handling financial matters, however, it has found application in management, in particular in stock control and project management. Critical path networks were described at a computer conference in 1959: these identify those parts of a project where timing is critical. For

instance, non-delivery of material required for this stage inevitably delays completion. As another example, two or more trades may be engaged, and it may be essential for one trade to finish its task before the others can continue.

A distinct, but allied, use of computers is the estimation of annual energy use. Since the computer time for a single calculation is short, it is entirely feasible for computations to be made for each hour (or any other interval) of a day, and to repeat this for as many days as one wishes. If actual meteorological data are supplied, the resulting print-out is an estimate of the energy which would have been used hour by hour and day by day in that particular period. Alternatively (and this is now preferred) a real or synthetic "reference" or "example" year can be used. The effects on the operation of systems of especially severe weather are easily computed. Example programs are BEEP (Electricity Council, UK), THERM (British Gas Corporation) and TRACE (Trane Company, USA).

For ordinary design-office use, the value of the computer calculation is not the saving of time (since an ordinary designer spends as little as 5% of his time on calculation) but the possibility of increased accuracy and of optimising the variables. Programs which merely use pre-existing tables, or adopt manual methods, do not provide higher accuracy. Those which are based on the fundamental equations of heat diffusion (e.g. by computing admittance or response factors) do give more accurate data, but at the expense of increased computer time.

Computer-aided design is most valuable in air conditioning, for the first cost of over-sized plant is high, and the energy cost of inefficient operation is also high.

The computer revolution is well illustrated by BSRIA Bibliography LB106 "Computers for building services", issued in 1978. It includes 311 abstracts of articles written in the 1970's, and lists some 133 programs then available.

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