THE EVALUATION OF AN IMPROVED METHOD OF MONITORING ENERGY CONSUMPTION USED IN THE HEATING OF INDUSTRIAL BUILDINGS.

by

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CERTIFICATE OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this thesis, that the original work is my own except as specified in acknowledgements and in foot notes and that neither the thesis nor the original work contained herein has been submitted to this or any other institution for a higher degree.

J. H. Y. Katima

DEDICATION

This thesis is respectifully dedicated to:

My wife: HADIJA

and

My Daughter: ZAYNAB

ACKNOWLEDGEMENTS

I am indebted to many people who have helped me in the preparation of this thesis. And those whose opinions and approaches have over the months, helped to shape my own. I greatly value what I have learned from them. For specific help my thanks are due to a number of people and institutions whose names are given below.

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Lastly but not least to Miss Salome for her patience in typing the scripts.

J.H.Y. Katima

July, 1985.

ABSTRACT

The prospects for various forms of energy continue to engage the thoughts and pens of many people concerned with our future prosperity. The vulnerability of forecasts on energy especially those concerned with oil, to political, military and general economic activity, or lack of it highlights the importance of energy to practically everything we do.

For commercial and industrial users as well as public bodies, the cost of energy often has a significant impact on their present and future plans. Many have found that, only through energy conservation are they able to attain financial viability.For a sound energy conservation programme, the monitoring of energy consumption is very important. The cheap method of monitoring energy is to predict the energy use for particular prevailing operating conditions.

The degree day method for space heating plants, was adopted by the Department of Energy in 1977 and has been widely used ever since. This method assumes that the heat loss through the building structure and hence fuel consumption increases as the average outdoor temperature decreases. By using this method in conjunction with the Cumulative Sum technique, the efficiency of the space heating plant can be monitored.

When using this method errors of $\pm$ 25% should be expected, because the heat loss through building structure is not solely dependent upon the outside temperature. There are other factors such as the physical properties of the building and other meteorological features such as wind speed, humidity, solar radiation and precipitation which are not included in the simple mathematical model used for calculating degree days.

In this report attempts are made to quantify the effects of wind speed and solar radiation. The weekly energy data was supplied by a local

company and the daily weather data by Sutton Bonington Meteorological Office.

The weekly degree days based on the 15.5 °C base temperature were calculated and plotted against the weekly energy consumption. A Linear regression analysis was used and a correlation coefficient of 0.9392 and a standard error of 11.51% of mean were obtained.

The effects of wind and solar radiation were considered individually in conjunction with the degree days. Improvements of about 5.5% and 4.2% respectively were obtained. The combined effect of wind speed and solar radiation yields an improvement of about 7.5% on the original method. The significance of these improvements are discussed in more detail in the thesis.

Because of the nature of work carried out by the Company, the average inside temperatures in some areas are kept above the value of 18.3°C, recommended for normal office occupation. Thus the effect of changing the base temperature on the correlation coefficient and the standard error was considered. The results indicate that a higher base temperature of approximately 16.5°C is suitable for the situation.

Other non-linear relationship between energy consumption and the above factors were considered. The Cumulative Sum technique as the tool for energy control was able to highlight periods of significant change in consumption, although, because of the lack of complete historical data these were not all totally explained.

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NOMENCLATURE

A = Area, m^2

a = Solar absorptivity of the surface, (-)

C = Constant, (-)

C_p = Specific heat capacity, kJ/kg K.

DD = Degree days, °C

E = Energy consumption, kJ/day

E = Emissivity, (-)

e = Constant depending on location, (-)

F = Pressure coefficient, (-)

F_s = Surface factor, (-)

fr = Retransmission factor, (-)

GR = Global radiation, mWh/cm^2

g = Accelaration due to gravity, m/s^2

h = Heat transfer coefficient, W/m^2K

I = Infiltration rate, air change/hr or m^3/hr

I_G = Global solar irradiance on surface, W/m^2

I_L = Long wave radiation, W/m^2

K = Coefficient depending on terrain, (-)

k = Thermal conductivity, W/mk.

l = Length, m.

m_a = Mass flow rate, kg/s

m = Exponent, (-)

n = Exponent, (-)

P = Pressure, N/m^2

P_o = Power, W

P_r = Prandtil number, (-)

P_{ro} = Probability, (-)

Q = Heat flow, W.

q = Heat flow per unit area, W/m^2

R = Thermal resistance, m^2K/W

Re = Reynold's number, (-)

Rw = Run of wind, km/day

r = Correlation coefficient, (-)

S_e = Estimated standard deviation, (-)

T = Temperature, ^oC (K)

t = Time, s

U = Overall heat transfer coeffience, W/m^2K

V = Volume, m^3

v = Wind speed, m/s

W = Work, J

X = Thickness, m

Y = Admittance, W/m^2 K.

y = Fuel consumption, litres/week

Z = Height, m.

OTHERS (GREEK)

ρ = Density, kg/m^2

σ = Standard deviation, (-)

σ_o = Stefan - Boltzman's constant, (-)

ε = Effectiveness of opening, (-)

α = Proportion of radiation absorbed, (-)

γ = Proportion of radiation reflected, (-)

τ = Proportion of radition transmitted, (-)

μ = Decrement factor, (-)

CHAPTER 1

INTRODUCTION

1.1 The energy problem

To date, the world has witnessed three major international energy crises, viz:

(a) The 1973/74 oil embargo.

(b) The 1979 crisis, arising from the political upheaval and subsequent revolution in Iran.

(c) The current crisis, arising from the Gulf War which has entered a serious phase in that oil tankers and oil installations have become military targets.

There are also, some national domestic problems such as the 1984/85, 51 weeks long, U.K. miners' strike, which may not always be regarded as energy crises, although their impact on energy availability and prices can be significant.

The consequences of the above events are in general terms:

(a) The scarcity of the commodity on the market.

(b) The accelerated depletion of other relatively cheap fossil fuel reserves, especially those of coal and gas.

(c) The escalation of fuel prices.

Thus, it should be apparent to any energy consumer be he in the industrial or domestic sector that, as long as fossil fuels remain his major source of energy, the efficient use of that energy is very important. As it will not only enable him attain financial viability, but also will eventually help to extend the period of time for safe and proper development of nuclear power and renewable sources of energy such as solar, wind and tidal power.

It should be appreciated that there is no shortage of symposia and publications on energy conservation in industry. However, it is human nature not to be careful about wastage of energy especially when the cost of it is not borne by the individual. That is why over half of the energy used by man is believed to be wasted (34,75), despite the fact that the campaign to save energy started more than a decade ago.

1.2 Energy demand

Tables 1 - 1 to 1 - 3 (16) show the total primary energy inputs for the whole world, U.S.A. and Western Europe respectively, over the period 1965 to 1982. Fig 1 - 1 and 1 - 2 show the same data for U.K. between 1950 and 1980. The statistics show that there was an upward trend in total energy consumption which subsequently levelled off after the 1973/74 energy crisis.

Evidently oil is the most important energy input during this period since its proportion in the energy mix has increased considerably while that of coal has shown a steady decline. The proportion of natural gas has increased also from a minor 0.1% in 1963 to 26.9% in 1983. Fig.1 - 1 to 1 - 5 illustrate these trend clearly.

Many factors affect the choice of industrial fuel, but cheapness, ease of transportation,conversion to heat could explain the obvious switch from coal to oil between 1950 and 1970. After the 1973/74 oil crisis the proportion of oil started to fall.

The dotted lines in Fig. 1 - 1 and Fig. 1 - 2 indicate the projection of U.K. energy demand, up to the end of the century (the projections were made just after the 1973/74 oil embargo). The following assumption were made in working out these forecasts:-

(a) A low economic growth rate of 1 - 2 % in the U.K.

TABLE 1 - 1 :- WORLD PRIMARY ENERGY CONSUMPTION 1965 - 1982 (16)

PRIMARY ENERGY	1965		1970		1972		1974		1976		1978		1980		1982	
	GJ X10^9	%	GJ X10^9	%	GJ X10^9	%	GJ X10^9	%	GJ X10^9	%	GJ X10^9	%	GJ X10^9	%	GJ X10^9	%
OIL	68.54	39.0	102.14	44.0	115.58	47.7	122.30	44.6	129.02	44.6	137.98	45.7	134.40	43.6	128.34	41.2
GAS	29.12	16.5	43.00	18.5	45.25	18.7	49.73	18.1	51.07	17.6	55.10	18.2	87.34	18.6	58.69	19.2
COAL	67.20	38.0	72.58	31.0	76.61	31.6	83.33	30.4	88.26	30.5	84.22	27.9	90.50	29.3	91.39	29.8
NUCLEAR	11.20	6.5	0.90	0.5	0.45	0.2	2.69	1.0	4.48	1.5	6.72	2.2	7.62	2.5	9.86	3.2
HYDRO			13.89	6.0	4.48	1.8	16.13	5.9	16.58	5.9	17.92	5.9	18.37	6.0	20.16	6.6
TOTAL	176.06	100	232.51	100	242.37	100	274.18	100	289.41	100	301.95	100	308.22	100	306.43	100

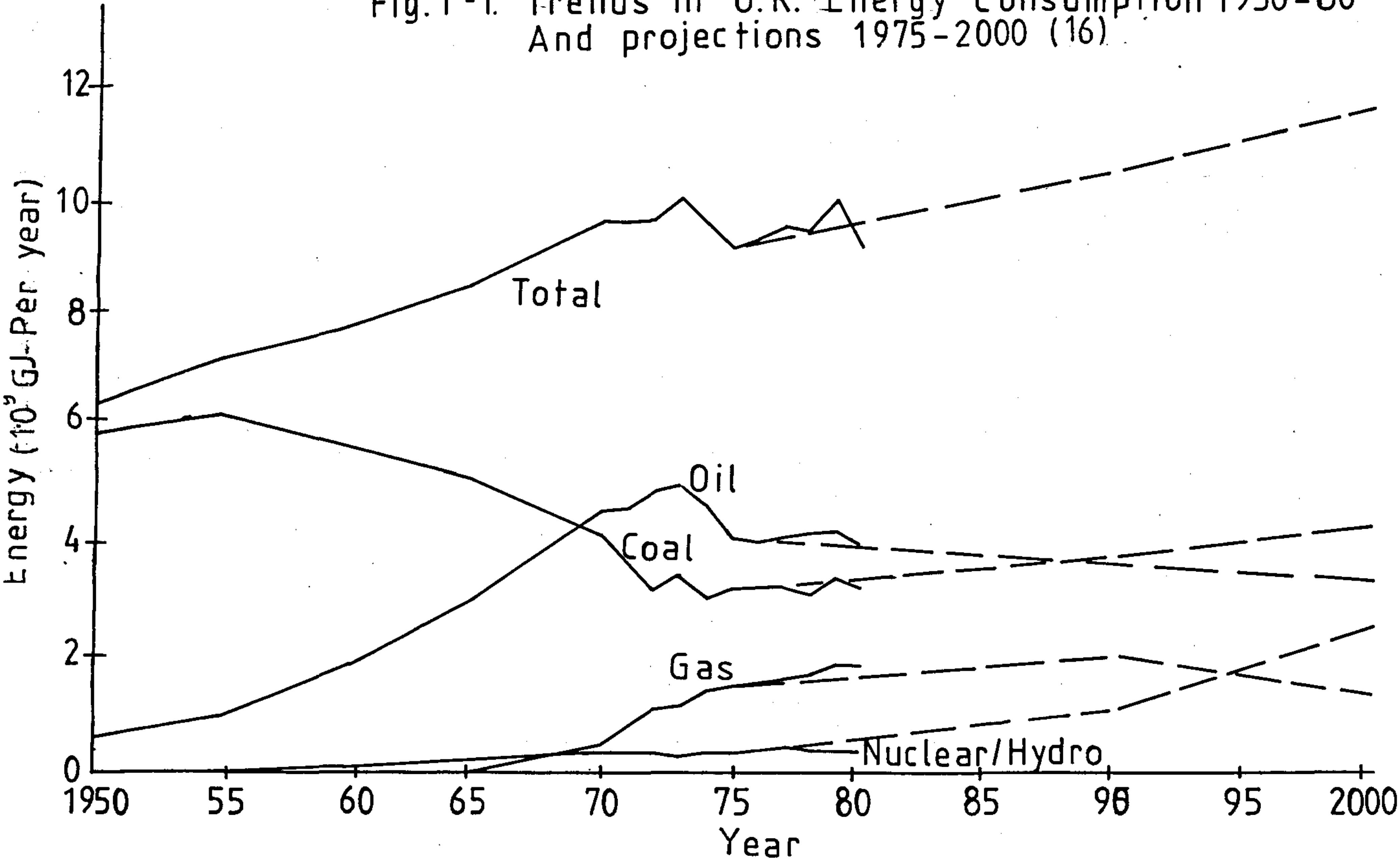
TABLE 1 - 2 :- U. S. A. PRIMARY ENERGY CONSUMPTION 1965 - 1982 (16)

<table>
<tr><th rowspan="2">PRIMARY ENERGY</th><th colspan="2">1965</th><th colspan="2">1970</th><th colspan="2">1972</th><th colspan="2">1974</th><th colspan="2">1976</th><th colspan="2">1978</th><th colspan="2">1980</th><th colspan="2">1982</th></tr>
<tr><th>GJ X10^9</th><th>%</th><th>GJ X10^9</th><th>%</th><th>GJ X10^9</th><th>%</th><th>GJ X10^9</th><th>%</th><th>GJ X10^9</th><th>%</th><th>GJ X10^9</th><th>%</th><th>GJ X10^9</th><th>%</th><th>GJ X10^9</th><th>%</th></tr>
<tr><td>OIL</td><td>24.64</td><td>41.0</td><td>28.67</td><td>44.4</td><td>30.91</td><td>41.6</td><td>34.94</td><td>46.7</td><td>34.94</td><td>43.8</td><td>36.87</td><td>46.6</td><td>35.39</td><td>42.9</td><td>31.36</td><td>40.7</td></tr>
<tr><td>GAS</td><td>19.26</td><td>32.1</td><td>20.61</td><td>31.9</td><td>25.09</td><td>33.7</td><td>25.09</td><td>33.7</td><td>25.09</td><td>31.5</td><td>22.4</td><td>26.2</td><td>21.95</td><td>26.6</td><td>20.61</td><td>26.7</td></tr>
<tr><td>COAL</td><td>13.89</td><td>23.1</td><td>14.34</td><td>22.2</td><td>14.78</td><td>19.9</td><td>13.44</td><td>18.0</td><td>15.23</td><td>19.1</td><td>16.13</td><td>18.8</td><td>18.37</td><td>22.3</td><td>17.47</td><td>22.7</td></tr>
<tr><td>NUCLEAR</td><td rowspan="2">2.24</td><td rowspan="2">3.8</td><td rowspan="2">0.90</td><td rowspan="2">1.5</td><td>0.45</td><td>0.6</td><td>0.45</td><td>0.6</td><td>1.34</td><td>1.7</td><td>3.58</td><td>4.2</td><td>3.14</td><td>3.8</td><td>3.58</td><td>4.7</td></tr>
<tr><td>HYDRO</td><td>3.14</td><td>4.2</td><td>0.90</td><td>1.2</td><td>3.14</td><td>3.9</td><td>3.58</td><td>4.2</td><td>3.58</td><td>4.4</td><td>4.03</td><td>5.2</td></tr>
<tr><td>TOTAL</td><td>60.03</td><td>100</td><td>64.52</td><td>100</td><td>74.37</td><td>100</td><td>74.82</td><td>100</td><td>79.74</td><td>100</td><td>85.56</td><td>100</td><td>82.43</td><td>100</td><td>77.06</td><td>100</td></tr>
</table>

TABLE 1 - 3 :- WEST EUROPEAN PRIMARY ENERGY CONSUMPTION 1965 - 82 (16)

PRIMARY ENERGY	1965		1968		1970		1972		1976		1978		1980		1982	
	GJ $X10^9$	%	GJ $X10^9$	%	GJ $X10^9$	%	GJ $X10^9$	%	GJ $X10^9$	%	GJ $X10^9$	%	GJ $X10^9$	%	GJ $X10^9$	%
OIL	17.47	45.9	22.85	56.7	28.22	58.9	31.36	63.1	31.81	56.8	32.26	56.3	30.46	53.1	26.88	49.6
GAS	0.90	2.4	1.79	4.4	3.14	6.5	4.93	9.9	7.17	12.8	7.62	13.3	8.06	14.1	7.62	14.0
COAL	16.13	42.3	14.34	35.6	12.10	25.2	11.65	23.4	11.65	20.8	11.20	19.5	12.10	21.1	11.65	21.5
NUCLEAR	3.58	9.4	1.34	3.3	0.45	1.0	0.45	0.9	1.34	2.4	1.79	3.1	2.24	3.9	3.14	5.8
HYDRO					4.03	8.4	1.34	2.7	4.03	7.2	4.48	7.8	4.48	7.8	4.93	9.1
TOTAL	38.08	100	40.32	100	47.94	100	49.73	100	56.0	100	57.34	100	55.55	100	54.21	100

Fig. 1-1. Trends in U.K. Energy Consumption 1950-80 And projections 1975-2000 (16)



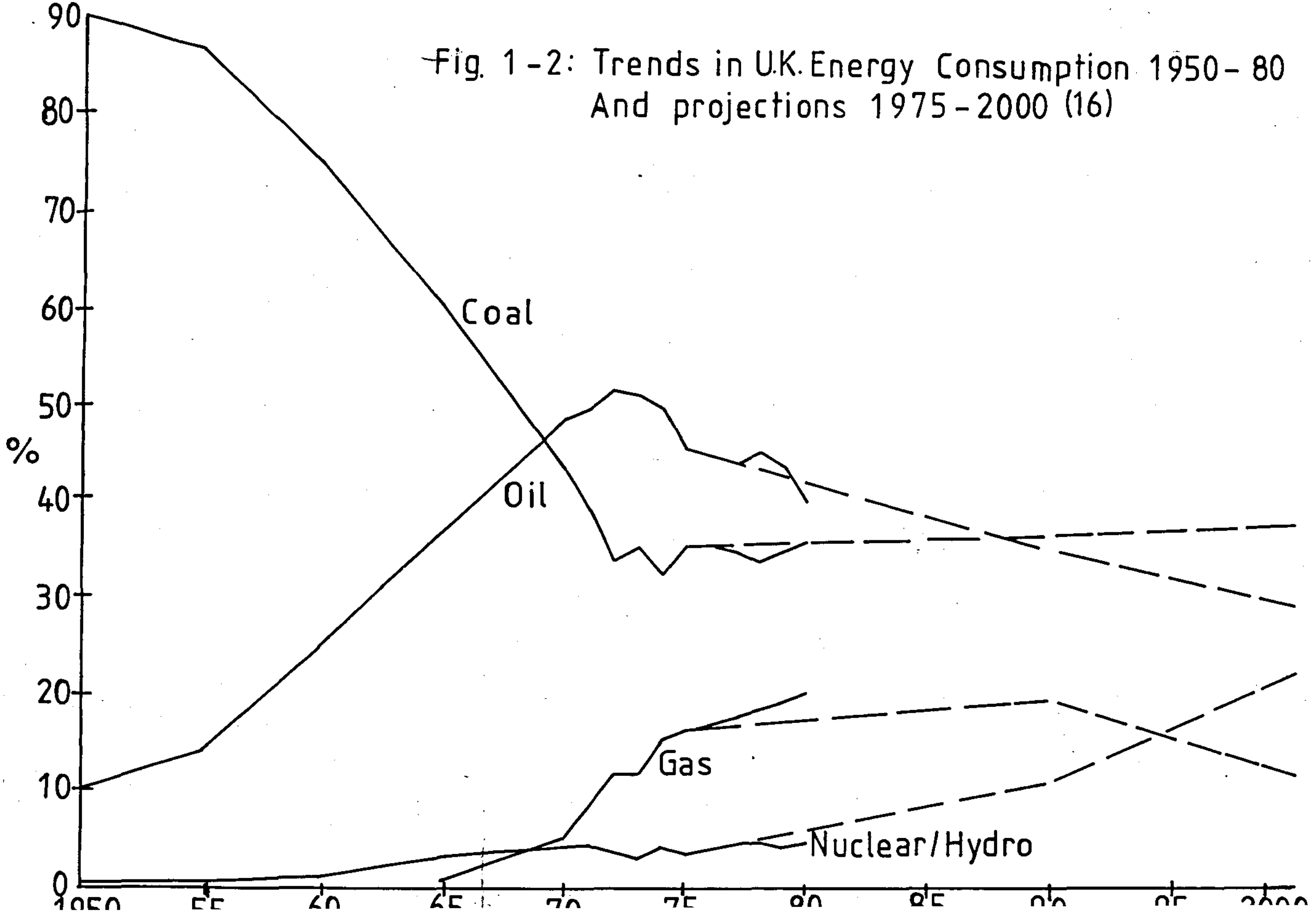


Fig. 1-2: Trends in U.K. Energy Consumption 1950-80
And projections 1975-2000 (16)

U.K. PRIMARY FUELS CONSUMPTION PERCENTAGE SHARES.

Fig.1-3: 1963 ÷ 8.48×10^9 GJ (16).

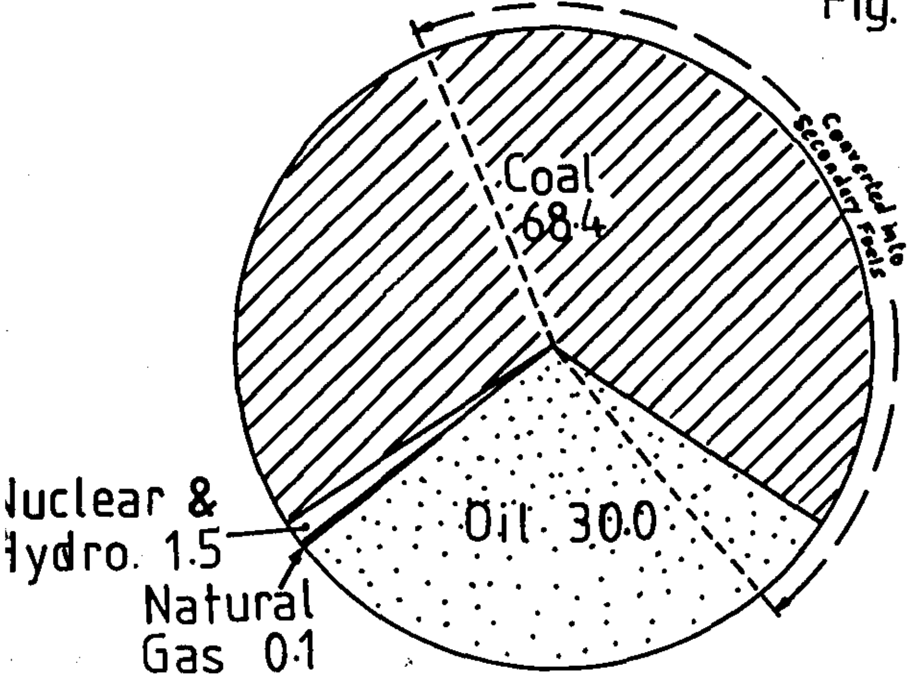
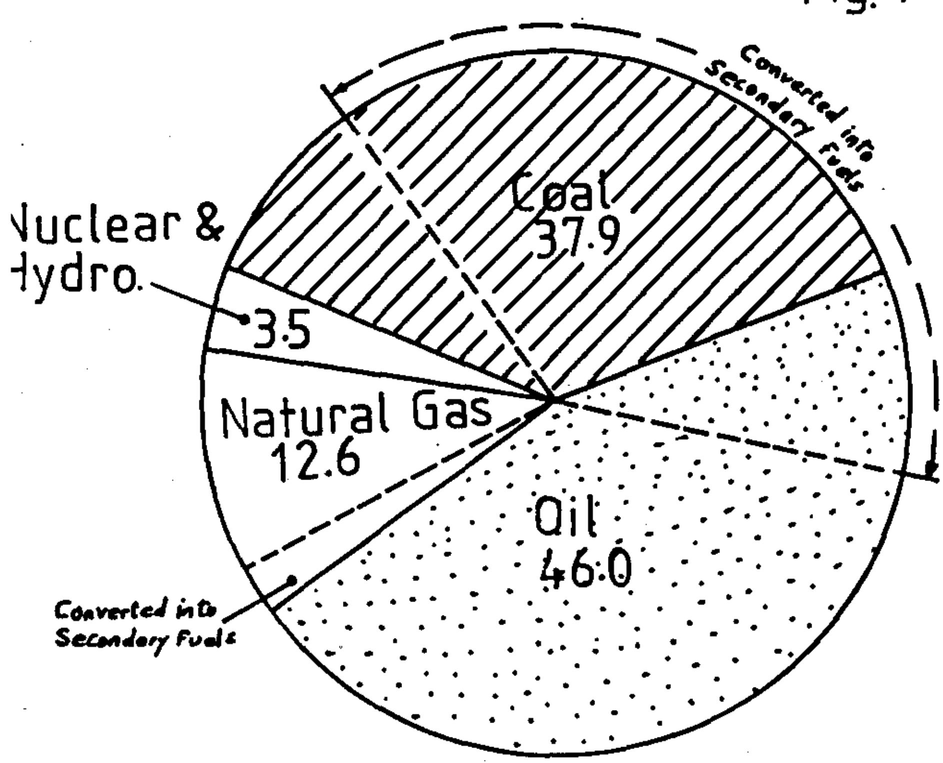
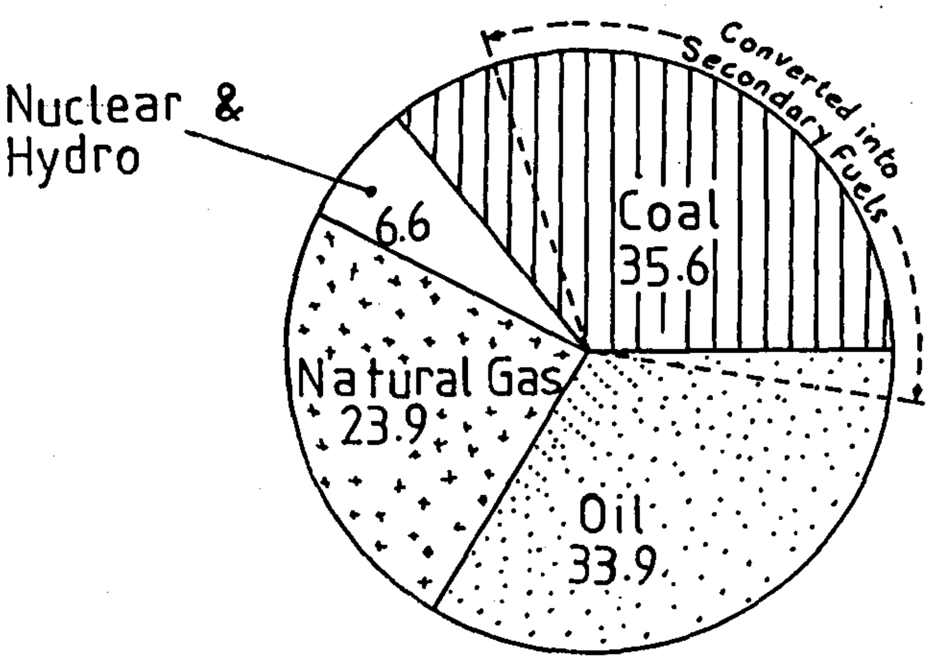


Fig. 1-4: 1973 ÷ 10.34×10^9 GJ (16)



U.K. PRIMARY FUELS CONSUMPTION
PERCENTAGE SHARES

Fig.1-5: 1983 :- 9.35×10^9 GJ



up to the end of the century.

(b) Energy prices were predicted to rise more slowly during the rest of the century than during the last decade.

(c) Increased implementation of energy saving measures will yield an approximate 15% reduction in energy consumption below what would otherwise - have been used.

The forecast show a continued rise in total energy consumption, but at a slower pace than that experienced in the 50's and 60's when prices were very low. During the period of 1950 to 1970, for example, the increase in energy consumption averaged a robust 3.3% per year but fell to 2.7% in 1977 and 2.1% in 1978. Similarly electricity increase averaged a remarkable 7.4% during the period 1950 to 1977 but only 5.1% in 1977 and 3.7% in 1978.

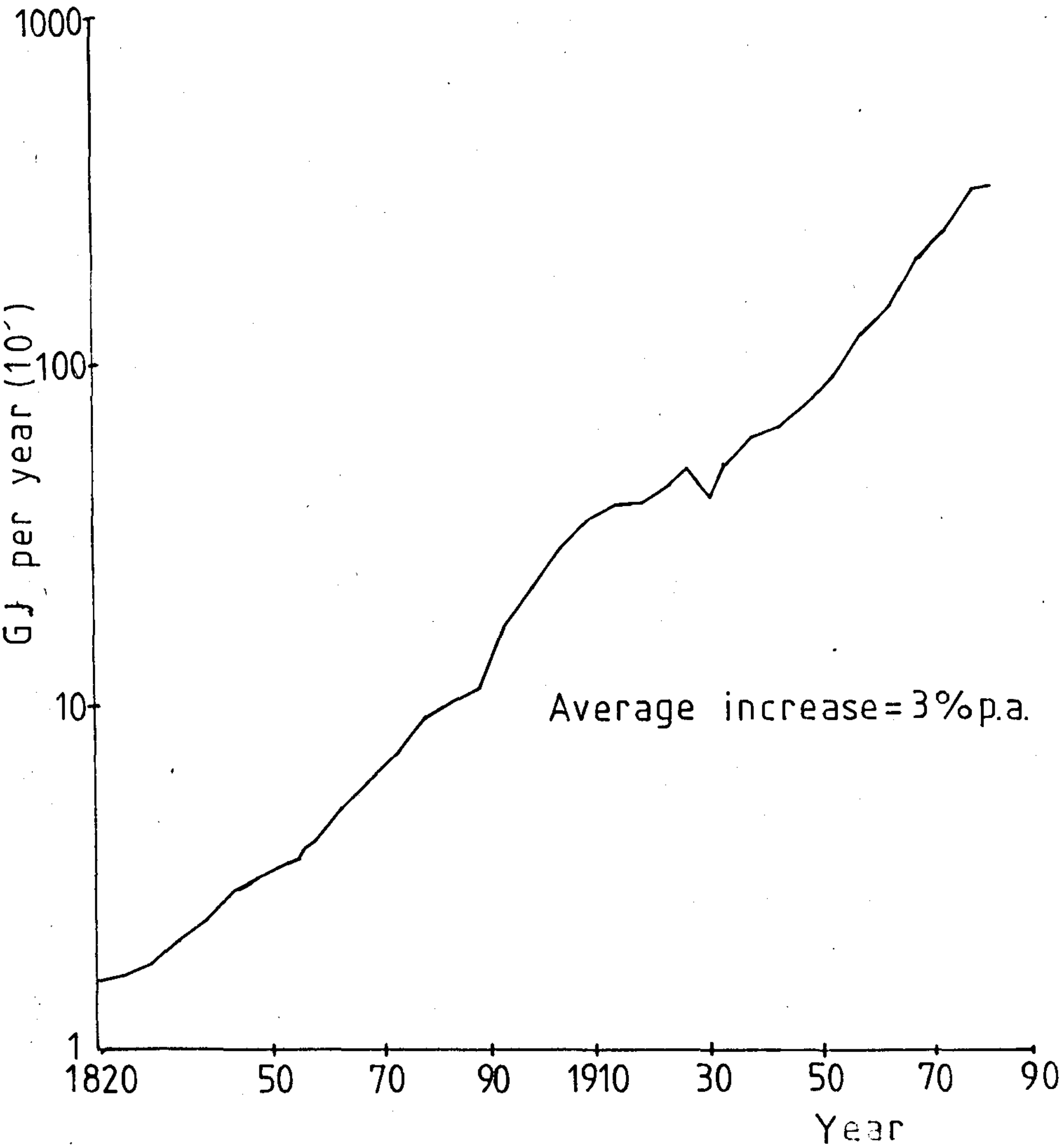
However, Fig. 1 - 6 and 1 - 7 show the world energy consumption over the period 1820 to 1980 and U.K. electricity output over the period 1920 to 1980(1). These two illustrate that, in general terms, these resent changes are still fairly small on the total upward trend of the energy consumption and thus the fear for rapid deplition of fossil fuels is still persistent.

1.3 Changes in Energy Markets Since 1973 - 74 Oil Embargo

Two fundamental changes have occured in energy markets since the beginning of 1973. First, energy supplies display extreme short term sensitivity to polytical events. Second, high energy prices produce dramatically different energy demand - supply relationships (16, 45). Each of these changes, which may now say were predictable, caught the energy world by suprise and, as a result, had major concequences.

In this period, the price of oil has increased by 345% in real terms, even after the recent downturn. The sharp hikes of 1973 - 74 and

Fig.1-6:WORLD ENERGY CONSUMPTION (1)



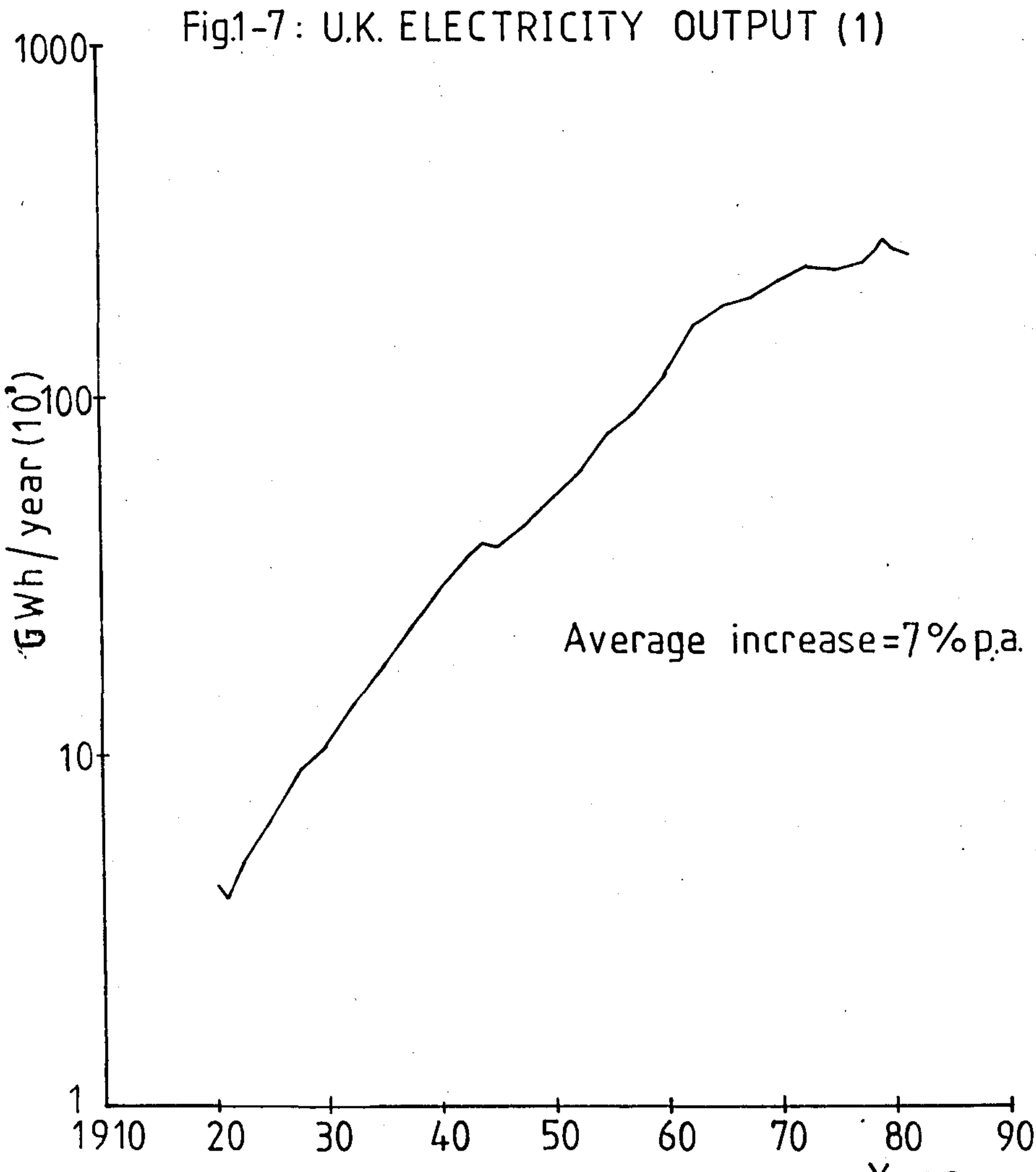


Fig.1-7: U.K. ELECTRICITY OUTPUT (1)
1000
100
10
1
GWh/year (10^3)
Average increase = 7% p.a.
1910
20
30
40
50
60
70
80
90
Year

1979 - 80 were at least partly responsible for trigering deep, worldwide recessions. In the developed nations, households near the poverty level often were pushed over the edge by soaring heating bills. The developing nations' oil - related depts placed unprecedented strains on the international financial system. Perhaps most ominous of all the United States of America was forced to announce that, because of oil, the Persian Gulf had become an area of vital strategic interest - making more conceivable a catastrophic confrontation with the Soviet Union.

Not all of this should be ascribed to the failure to anticipate the two previous mentioned market changes. As a depleting resource, oil may face long term increases in cost. And even if prepared for the new energy developments, industrial economies could not make easy and swift adjustment in energy use perterns. But in the final analysis, during this period, things were worse than they had to be, and a good part of the reason is that government and industry responses always seemed to be one step behind the market.

Take the first important change - the increasing sensitivity of energy supplies to political events. The inability to anticipate this was, of course, a major contributing factor to the distruption caused by the 1973 - 74 embargo and to the ability of Saudi Arabia and other oil producers to sustain price hikes that followed. Even after this event the forecasters predicted Opec's Collapse and emergency planning got little attention.

Thus, when the Iranian revolution began to affect oil production in November 1978, the oil consuming nations were unprepared. Consequently the oil prices wet up by around 140%.

1.4 Overall Energy Consumption in the U.K.

A breakdown of the energy consumption for the period between 1979 to 1983 by all U.K. end users in terms of Sectors is presented in table 1 - 4 (16).

It is very interesting to note a falling trend of the industrial energy consumption (i.e. 37.6% to 31.3% of the total energy consumption between 1979 and 1983). Nevertheless, the industrial sector, remains the

TABLE 1 - 4 :- SECTORAL ENERGY CONSUMPTION IN U.K. 1979 - 1983 (16)

Sector	ENERGY CONSUMPTION									
	1979		1980		1981		1982		1983	
	GJ $X10^9$	%	GJ $X10^9$	%	GJ $X10^9$	%	GJ $X10^9$	%	GJ $X10^9$	%
Industry	2.44	37.6	2.02	33.9	1.91	33.0	1.84	32.1	1.79	31.3
Transport	1.48	22.8	1.49	24.9	1.41	24.8	1.47	25.6	1.63	26.4
Domestic	1.78	26.7	1.67	29.2	1.66	28.7	1.64	29.7	0.36	28.6
Public Administration	0.4	6.1	0.37	6.3	0.37	6.4	0.36	6.3	0.36	6.4
Agriculture	0.08	1.2	0.06	1.1	0.06	1.0	0.06	1.0	0.06	1.0
Miscellaneous	0.37	5.6	0.35	5.9	0.35	6.1	0.36	6.2	0.06	6.3
Total	6.51	100	5.96	100	5.79	100	5.73	100	5.71	100

greatest energy user. However, the importance, in terms of energy conservation, lies in the potential of saving energy and the cost of conservation measures. The cost of most of the conservation measures in industry are rather high and makes them less attractive when compared to those in the domestic sector, which is almost as large. Most of the energy in the domestic sector is used in space heating, and thus insulation in buildings, which is cost effective, if applied universally, could reduce the domestic energy consumption by 25% to 30% (i.e. 10% of the total) (48, 49, 50). Thus, the potential of saving energy in industry is largely dependent upon the commitment to energy monitoring and control.

1.5. Project Objectives

The industrial conservation programme includes;

(a) Efficient production, distribution and utilisation of energy.

(b) Combined heat and power generation.

(c) Heat and power recovery.

(d) Good housekeeping.

(e) Energy management.

(f) Energy auditing, monitoring and control.

In order to have a sound and successful energy conservation programme, a mechanism of assessing the effect of the programme is essential. Thus, a reliable monitoring and control mechanism should be devised.

It has been common practice to use the degree day method to predict the energy consumption for space heating plants (12, 39, 44, 47). By definition the degree days are equivalent to the number of degrees (^{o}C of ^{o}F) that the 24 hour mean outside ambient temperature lies below a selected base temperature. This method states that energy consumption for the space heating plant is proportional to the degree days. By plotting the accumulated difference between actual consumption and the predicted consumption, the trend of the energy consumption can be monitored and hence controlled.

The term degree day originated with the gas industry of America as early as 1915 and was later standardised by the American gas Association (44). Barker (7) in 1933 used it to assess the fuel consumption for space heating. It was adopted by the Department of Energy in 1977. It has been found that when this method is used errors ranging from ±15% to ±25% should be expected. However, it is stated in the fuel efficiency booklet number 7 (27) that "Fuel consumption is not solely related to ambient temperature (degree days)". There are other meteorological influences such as wind speed, humidity, solar radiation and clouds and physical properties of the building fabric, which are not included in the original method.

Thus the objective of this thesis is to try to improve the accuracy of this method by including the effect of wind speed, solar radiation and other factors in energy equation.

A local pharmaceutical research and development firm, supplied the energy data dating back to 1975, whilst the daily weather data dating back to 1979 was supplied by sutton Bonington Meteorological office, which is situated around 15 miles from the company site. The energy in this firm is used almost entirely for space heating, although there are some small pilot plants.

Different mathematical models, linear and non-linear are considered with variable base temperatures. The results are presented in chapter 6 and discussed in chapter 7 of this thesis.

CHAPTER 2

GENERAL BACKGROUND

2.1 Aspects of Energy Conservation

Energy conservation is broadly categorised into three main actions, viz: (27, 43, 89, 105)

(a) Short term, "Good housekeeping measures" with immediate application at minimum or no capital cost.

(b) Medium term, one to three years, requiring detailed evaluation of capital investment schemes, using almost entirely existing technology.

(c) Long term, more than three years, generally involving major process changes and large capital cost or alternatively, involving developing of new, more energy efficient processes, with relatively low capital cost.

The important ingredients in achieving either of the above categories include some or all of the following:-

(i) Efficient production and distribution of energy.

(ii) Energy Management.

(iii) Energy auditing and monitoring.

(iv) Combined heat and power generation.

(v) Good housekeeping.

(vi) Heat and power recovery.

These aspects are briefly discussed, in the above order, in this chapter.

2.1.1 Efficient Production and Distribution of Energy

Fossil fuel is not an end in itself. The chemical energy that it contains is converted into heat when it burns. This energy is transmitted through the wall of the boiler furnace to the water whose temperature is raised until it forms steam. The steam is produced solely to serve as a convinient carrier of energy in the form of heat and pressure. As used in this context efficient production and distribution of energy refers to steam generation and distribution.

2.1.1.1 Boiler Selection and Operation

The term "boiler" covers a multitude of designs from simple low pressure vessels to provide hot water to advanced designs for high pressure steam for power generation. In addition various firing methods are employed to match the fuels available, from simple atmospheric gas burners to pressurised fluidised bed combustors (5, 34, 43, 68, 97). In order to attain a high thermal efficiency and economical operation attention should be given to the initial selection of the boiler plant, i.e. the plant that will;

(a) Meet current and future maximum load.
(b) Match daily and seasonal load variations (in temperate countries).
(c) Be robust and reliable.
(d) Be capable of high thermal efficiency.

Ofcourse, much of the responsibility for high thermal efficiency rests with the designer of the boiler. Once it is installed the measures that can be taken to maintain high efficiency are often limited to keeping heat exchange surfaces clean, finding the minimum excess air levels and good maintenance practice.

Boiler ratings, commonly expressed in kW, Btu per hour or pounds per hour in the U.K., defines the capability of the boiler, at atmospheric pressure, to evaporate water at its boiling point to produce dry saturated steam (2, 5). Ratings, however, are determined under ideal test-bed conditions without blowdown. In actual operation the feed water supply to the boiler may be much less than 100 ^{o}C and the operating pressure will be higher than atmospheric therefore, the actual steam production will be lower than the rated value.

Nevertheless, the boiler capacity selected must not be higher than necessary for the intended duty. However, there is a natural tendency to oversize boilers, probably because a shortfall in steam availability generally creates more rumpus than the higher initial and operating costs that arises from over - capacity.

It is sometimes advantageous to spread the total load over two or more boilers whenever possible, as it enables the total load to be matched more satisfactorily, it facilitates regular maintenance and statutory inspection and complete shutdown is avoided if a boiler should fail (34, 75). However, this decision increases the capital cost, therefore, it should be handled with great care.

2.1.1.2 Boiler Thermal Efficiency

The boiler thermal efficiency is the percentage of available energy that is converted into useful energy as heat in the steam or hot water (5, 43, 76). Minimum operating costs are achieved by running boilers at high thermal efficiency, and for modern boilers, based on gross calorific value, 80% is regarded as a reasonable target at full load (90, 98).

For high thermal efficiency, the user should pay close attention to the various sources of heat loss, viz:

(a) Convection and radiation losses from outer surfaces of the boiler, sometimes erroneously referred to as "radiation" losses.
(b) Incomplete combustion.
(c) Heat carried out in the boiler blow down.
(d) Heat carried out in flue gases.
(e) Residue heat lost during non-firing periods.

Convection and Radiation Losses

These are of decreasing signficance with improved boiler insulation and hoods and shrouds to enclose firing equipment. Nevertheless, as the outer shell of the boiler is usually at constant temperature, and hence constant convection and radiation losses regardless of the boiler load, these losses represent a higher proportion of the total heat input as the boiler load decreases (89).

Incomplete Combustion

This is dependent upon the amount of excess air used for combustion, (i.e. air supplied to the boiler above the minimum stochiometric amount of oxygen required for complete combustion), which in turn depends upon the control of the quantities of fuel and air used, the effectiveness of their mixing and on the environment in which combustion is taking place. If for example, burners, nozzles, atomisers, grates, quarls, arches and so on are in good condition and air and fuel supply rates are well controlled, then the excess air can be reduced to low levels without producing significant quantities of carbon or carbon monoxide (43, 75).

Heat losses due to incomplete combustion

The heat evolved when 1 kg of carbon is completely combusted is given by: (93)

$$C + O_2 \longrightarrow CO_2 \qquad \Delta H = 33.96 \text{ MJ/kg K}$$

When the combustion is incomplete however, the heat evolved on formation of carbon monoxide is:

$$C + \tfrac{1}{2}\, O_2 \longrightarrow CO \qquad \Delta H = 10.23 \text{ MJ/kg K.}$$

Thus, for every kg of carbon which forms carbon monoxide rather than carbon dioxide, (33.96 - 10.23) = 23.73 MJ are lost, which is often a significant proportion of the calorific value of the fuel. Where carbon is completely unburned as in formation of smoke or in loss of carbon in ash, then all of the 33.96 MJ are lost.

Flue gas losses

This is also related to the amount of excess air used and the temperature of the flue gases. While it is essential to supply excess air to ensure complete combustion, there is a need to minimise the excess air in order to control SO_x formation and to reduce stack losses, because any air entering the boiler removes heat in the form of sensible heat (34, 43, 97).

A flue gas temperature rise of 17 $^{\circ}$C results in a reduction in efficiency of about 1% at the same excess air levels (75). The temperatures can be limited by cleaning tubes and other heat exchange surfaces.

The best opportunity for cutting flue gas losses lies usually in control of excess air used, or, rather, not used in combustion, determined, conveniently by measurement of carbondioxide or oxygen content in the dry flue gases. The relationship between excess air and carbondioxide and oxygen are shown in Fig.2 - 1 (27).

Numerical example:

We want to quantify the percent heat loss in the flue gas when methane is burned with its theoretical air. The flue gas temperature is 257 $^{\circ}C$ and ambient temperature is 15 $^{\circ}C$. (Calorific value of methane 44.6MJ/m^3 (93), mean specific heats are CO_2 1.516, H_2O 1.409, N_2 1.245, air 1.316 kJ/m^3 K).

Basis : 100 k mol CH_4

$$CH_4 + 2\ O_2 \longrightarrow CO_2 + 2\ H_2O$$

100 kmol CH_4 require 200 kmol O_2 or $(200 \times \frac{100}{21}) = 952$ kmol air.

Therefore, theoretical air = 952 kmol/100 kmol CH_4

$= 9.52\ m^3/m^3\ CH_4$

In the combustion, 200 kmol H_2O and 100 kmol CO_2 are produced.

Nitrogen in air supplied = (952 x 79/100) = 752 kmol

Thus, on the basis of 1m^3 CH_4 burned, the flue gas contains:

1.0 m^3 CO_2, 2.0 m^3 H_2O and 7.52 m^3 N_2 (total 10.52m^3).

Sensible heat:

$= ((1 \times 1.516) + (2.0 \times 1.409) + (7.52 \times 1.245)) \times (257 - 15)$

$= 3.52\ MJ/m^3\ CH_4$

Heat in air = $9.52 \times 1.316\ (15 - 0) = 188\ kJ/m^3\ CH_4$

Sensible heat lost in flue gas $= (3.52 - 0.188)$

$= 3.332\ MJ/m^3\ CH_4$

$= \frac{3.332 \times 100}{44.6}$

$= 7.47\%$

Taking the latent heat of water vapour into account:

% by volume of water vapour = 2 x 100/10.52

= 19%

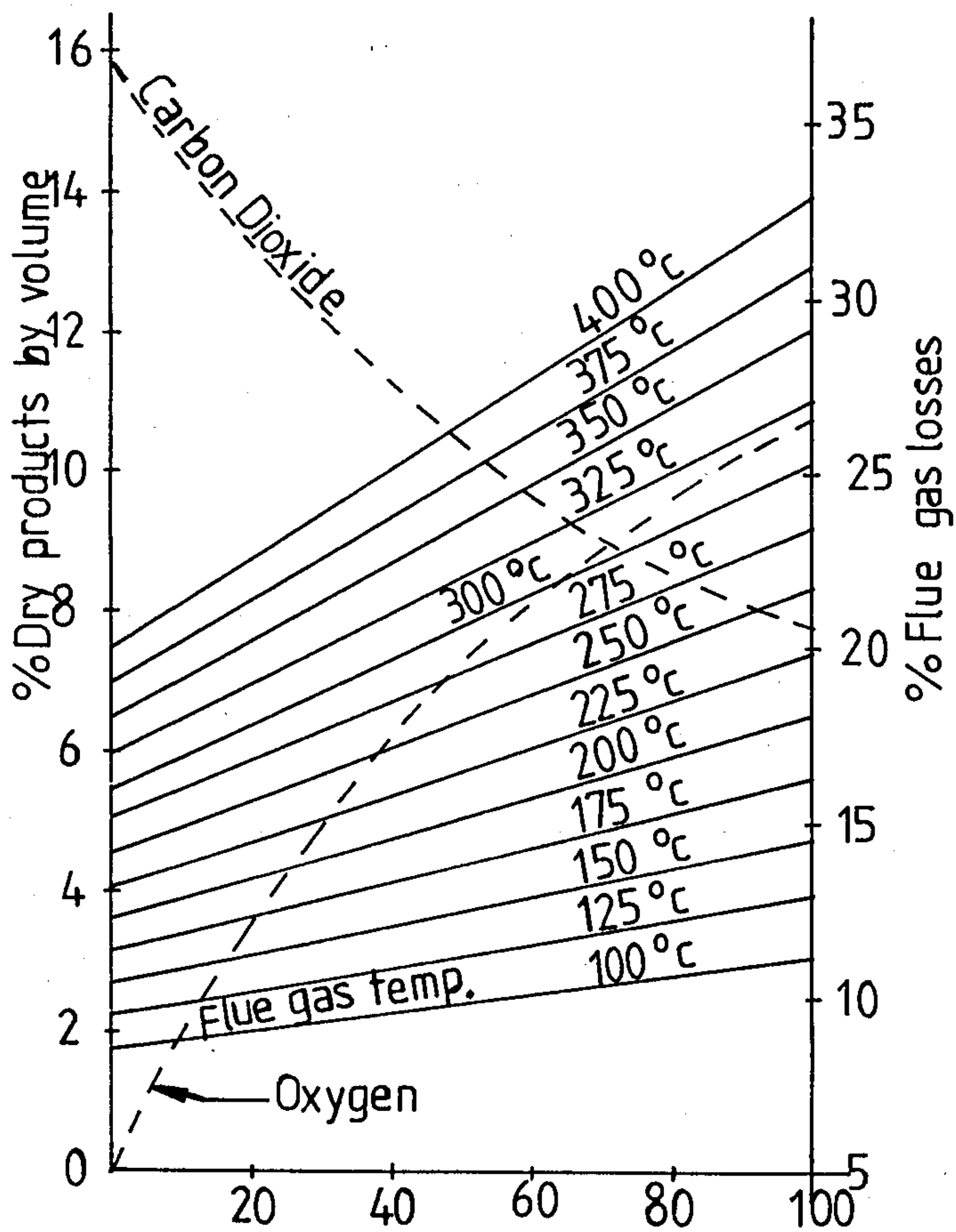


Fig.2-1: Flue gas losses—heavy fuel oil.
Based on gross calorific value and ambient te
(27).

Therefore, partial pressure of water vapour

$$= 19 \times 101.3/100 = 19.3 \text{ kN/m}^2$$

from tables (93), the vapour pressure of water vapour is 19.3 kN/m^2 at 60 °C at which the latent heat = 2353 kJ/kg.

For 1 m^3 CH_4 burned, 2 m^3 H_2O is formed or (2 x 10/22.4) = 1.61 $kg/m^3 CH_4$.

Therefore, heat in water vapour = (Sensible heat in water)+ (latent) + (sensible in vapour)

$$= 1.61\ (4.187\ (60 - 0) + 2353 + 2.09\ (257 - 60)$$

(where: 2.09 = specific heat of water vapour, kJ/kgK)

$$= 4.856 \text{ MJ/m}^3\ CH_4$$

Heat in CO_2 and N_2

$$= ((1.0 \times 1.516) + (7.52 \times 1.245))\ (257 - 0) = 2.796 \text{ MJ/m}^3\ CH_4$$

Total heat loss in flue gas = (4.856 + 2.796)

= 7.652 MJ/m^3

Heat in air (as before) = 188 kJ/m^3

Therefore sensible and latent heat in flue gas

$$=(7.652 - 188 \times 10^{-3}) = 7.464 \text{ MJ/m}^3$$

or (7.464 x 100)/44.6

=16.7 % which is quite significant.

Boiler blowdown

Sludge and precipitated salts in the boiler water are controlled by blowing a proportion of water from the boiler. The amount of discharge depends upon the method of water treatment used. However, blowdown is a source of heat loss which could account for about 3% of boiler heat input (4). Thus, blowdown should be no greater than necessary to maintain the level of total disolved solids (TDS) in the water below the limit recommended by the boiler manufacturer. In modern boilers this is usually between 2500 and 4500 parts per million. Losses for various boiler pressures and percentage blowdown are shown in Fig.2-2 (27).

Blowdown is much reduced when a high proportion of steam condensate is returned to the boiler. If this is not possible, expenditure on flash steam or other means of heat recovery from blowdown may be justified. Heat recovered is often conveniently used to preheat feedwater,

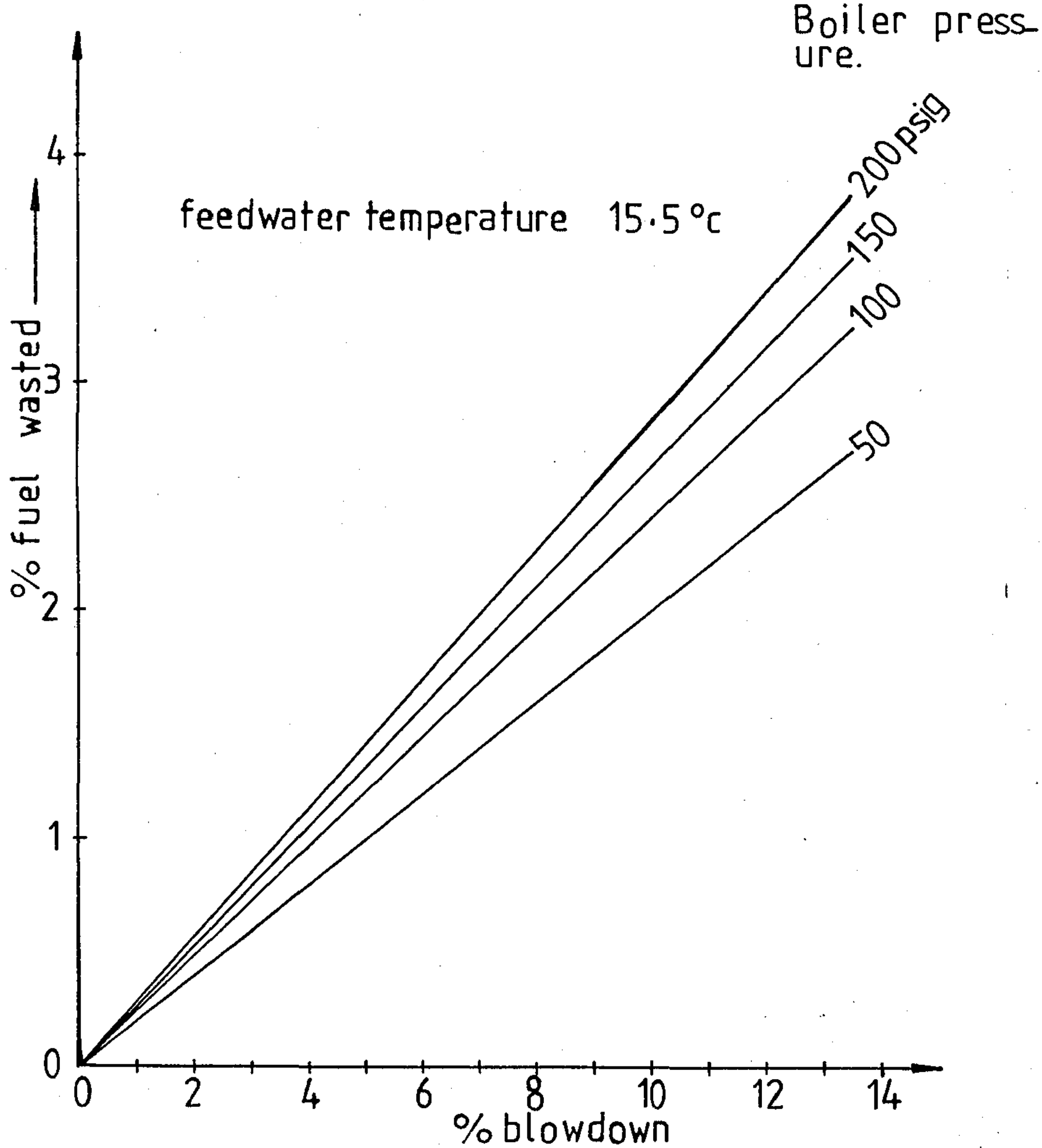


Fig. 2-2 : Heat loss due to blowdown (27).

thereby helping to maintain the rated output of the boiler. Heat recovery also reduces the temperature of water that must ultimately be discharged from the plant, so that the limits of river and water authorities can more readily be met (14, 27).

Residue heat losses

Further heat losses arrises from purging the boiler with cold air when they are started up and shutdown. Losses also occur from cold air flow due to natural buoyancy through off load boilers especially when they are connected to common flues and chimneys (34, 43). Thus purging should be no longer than necessary to fully meet safety requirements i.e. to completely clear flues and chimneys of all traces of flamable gases before ignition and at the end of the firing cycle.

2.1.1.3 Economisers

An economiser is a heat exchanger located in the flue gas stream of a boiler to improve overall efficiency by preheating water for boiler feed or for a separate process application (66). The economisers have a limited scope of application in modern boilers with efficiencies nearer to 82% and with flue gases leaving the boiler at about 220 ^{o}C, there is a danger of low - temperature corrosion occuring (particularly in boilers fired with residual oil fuel) if further heat is extracted (89).

Economisers, however, are viable on modern boilers fired with natural gas where corrosion problems are minimal.

2.1.1.4 Fireside Deposits and Corrosion

Heat transfer in boilers is reduced by the accumulation of deposits on the "fire" side of furnace and tubes. Such deposits normally consists of unburned fuel (carbon), ash, sulphates, oxides and other corrosion products. The maintenance of high heat transfer rates and high thermal efficiency is ensured by minimising the formation of deposits and removing deposits that have formed.

Carbon deposits are generally associated with worn or damaged atomisers and air guide vanes, insufficient air supply to the firing equipment or to the boiler house, overfiring causing flame impingement on relatively cold surfaces and cold air infiltration causing local chilling. Air infiltration is often a fundamental cause, of low temperature corrosion as well as low flue gas temperature caused by low-load operation. Also operating boilers at below their designed working pressures, tends to cause low-temperature corrosion and deposition (89, 93).

High temperature corrosion is usually associated with water tube boilers, particularly superheaters and reheater tubes, (97). It is generally accepted that a 10% reduction in heat transfer occurs in boilers for a 1 mm thickness of soot or scale (58).

Boiler cleaning, however, is very expensive, thus a balance has to be struck between the loss in efficiency of a fouled boiler and the cost and incovenience of over - frequent cleaning. In practice this is done by monitoring the boiler efficiency and setting a limit on flue gas temperature, at full load, at which cleaning is carried out (34, 43).

2.1.1.5 Water Treatment

The scale formation on the water-side of any heat exchanger, reduces the heat transfer rate, and when high heat fluxes are involved there is a danger of the metal temperature rising above its safe working limit. Correct water treatment is essential to prevent this and other undesirable effects such as corrosion, priming and carryover and to ensure steam quality (5, 34, 43).

Corrosion

Oxygen is the main cause of corrosion in boilers and associated pipes and equipment (93). De-aeration of the feed-water by preheating is the cheapest method. Sodium sulphite or hydrazine, which is toxic, is normally used to remove residual oxygen after mechanical de-aeration or to deal with the entire oxygen content where a de-aerator is not available. Treatment in the boiler is needed to remove the final traces of oxygen. Carbon dioxide in the condensate may cause corrosion, the severity increasing as its temperature falls. Neutralising or filming amines could be used to prevent this although their toxicity may preclude their use in food industries. Corrosion throughout boiler, steam and hot water systems is minimised by the control of acidity, which ideally should be in the range of PH 8.5 to 9.0 (75).

Foaming

Foaming arises in boilers from the presence of traces of such foam formers as soaps, certain compounded lubricating oils and detergents. Due to foaming the water in the boiler becomes unstable, consequently:

(a) Float switches may become ineffective

(b) Gauge glasses may become difficult to read and

(c) Wet steam may result causing serious problems in pipes and equipment.

Foaming may be suppressed by controlling the total dissolved solids to below about 200 ppm and avoiding sharp changes in steam demand (5, 75). Foam formation is sometimes encouraged by low-pressure boiler operation, thus boilers should always be operated at their correct working pressure with reducing valves being used where lower pressure steam is needed.

2.1.2 Steam Distribution and Utilisation

All hot surfaces and pipework, including flanges valves etc. should have some thermal insulation properly secured and protected against damange and deterioration. Redundant sections should be effectively isolated, or better still removed completely. Leakage can be a source of substantial heat loss. Steam leaks past traps, pressure relief valves and drain valves or via by-pass valves left open accidentally are not likely to be noticed. Warm valve bodies and down stream pipework are useful indications that something is amiss (34, 43, 75, 89).

Many processes require steam at around 2 bar gauge. However, it is widely accepted that it is poor practice to operate a heavily loaded boiler at below 7 bar if carry over and instability are to be avoided. When a range of pressures is needed then local pressure reduction is likely to be most suitable. If a number of units are operated at identical pressure then distribution at that pressure may be feasible (43, 89). All pressure reduction valves should be preceded by adequate means to drain condensate and must be followed by a pressure relief valve. Distribution at high pressure requires small mains, so that despite the surface temperatures being high, the heat loss is relatively less because of the smaller heat transfer area.

Whenever the flow is substantial, and a large pressure reduction is required the generation of work should be considered, by driving a back pressure steam turbine to produce electricity, compressed air, or mechanical power.

2.1.2.1. Steam Pressure and Process Temperature Control

Steam pressure reducing valves enable heat transfer to be carried out at optimum temperatures and they protect plant by providing pressure within safe working limits. Valves should not be oversized for the maximum duty required and they should be preceded by a steam separator, with drain trap, and fine - mesh - screen strainer to remove dirty and scale (58).

Isolating valves and adjacent pipework should be adequately sized, so that all pressure reduction is effected by the pressure-reducing valve. To protect downstream plant or process from excessive pressure or temperature a correctly - sized relief valve should be fitted (93). Thus, having steam available at the point of use at the correct temperature is ensured by controlling steam pressure.

Steam traps and air vents

The purpose of a steam trap is to discharge steam condensate as well as air and other non-condensables, whilst retaining steam in the system. Ideally it should do this over a wide range of condensate flow rates and with minimum of maintenance (87). There are three types of steam traps, viz: (87, 93)

(a) Mechanical traps: These incorporate an inverted bucket or float, which use the buoyancy provided by condensate in the chamber to open the outlet valve, which closes when condensate is cleared and buoyancy is lost.

(b) Thermodynamic or impulse traps : these close when condensate (which is released periodically) flashes to steam, this produces low pressure under a disc, which seats, but then lifts (to repeat the cycle) when pressure above it declines, as the flash steam condenses.

(c) Thermostatic traps: these respond to the difference in temperature between steam and condensate and use a bimetallic element or bellows to open the outlet valve to release condensate when its temperature has fallen.

The traps should always be sized according to the volume of the condensate that they will be required to handle (allowing for any back pressure) and not the size of the pipes into which they will be incorporated. They should be fitted close to the equipment that they serve and below natural drain points. By - passes round traps may be essential in some instances for purging the system after shutdown or to aid maintenance. They should be closed during normal plant operation (5, 34, 87).

Steam losses through thermodynamic traps depend upon their opening frequences, which in turn depend upon ambient temperatures. The thermostatic and ball float traps cannot pass steam when operating correctly, but the latter are bulky and convection and radiation losses from them are correspondingly high.

As a suitable substitute for steam traps orifice plates are sometimes used, sized to handle the condensate load. They are simple to maintain and convection and radiation losses from them are low. Because of the wide difference in density between condensate and steam they cannot waste any great amount of steam, however, the diameter of the hole of the orifice plate is the main limitation. A 1mm diameter hole can still waste over 2 kg/hr of steam at "no load" conditions and may be inadequate to cope with air at start up (87).

Air present in steam lowers the temperature of the mixture to below the saturated temperature for the steam alone. In addition air films on heat transfer surface, dramatically reduces heat transfer rates and the presence of air may also aggravate internal corrosion. Thus air removal is quite essential. Air vents near steam traps could solve the problem.

2.1.2.2. Flash Steam and Condensate Recovery

When hot water, under pressure is released to a lower pressure, flash steam is produced by the heat made available by the reduction in the specific enthalpy of the water, the amount depending upon the pressure difference (14, 27, 58).

It is not easy to utilise the heat that will be lost in this way because the flash steam has to be collected and then used in an appropriate application, preferably one that is not temperature sensitive. Preheating is an obvious possibility with addition of a lower pressure stage, to which flash steam can be applied, to precede the main heating stage Alternatively steam may be condensed by a fine cold water spray, for use as warm water.

Recovered steam condensate is a valuable source of heat and pure water, provided it remains uncontaminated. An aditional advantage is that, because it raises the temperature of the boiler feed water, it increases the effective output of the boiler. A 5 ^{o}C increase in the boiler feed water temperature is equivalent to an energy saving of about 1% (27).

The relationship between fuel saved and condensate recovery is shown in Fig 2 - 3 (27).

2.2 Energy Management

Implementation of an energy saving programme in an industrial environment is not always free from difficulties, some being due to ingrained customary attitudes and others to purely technical problems. Thus, developing and maintaining a systematic approach to energy conservation requires some sort of management, and the most important ingredients in the effective management of energy are a genuine commitment at the top level and a capability within the organisation to take actions rather than to talk about it (42, 43, 53, 85, 89, 97).

The most common first step in energy management programme is to appoint an Energy Manager, whose formal qualifications are not particularly important. An enquiring mind, the tenacity to get to the root causes of waste and the ability to generate and sustain enthusiasm at all levels are the main requirements. However, the larger and more complex the uses of energy, the more likely it is that the energy manager will benefit from an engineering or scientific training, that will enable him to appreciate fully the significance of energy conversion and utilisation processes.

2.2.1 Energy Management Programme, Motivation and Training

Generation of interest at all levels is essential, bearing in mind that people have short memories, thus, sustaining their interest needs continued effort and imagination. A better perspective of the task ahead is obtained when some basic data on energy supply and use have been gathered and analysed (85, 97, 99). A continued energy audit is usually desirable to monitor consumption and cost trends and to provide hard facts on which future planning can be based. Analysis should be made to differentiate between simple "good housekeeping" measures and those that require new methods and equipment. Improvements that require capital expenditure should meet whatever pay-back and cash flow criteria normally apply within the organisation using realistic estimates of future energy costs, and resultant savings should be assessed (41, 42, 43).

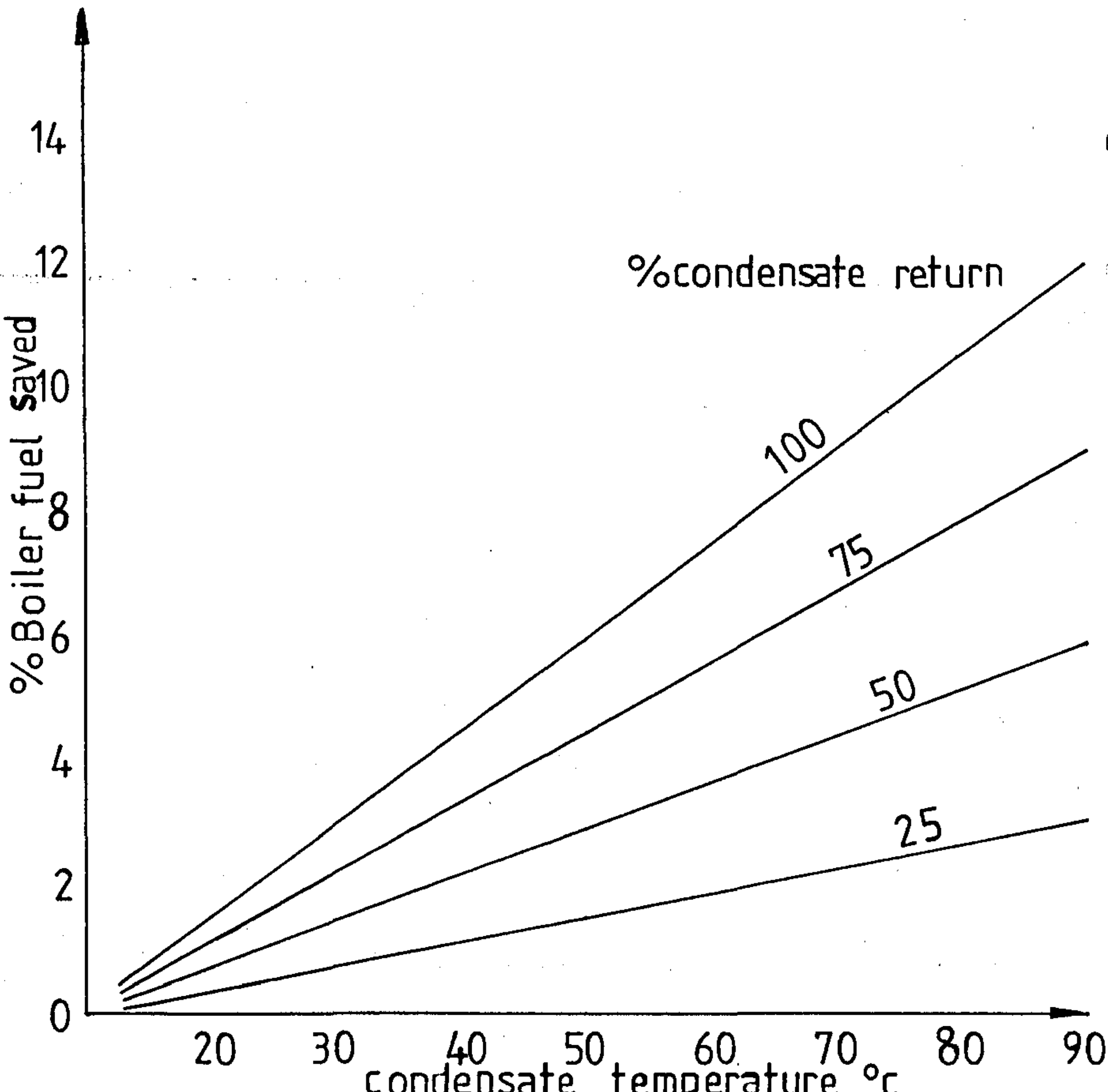


Fig.2-3: Fuel saved by condensate return (27).

In many organisations, a shortfall in energy can have serious consequences and contingency plans, low-energy strategies, should feature in all energy management programmes. Priorities should be determined, emergency work plans devised and stand by generation and fuel stock requirements assessed (84, 85). Health, safety and environmental requirements should be carefully monitored. Closer Control and better maintenance of plant in order to effect energy savings however, usually raise safety standards (42, 43, 89). Changes that directly affect employees in any way should be approached with care (99).

The schemes must be given adequate promotion on site and, the objective must be clearly stated so that every individual is made aware of the value of his contribution. Suggestions for improving energy consumption could be obtained or encouraged through existing suggestion schemes. Maximum publicity should be given to this activity in house magazines, works committees and posters, and the results achieved must also be communicated (27, 28, 29, 99).

For those actually in charge of energy plant, especially boilers, specific training should be given. A most useful training opportunity occurs during commissioning of new plant, and whenever possible operators should be present. Subsequent training should be given to new operators. Training should be repeated periodically to update operators knowledge, especially when new equipment or techniques are introduced (43, 89, 97).

2.3 Energy Auditing

The "Energy audit" refers to an initial, often one-off, analysis of the pattern of energy usage within a works, identifying and quantifying the main areas of operation and types of energy usage. Sometimes energy audit is used interchangebly with energy monitoring, but energy monitoring is a complementary aspect of energy auditing and it refers to the regular systematic measurement of energy usage in relation to targets based on production level or prevailing weather conditions (27, 35, 89, 97, 108).

Every energy audit is particular to the organisation in which it is carried out and whose overall strategy it should assist (35). A few basic rules may be applied to all audits but there are certainly no common standard forms nor should there ever be. Thus, as a matter of priority the energy audit procedures will be related to the energy consumption of a building services. However, the principles of conducting the energy audit to all energy users, with only common factors for comparative consumption and the methods of data collection being different for individual cases.

The way to prevent the energy audit from appearing a formidable and daunting prospect is to take a small step at a time. Thus, energy audit should, be brocken into phases that are concerned with:

(a) Records.

(b) System and equipment.

(c) Building data.

2.3.1. Records

These are the basis of the audit and comprises

(a) Fuel bills : If possible fuel bills covering three years, to provide an overall consumption and the cost of energy during this period. Common units such as therms, kWh, or GJ should be used so that individual energy consumption can be compared with the individual costs. It is necessary at this stage to organise a system to ensure that the necessary information is collated on an on-going basis for future updating of the audit.

(b) Drawings: These gives information about the structure of the building and the services within it. The extent of manufacturing facilities and production rate should be indicated together with the services and utilities required by each item of manufacturing equipment.

(c) Modifications : Even if the necessary drawings are available, these may not be up to date. Over a period of time, buildings may have been modified or new equipment may have been installed. Equally, legislation may have necessitated modifications to the building or systems. It is not unusual to find an absence of records for such modifications, so that the auditor may have to rely on visual inspection or verbal information.

(d) Meter readings: These fall into three main categories; overall consumption, Departmental consumption and plant consumption. Whilst main supply information would normally be available, departmental and plant consumption may not. In such cases it may be necessary to estimate the costs initially and also to carry out plant tests to arrive at information related to a specific category of use.

2.3.2 Building Data

(a) Construction: This is important for determining the thermal transmittance of the structure. This should include type and thickness of the material of construction and finishes which have been applied.

(b) Usage: The use of the building may have a major bearing on the energy requirement of the building, and will be determined by the environment required, process requirements and the services and utilities required to serve that purpose.

(c) Occupation periods: These will have to be taken into account for the normal operation, extended periods of operation due to overtime, cleaning and security requirements.

2.3.3 System and Equipment

Data should be collected either on a long term basis to provide average values during the extended period, or on individual short term tests with correction factors for run time, to be followed by the more accurate long term tests. Both data collection exercises consists of:-

(a) Listing of:- All utilities.

(i) Systems (e.g. heating, refrigeration, ventilation systems) and each systems operation, use and design parameters.

(ii) Energy consuming equipment within each system (e.g. heating coils, compressors, ventilation fans). Again each equipment will have its design parameters and operation to be investigated and recorded.

(b) Function :- The detail of each energy consuming equipment must include the function of that equipment within the system and its importance in the overall performance and/or manufacturing facility.

Consumption :. The detail consumption of the individual equipment will be determined by either short or long term plant tests and/or metering on a long term basis. A flow chart for conducting the systematic energy auditing is shown in Fig. 2 - 4.

The next step is to analyse the results which should now be in a form in which individual plant and system consumption for each building area can be quickly compared on a common basis. Heating systems, for example, could be evaluated using monthly or weekly degree days and for processing industries these can be compared with the production rate.

Carrying out an energy audit may reveal a number of immediate and easy conservation measures which themselves can justify the effort, even without the possibility of using the results as a basis for more extensive conservation measures.

2.4 Combined Heat and Power Generation

Combined heat and power generation systems are well established in the chemical and process industries and represent major energy conservation measures. Around 30% of electricity used by industry in the U.K. is self generated (39), and if allowance is made for direct drives, self generated power in the U.K. probably represents 40 - 50% of the total power requirements. With most combined heat and power systems, the power is generated with a marginal efficiency of at least 75% compared with the efficiency of around 30% characteristic of electricity from the public supply industry (11, 32, 33, 34, 43). The power can be in the form of direct mechanical drives or in the form of electricity generated with turbo - alternators. The heat is usually in the form of low pressure or intermediate pressure steam (14, 31). The choice of the plant is determined by the process heat/power ratio. Table 2 - 1 indicates the range of heat/power ratios used in different types of generating plants.

Fig. 2-4 : Flow chart for energy audit (35).

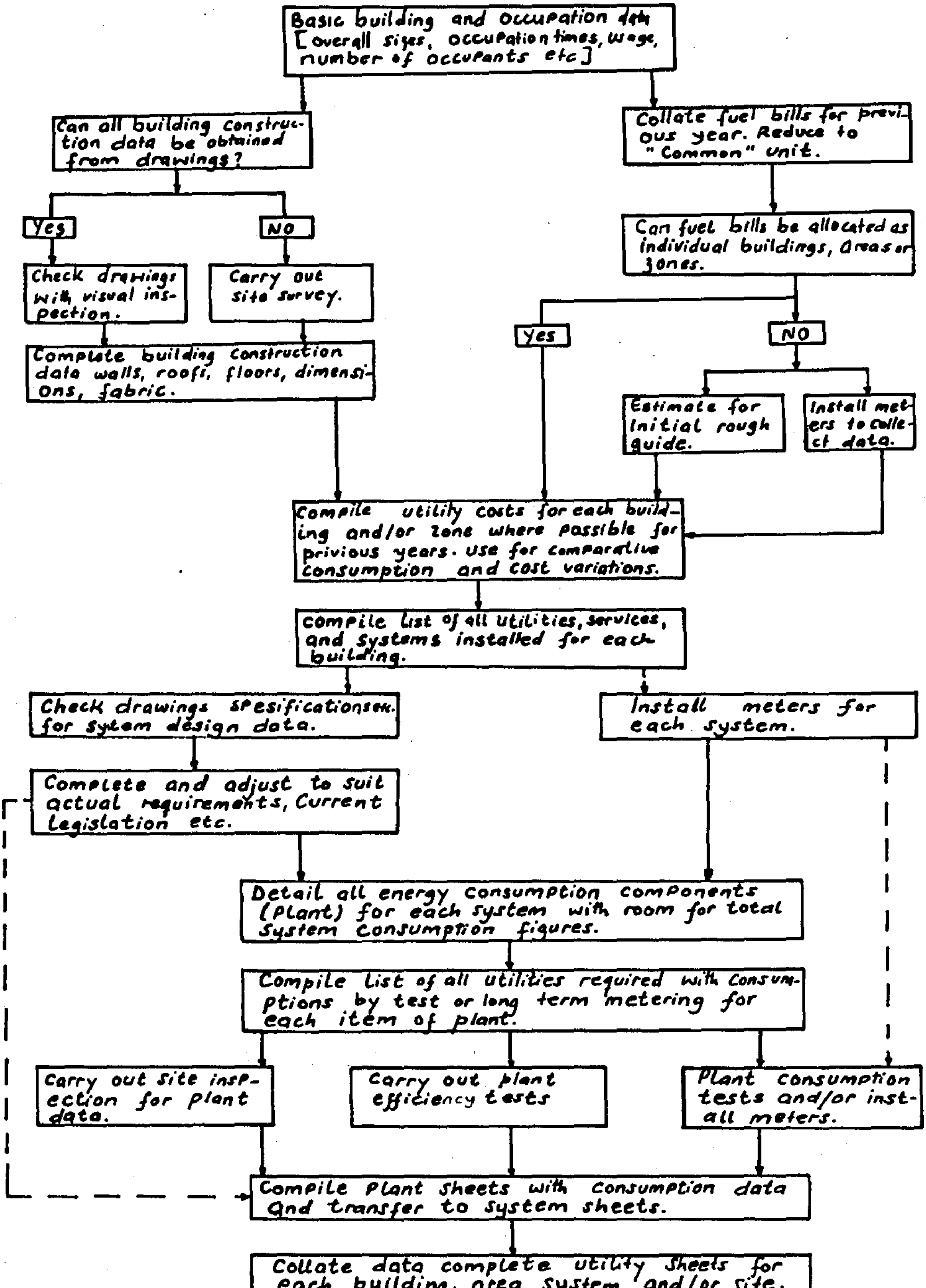


Table 2 - 1 : Types of plant for combined heat and power systems.

Prime mover	Heat/Power ratio	
Diesel engine	Up to 1 : 1	
Gas turbine	Up to 3 : 1 without supplimentary firing	
	Up to 15 : 1 With supplementary firing	
Steam turbine	Up to 15 : 1 Typically	
	high heat/power	10 : 1
	low heat/power	5 : 1

Diesel engines are not significant in the chemical industry as a whole and most of the current systems are based on back pressure steam turbines.

The back pressure turbine takes steam from the boiler and produces power by expanding the steam through its stages, before finally exhausting the whole of the steam flow to the heating process (31). The supply of process steam is thus taken from a point in the system after final stage of expansion. The schematic arrangement of the back pressure steam turbine is shown in fig. 2 - 5.

The amount of power generated by the steam is usually determined by the demand of heat. Most combined heat and power systems are connected for parallel operation with public supply, see Fig. 2 - 6. This provides security of supply and helps in balancing the system through the transfer of electricity in either direction, provided the local electricity authority is agreeable. Exact balancing of loads at the optimum design value for the system is usually not possible, nor is it essential since most systems have some degree of flexibility (31, 43).

Present commercial considerations favour the minimum transfer of electricity to or from the public supply so that this imbalance may be dealt with by meeting the power requirements as far as possible by utilising the maximum available generation capacity and/or by venting excess steam.

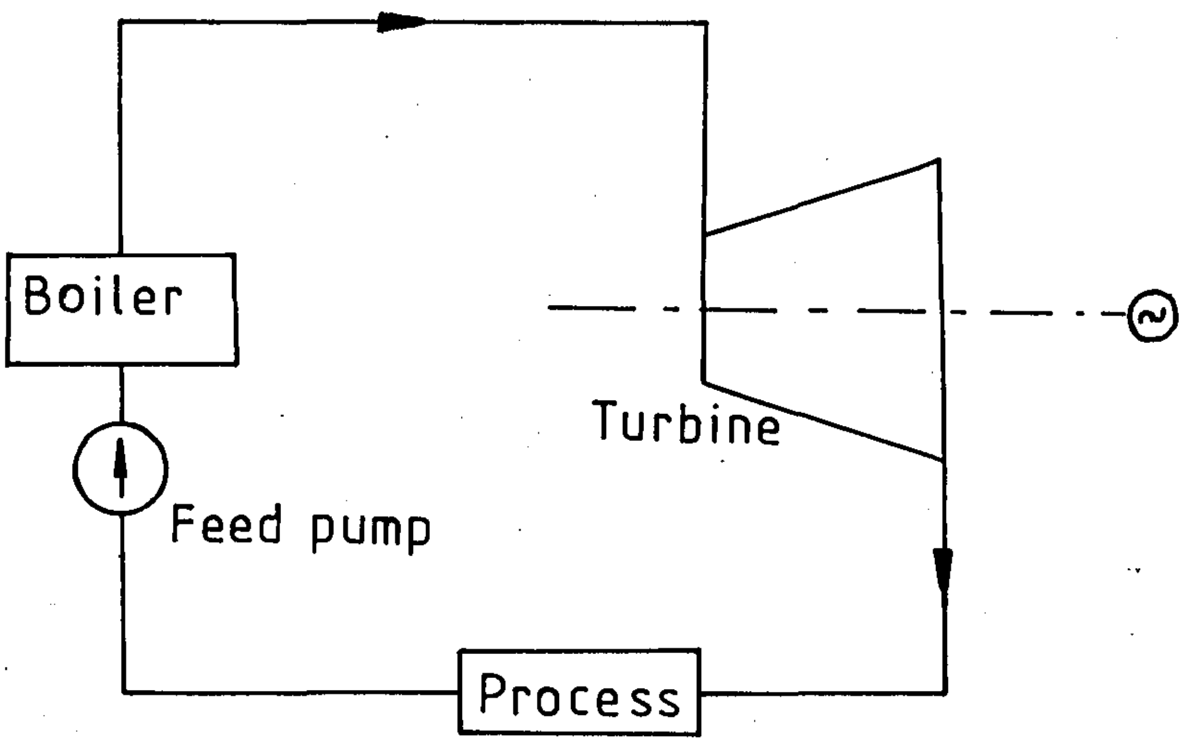


Fig. 2-5: Schematic arrangement of pure back-pressure steam turbine (31).

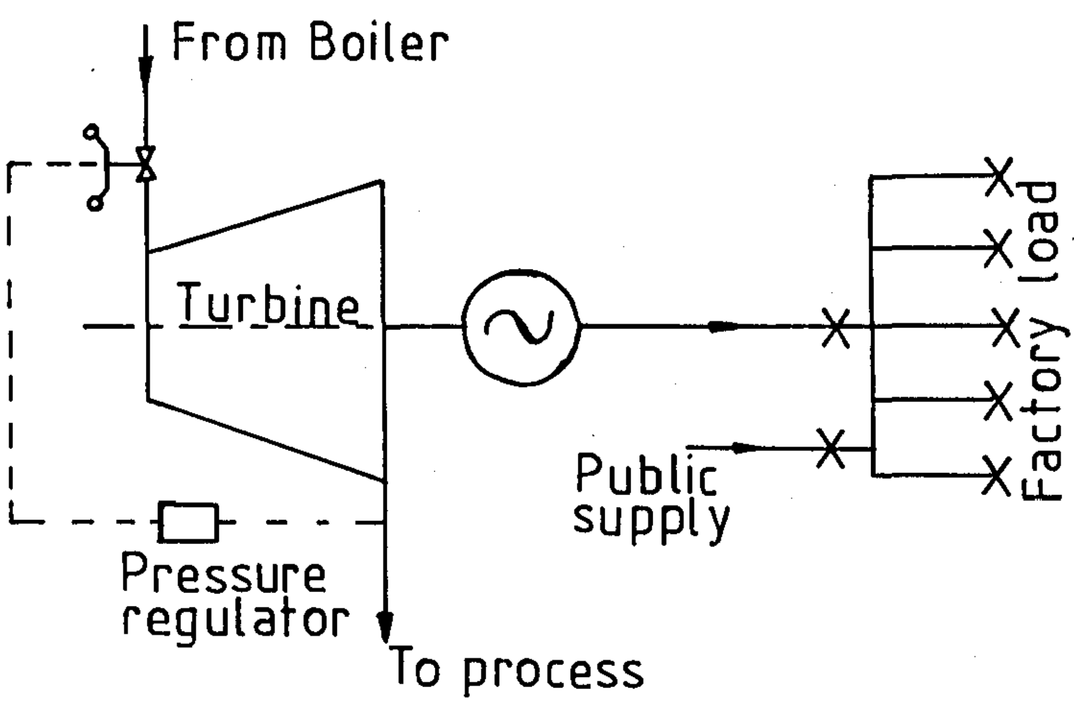


Fig. 2-6: Simple back-pressure turbine in parallel with public supply (31).

This may be economically justifiable but it is wasteful of energy, conspicuously so, if low pressure steam is being vented. A better approach is to redress the balance by finding a use for the surplus low-pressure steam, for instance by installing another high pressure steam using process. While this approach will be obvious to companies in this position it may not be easy or feasible in practice. Thus, there is no easy solution to a large imbalance, and forecasting the heat and power ratio accurately at the design stage is therefore very important.

2.5 Good Housekeeping

Information and check list describing good housekeeping measures are readily available in, for instance references 27, 28, 30, 43, 97, 110. A list of measures compiled from these sources, with the exclusion of those concerned with boilers and steam systems which have been covered in previous sections, are given in this subsection.

The measures generally represent what ought to be good practice and do not involve any particularly novel or sophisticated technology. However, their implementation requires significant investment of time and manpower. Moreover, most of the measures are not once-and - for - all, but require regular attention if energy savings are not to be erroded.

2.5.1 Good Housekeeping Check List.

(a) Insulation

(i) Check optimum thickness, increase if necessary.

(ii) Ensure flanges and manholes are lagged.

(iii) Improve protection of insulation from rain and water.

(iv) Replace insulation after maintenance work.

(v) Consider use of infrared surveys to indentify major heat losses.

(b) Pumping and compression

(i) Ensure pumps have correct characteristic and impeller size.

(ii) Consider replacement of undersize lines and valves.

(iii) Consider removal of discharge line restrictions.

(iv) Consider whether flows can be reduced (reduced cycles).

(v) Switch off when not required.

(c) Electrical systems

(i) Improve power factor and minimise other losses.

(d) Compressed air system

(i) Use minimum operating pressure.

(ii) Reduce leaks.

(iii) Reduce pressure drop in pipeworks.

(iv) Maintain minimum inlet air temperature at compressor.

(e) Nitrogen system

(i) Reduce leaks.

(ii) Reduce pressure drop in pipeworks.

(iii) Check whether any other inert process gas could be used instead.

(f) Cooling water system

(i) Minimise unecessary flow of cooling water.

(ii) Investigate scope for "cascade" cooling system i.e. running cooling water in series through a number of units e.g. condensers operating at different temperatures.

(g) Vacuum systems

(i) Reduce leaks.

(ii) Consider replacing steam ejectors by mechanical vacuum pumps of the liquid ring type which require much less energy for operation than steam ejectors.

(h) Space heating and illumination

(i) Provide adequate control of heating and ventilation, reset all heating in offices and laboratories to the minimum commensurate with reasonable comfort.

(ii) Provide self closing doors for personel and vehicular traffic.

(iii) Use waste heat from process streams, whenever possible for space heating.

(iv) Where the uniform heating of a given area is not necessary use selective local heating.

(v) Removal of light diffusers can reduce the number of lamps required by up to 20% for the same illuminance. Clean lamps regularly.

(i) Instrumentation and Control

This is an essential aspect of most of the previous measures and is therefore only listed as a separate measure for completeness. However, it is an expensive measure which should be handled with great care. For example, Fig. 2 - 7 show a typical piping and instrumentation diagram for a simple evaporator unit (this is one possible solution of achieving of a system reliability). It can be seen that a lot of instruments and control loops are involved. It will obviously be prohibitive expensive to have all equipments, in the plant, at such control level.

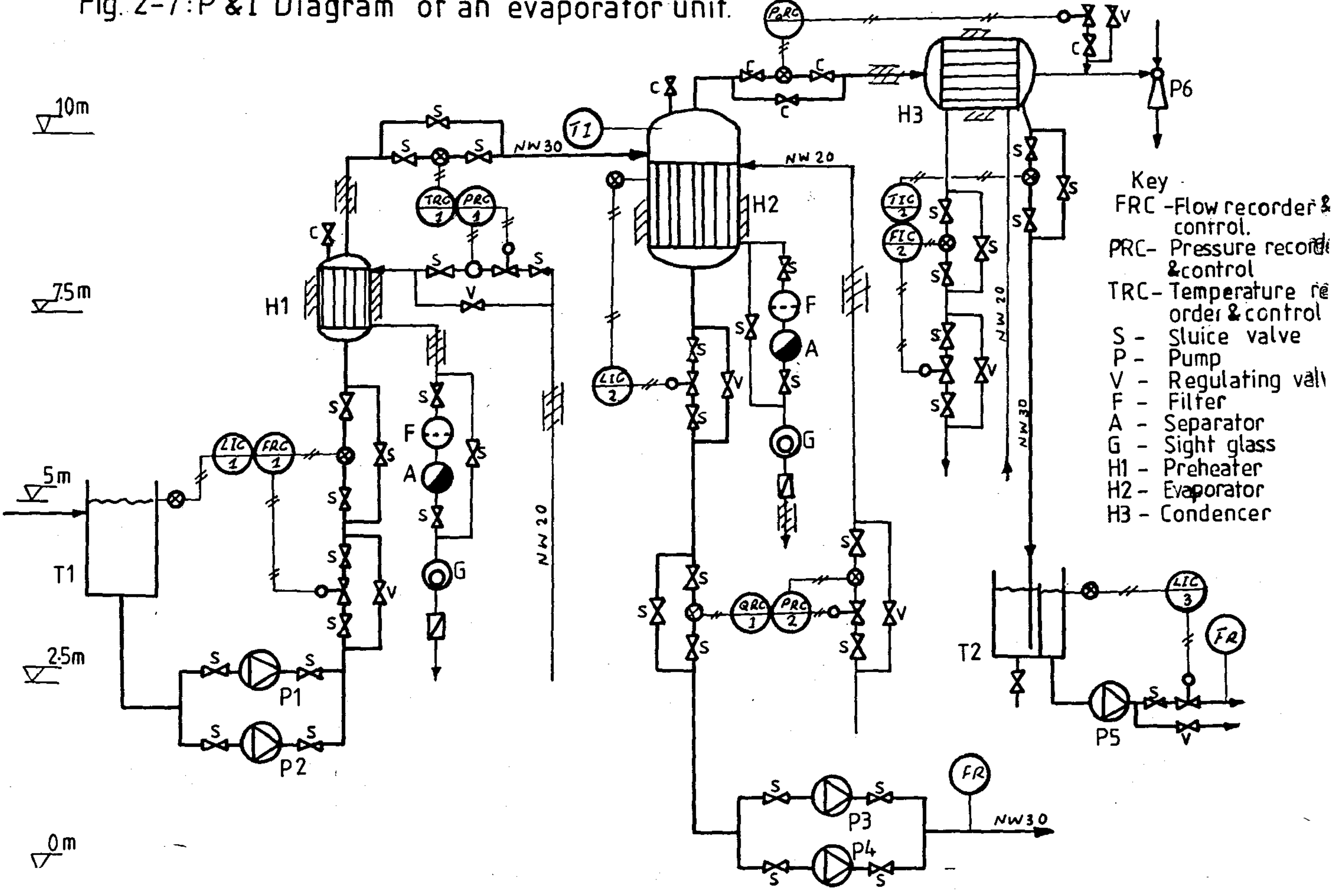
Thus expenditure on good-quality, permanent instrumentation and controls can be justified on large items of energy using plant, especially steam boilers and high temperature furnaces. For smaller applications, for spot checks and investigation and in situations where fixed instruments would be vulnerable to damage, portable test equipment and easily fitted meters should be used.

An ambitious, overall energy saving of up to about 10% has been claimed to be possible through good housekeeping measures alone, but the potential will be less than that for a reasonably efficient works (43, 89, 97). The key to maintaining interest, effort and commitment to good - housekeeping measures at all levels, appears to lie in developing comprehesive and reliable energy monitoring and measuring system so that wastage is clearly recognisable.

2.6 Heat Recovery

Heat recovery or re-use is one of the most important energy conservation measures in the Chemical and Process industries. The actual economic advantage from any heat recovery depends upon the availability and cost of fuels. Obviously, savings from heat recovery increase as fuel costs rise. The cost of fuel saved must be compared to the capital investment,

Fig. 2-7: P &I Diagram of an evaporator unit.



amortization, maintenance, operating costs and any other cost factor involved in owning and operating heat recovering equipment (10, 16, 41, 48, 57, 89, 110).

The concepts of heat recovery using waste heat boilers, recuperators and regenerators are well established and are not discussed here. Thus only the engineering principles governing heat recovery and scope of application of heat recovery are considered.

2.6.1 Engineering Principles

Decisions to recover energy are governed by the second law of thermodynamics, which states that, for an ideal reversible process, entropy does not change, but for real irreversible process it increases (93). Practically this means that, all energy degrades so that as soon as fuel burns, its ability to perform work is partly destroyed. Stating the second law differently "only a fraction of heat at certain temperature can be converted into work; the rest must be dissipated at ambient temperature as waste heat". Furthermore, "The carnot Cycle, Fig.2 - 8 is the cycle that gives the highest possible conversion rate of heat into work" The relationship between work and heat is given by:

$$W = \frac{T_1 - T_0}{T_1} \cdot Q \quad . \; . \; . \; . \quad 2 - 1$$

Where: W = theoretical maximum work, kW

Q = heat quantity, kW

T_1 = heat source temperature, K

T_o = heat sink temperature, K

The ratio $\frac{T_o}{T_1} \cdot Q$ is the heat that must be discharged to the atmosphere.

and $\frac{T_1 - T_o}{T_1}$ is the efficiency, η, of conversion.

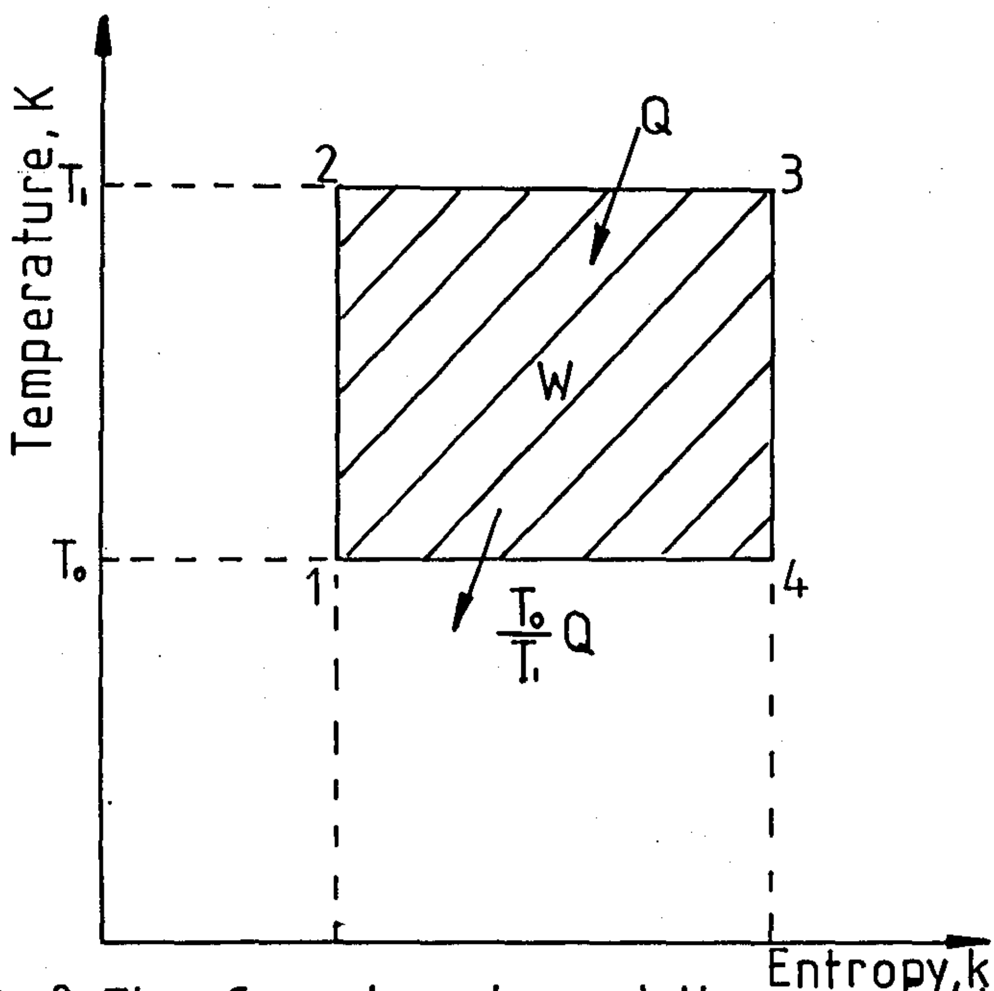


Fig. 2-8: The Carnot cycle and the conversion of heat into work (93).

From this foundation the following engineering principles can be deduced (41).

(a) The efficiency of extracting work from fuels approaches one so long as they have not been burned. But when they are burned this availability is partly degraded. Also all real processes are irreversible due to frictional losses, inefficiencies of real cycles and temperature drops in heat exchangers which all add to the quick degradation of energy.

(b) Heat has its specific value, closely related to the temperature level. The higher the temperature the higher the efficiency of the heat. Thus, it pays off more to recover heat at higher temperature.

(c) Work is the more valuable form of energy it is easily convertible into heat, at least theoretically, whereas heat is only partially convertible into work. This means that it is more valuable to recover work than to recover heat.

(d) Irreversibility should be avoided as much as possible. Thus, recovery by sequential use of energy is important. As many stages as possible should be built into the heat recovery, with each step going down gradually in temperature. Recovered energy should be used to preheat material streams, to generate steam for driving process gas compressors etc.

(e) Low - level energy should be used to create low - level heat with high-level energy reserved to create work or high-level heat.

2.6.2 Recovery of Heat

Recovery of heat is governed by the amount of heat available and its temperature level. Good thermodynamic practice dictates recovery of heat at the highest temperature to keep the efficiency as high as possible. However, there are limitations to this ideal; the process has to work and yield maximum profit. Thus a balance has to be struck between maximum heat recovery at a suitable temperature level and minimum raw material usage, the overall operating cost should be at a minimum. Also the need for recovered energy has to be determined, since feasibility depends upon the ability to use the energy.

Heat recovery can be divided into three possible applications based on the temperature range of the source (3, 14, 41, 57, 110), viz:

(a) Full temperature range applications

Feed-effluent exchangers can be used across the entire range of temperatures. For example, in the high temperature region(i.e.500 to 1000°C), exchange may be employed for preheating cracking and reformer furnace stream. In the medium temperature range (i.e. 300 to 500°C), preheating, reactor feeds, air for reaction purposes and so on. While in the lower temperature range (i.e. $<300^{\circ}$C), exchange may be employed for preheating distillation column feed, dryers and reactors.

(b) High-level heat recovery

Waste heat boilers can be used to recover heat from process or furnace flue gases to generate high-pressure steam. Typical examples are ethylene cracking units.

(c) Low level heat recovery. This can include all applications where it is impossible to develop superheated steam or generate shaft power. As soon as the temperature level of the heat source falls below 400 to 500°C, the efficiency of the steam cycle drops sharply and makes it an undesirable alternative. However, other alternatives for its use are possible, viz:

- Generation of low or medium- pressure steam from waste heat.
- Recovery of flash steam and steam condensate.
- Recovery of boiler blowdown heat for preheating boiler feed water.
- Exchange between process streams.
- Multiple effect evaporation, for instance in distillation (57).

Absorption refrigeration can be a very good alternative to mechanical refrigeration, especially if the electricity price for driving the refrigeration compressor is very high and if waste heat is available for the absorption system (15). The combination of a process stream that needs to be cooled from about 200 to 100 ^{o}C and refrigeration at about - 10 to -20 ^{o}C, for example, is very attractive for this approach (41).

An interesting application would be the reduction in gas compressor power requirements by lowering the compressor inlet temperature in an absorption system driven by waste heat. Low - level heat can also be used in a Rankine cycle with an organic fluid, Fig. 2 - 9. Some specific organic products e.g. CCl_2F_2, CH_2Cl_2, C_6F_6 have more favourable thermodynamic properties than steam in the low temperature range (3, 6, 15), as steam exhibits an intrinsic thermodynamic inability to extract heat from the available source in these temperature ranges.

Rankine cycles deserve more attention for application where large amounts of low level heat are available which cannot be used for other more conservative recovery techniques, and when electricity costs are high (3, 4, 41, 43).

2.6.3 Heat Pump Applications

A heat pump reverses the natural flow of heat from higher to lower temperatures. Ideally, a quantity of heat Q_1 is transferred from a low-temperature source to a fluid, an amount of work, W is done on the fluid, and a total amount of heat, $Q_1 + W$ is transferred to a high temperature sink. The relationship between the heat rejected and the work is given by:

$$\frac{Q_2}{W} = \frac{T_2}{T_2 - T_1} \qquad \ldots\ldots \qquad 2-2$$

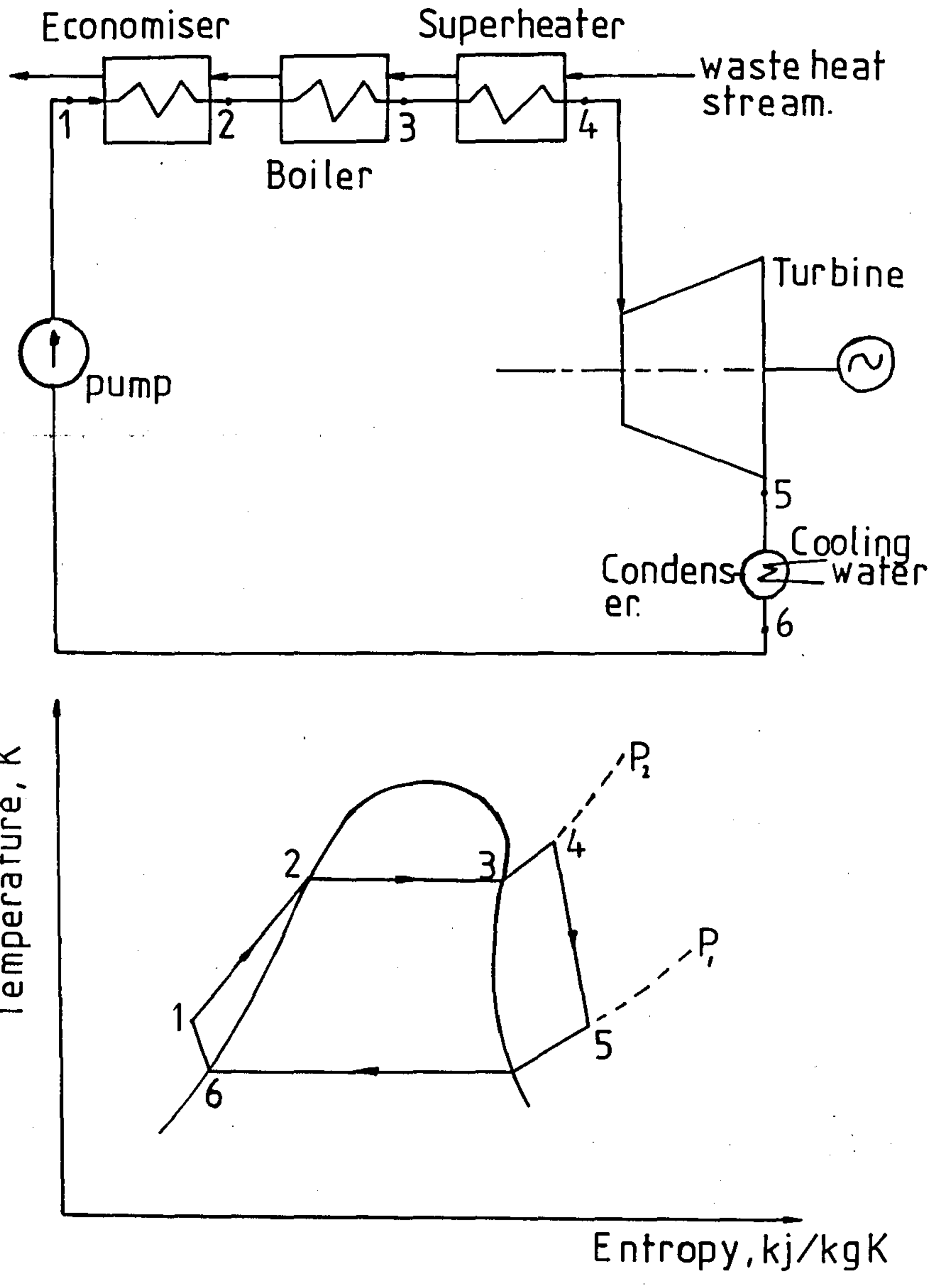


Fig. 2-9: Rankine cycle schematic representation of the installation and thermodynamic diagram (31).

Where: $Q_2 = Q_1 + W$

T_2 = Working fluid condenser temperature, K

T_1 = Working fluid evaporator temperature, K

Equation 2 - 2 is termed as coefficient of performance (C O P).

The ideal heat pump cycle is represented by a reversed heat engine operating as the carnot cycle, as shown in Fig. 2 - 10. The carnot cycle cannot be realised in practice and the operation of real heat pumps is better represented by Rankine cycle Fig. 2 - 11. In actual plant , in expanding the working fluid, the work is dissipated as mechanical friction and an expansion valve is used, resulting in the stage 3 - 4 lying on a constant enthalpy line. While no work is now obtained from the expansion process, this does not affect the heat extracted from the condenser, but reduces that taken up in the evaporator. The expansion valve makes the process irreversible. After expansion, the working fluid is in the form of a saturated liquid, which then picks up heat in the evaporator ideally being boiled off to saturated vapour, absorbing the heat at constant temperature (6, 14, 31, 57, 93).

A practical heat pump cycle differs from the reversed carnot cycle in two other respects, viz:

(a) The compression is normally carried out in the superheat region, with desuperheating occuring in the condenser.

(b) Some additional liquid cooling (subcooling)occurs between the condenser exit and the expansion valve, prior to the adiabatic expansion between points 3 and 4, whence the cycle is repeated.

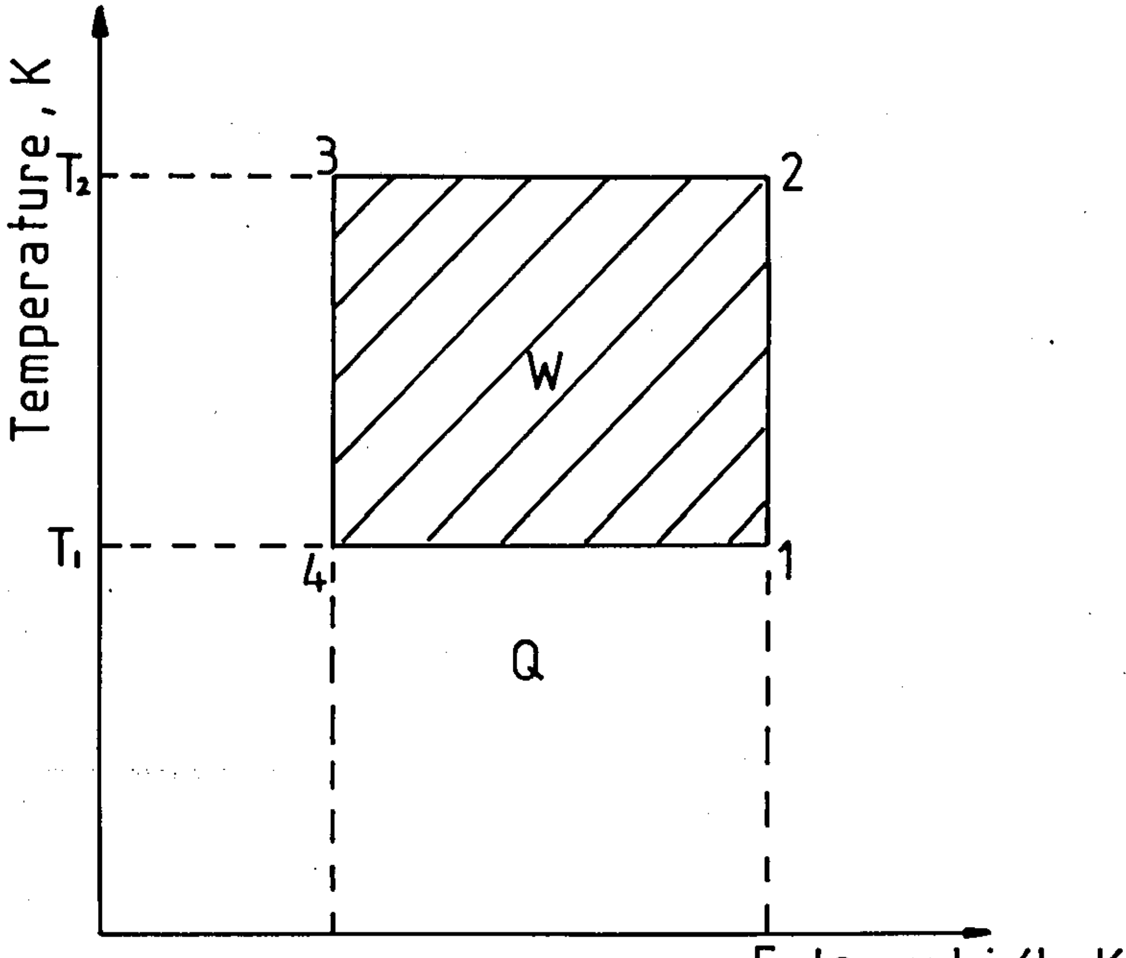


Fig. 2-10: Carnot cycle heat pump - thermodynamic reprensentation (93).

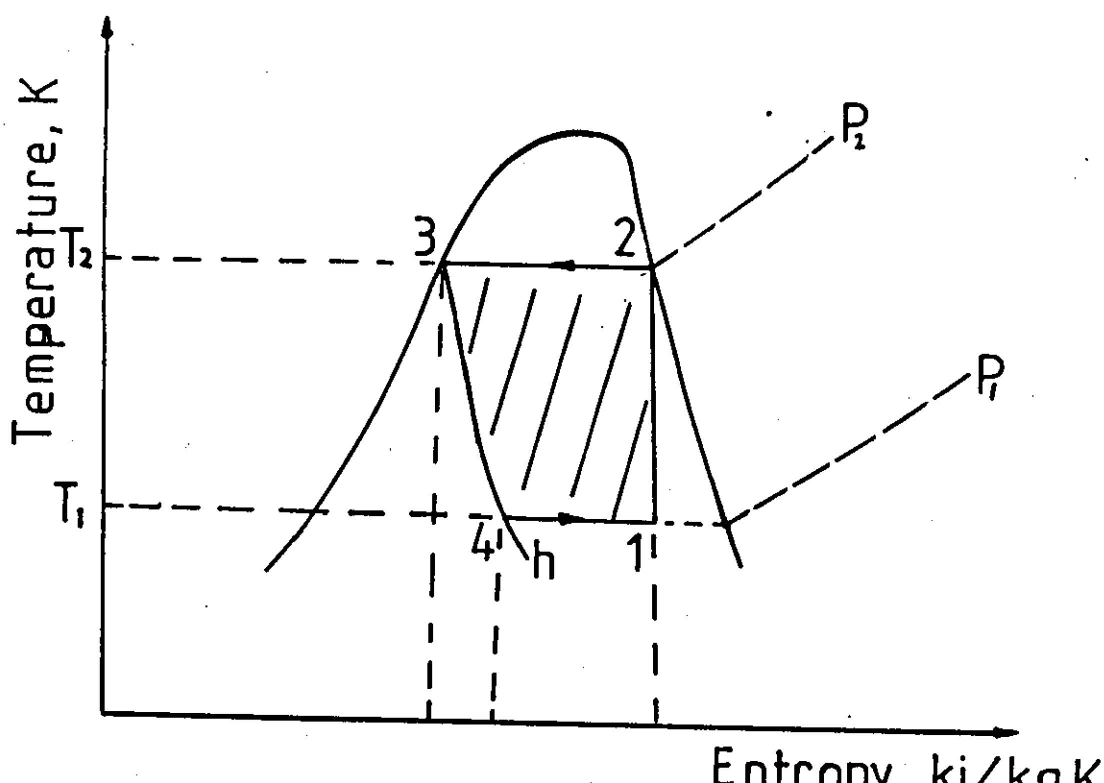


Fig. 2-11: The Rankine cycle which more accurately represents practical heat pump operation (93).

Thus the COP of real heat pumps is significantly less than that of the carnot cycle by 50% (31, 43).

For a heat - pump to show an overall energy saving the primary fuel equivalent of the heat recovered must be greater than the primary fuel required for the power input to the pump.

Applications

The heat pump cycles can be used to yield substantial energy savings especially

(a) When heat to be pumped has a high-thermodynamic and financial value.

(b) Where the temperature difference between the condenser and the evaporator is small.

(c) Where the primary energy usage is high. Examples of these would be in distillation, evaporation and drying. Two types of heat pumps can be distinguished (31, 43).

(i) The closed cycle heat pump with sealed recirculating working fluid as shown in Fig. 2 - 12.

(ii) The open cycle heat pump as shown in Fig.2 - 13. This is just another name of mechanical vapour recompression.

Furthermore, in a heat pump application a distinction can be drawn between recovery of sensible heat and of latent heat. Sensible heat recovery involves the use of closed cycle heat pump, which has relatively low working temperatures usually below 100^{o}C, although there is considerable development aimed at increasing the operating temperatures (43). The use of heat pump to recover latent heat of vapourisation is the application which offers greatest potential for the chemical and Process industry (43). In processes involving vapourisation the latent heat of vapourisation usually represents the major energy requirement and these processes are usually large energy users.

2.6.4 Recovery of Cold

Refrigeration energy increases with decreasing temperature. The theoretical amount of work, W, necessary to produce cold Q is given by equation 2 - 2.

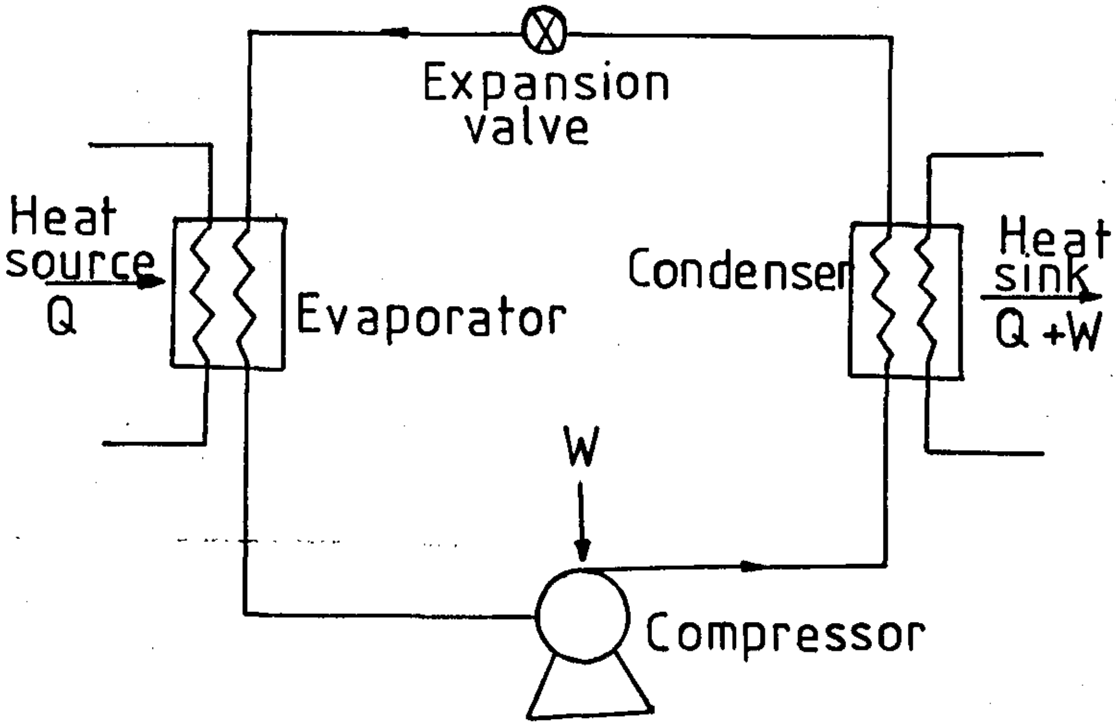


Fig. 2-12: Closed cycle heat pump (43).

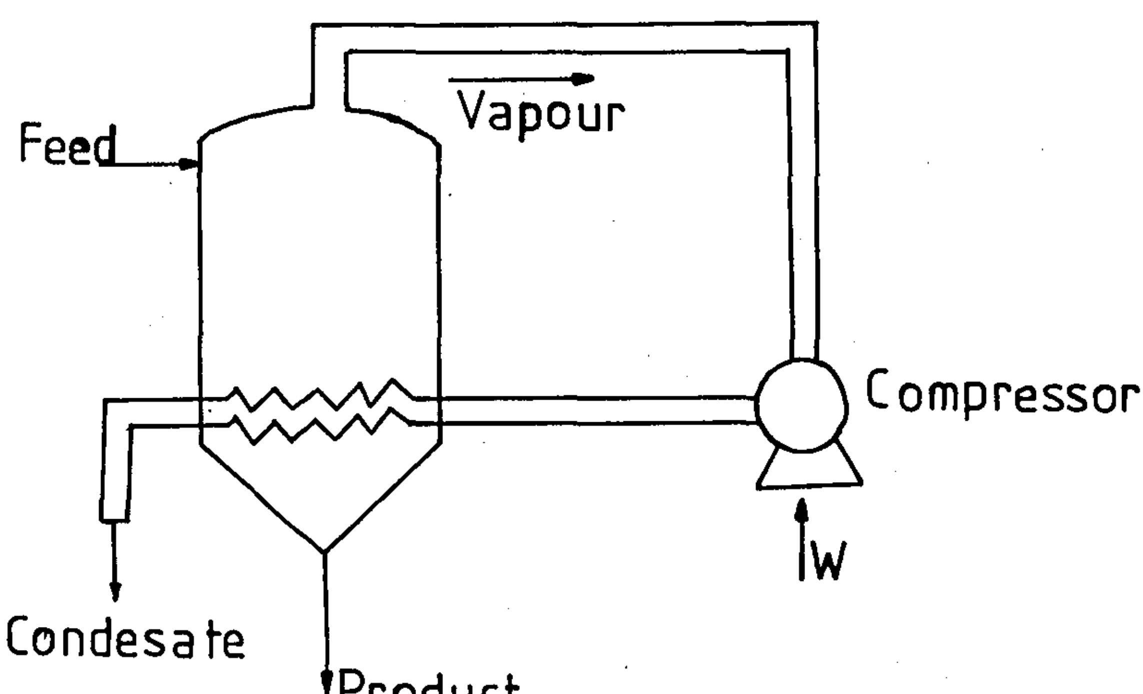


Fig. 2-13: Open cycle heat pump applied to evaporation (43).

Combined with the cost of producing power in the power station, it can be concluded that:

(a) Refrigeration requirements should be minimised.

(b) The cheapest form of developing refrigeration should be used e.g. absorption cooling (13) driven by waste heat, and the use of cold process streams or turbo - expenders.

(c) Cold should be recovered wherever possible. A few basic techniques to recover cold are commonly used in cryogenic processes (93) e.g. separation of air, recovery of ethane and propane from natural gas, purification or liquefaction of gases. These include:

   (i) Conserving refrigeration by using feed-effluent exchangers to cool feed gases with cold products.

   (ii) Limiting irreversibility in exchangers by minimizing temperature drop.

   (iii) Using turbo - expanders to produce cold, Fig.2 - 14, in a thermodynamically efficient manner especially at low, cryogenic temperatures.

2.7 The Economics of Conservation Measures

In previous section, we were looking into the aspects of energy conservation, and now we are in position to analyse the implication of the statement appearing in section 1.4 (i.e. "Most of the industrial energy conservation measures, are rather expensive and thus less attractive, when compared to those in the domestic sector or rather to those in space heating systems").

Consider the following two numerical examples.

(a) A typical factory in the East Midlands with a brick - brick cavity wall and light weight plaster on internal surface, is to have cavity fill insulation installed.

   Data :-

   - Type of insulation : 50mm Urea formaldehyde
   - External wall area : 2000 m^2
   - Heating season : 5500 hrs
   - Internal temperature: 20 ^oC

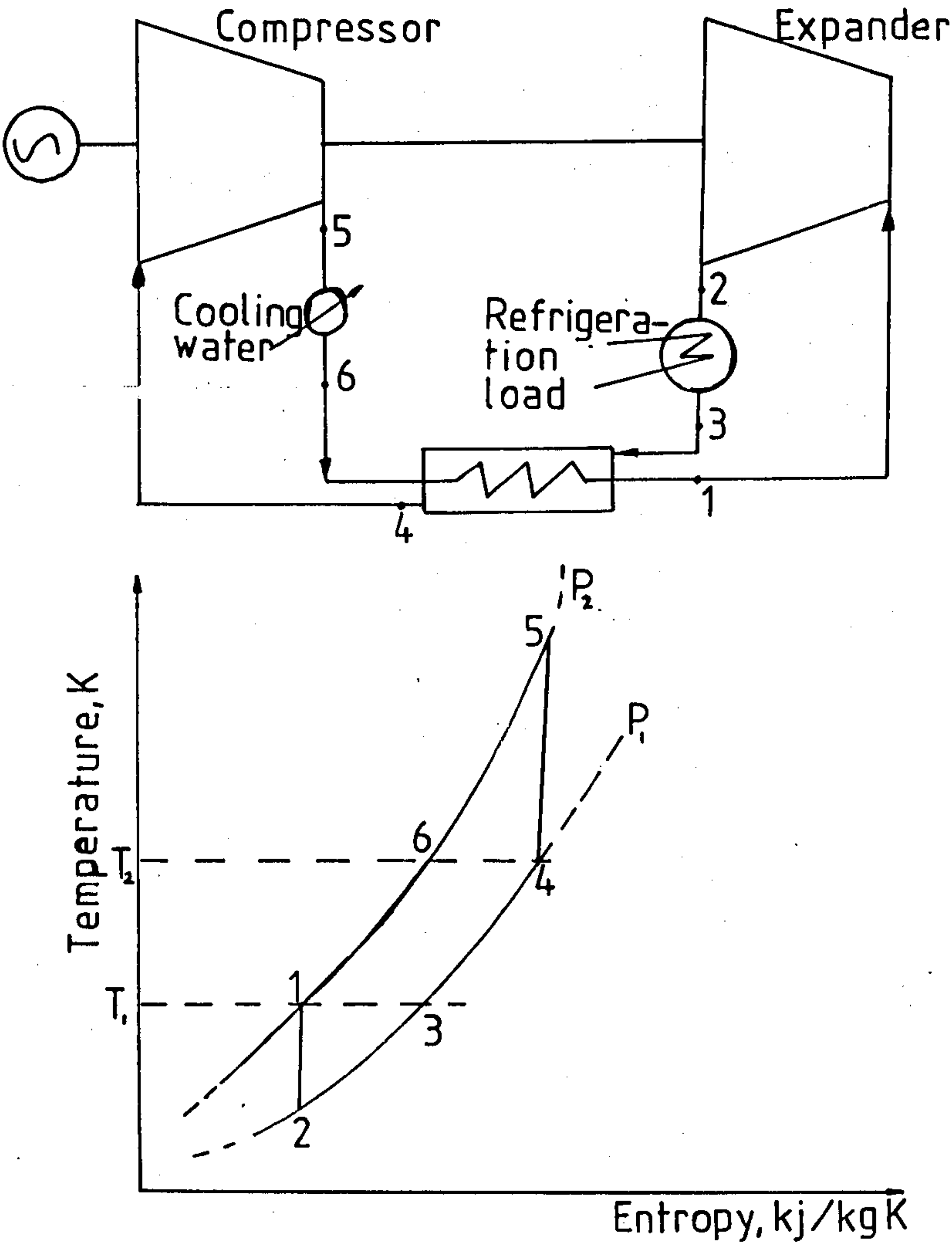


Fig.2-14: Turboexpander cooling-schematic representation of installation and thermodynamic diagram of an ideal cycle (43).

- External temperature : 8 °C
- Boiler efficiency : 80%.
- Fuel Cost : £4.5/GJ (1983 prices (27))
- Insulation Cost : Cost of injecting 50mm of urea formaldehyde is approximately £1.5/m^2 (1983 prices (27))
- Evaluation period : 1 year for determining annual saving.

The economic evaluation is as follows:

(i) From nomogram 1 (27) - gross cost of heat = £4.90 gross/GJ

(ii) From nomogram 2 (27) - Cost of useful heat = £6.0/useful GJ.

(iii) From nomogram 3 - Cost factor = £0.12/W

(iv) U - Values from table 19 (27);

- Uninsulated wall = 1.37 W/m^2K
- Insulated wall = 0.56 W/m^2 K

Heat losses;

- Uninsulated wall = 16.4 W/m^2
- Insulated wall = 6.7 W/m^2

This represents an energy serving of about 59%.

(v) from nomogram 4, for each reduction in heat loss at the cost factor of £0.12/W the total cost of heat over the evaluation period of one year are;

- Uninsulated wall = £2.0/m^2
- Insulated wall = £0.8/m^2

This represents an annual saving of £1.20/m^2, resulting in a payback period calculated as follows:-

$$\text{Payback period} = \frac{\text{installed cost of insulation}}{\text{Saving per annum}}$$

$$= \frac{£1.5}{£1.2} = 1.25 \text{ years}$$

- The total investment cost = £1.5/m^2 x 2000 m^2 = £3000

- Total amount of heat saved = (16.4 - 6.7) x 2000 = 19.4 kW

• Therefore the investment per kW saved

$$= \frac{£3000}{19.4} = £155/\text{kW saved.}$$

(b) A steel finned economiser was installed in the boiler flue of the same factory, with a total installed cost of £25,000 (97). The boiler operates mainly on natural gas. The economiser measures 2.5 m square and 300 mm deep in the direction of gas flow. It is installed in a horizontal flue.

Data:-

- Exhaust gas temperature : 232 °C
- After economiser : 160 °C
- Exhaust flow : 10.6 m^3/s
- Fuel cost : £4.5/GJ
- Evaluation period : 1 year for determining annual saving.
- The specific heat capacity of flue gases in that temperature range is approximately 1.47 kJ/m^3 K.

Thus, the total heat recovered is given by;

$$Q = \overset{*}{V} c_{PV} (\Delta T) \quad . . . . \quad 2.3$$

$$Q = 10.6 \times 1.47 (232 - 160) = 1122 \text{ kW.}$$

A payback period of less than 10 months (for a similar equipment) has been reported (97).

• The investment per kW saved

$$= \frac{£25000}{1122} = £22.3/\text{kW}$$

Despite the fact that, the payback period and capital expenditure per kW saved favours the flue gas heat recovery scheme, the magnitude of the money involved (i.e. £25000)is very big. In most cases such proposal will be met with scepticism by the management. On the other hand the £3000 invenstment with a payback period of about 1.2 years could be approved without difficulty by the management. These two examples illustrate the difficulty of assessing economically viable heat recovery projects from the investment point of view.

CHAPTER 3

ENERGY FORECASTING

Energy auditing requires investments in instrumentation, time and manpower which could be prohibitive to some industries. However, energy forecasting provides a simple and inexpensive means of assessing the energy usage and thus enabling the management to exercise control on this.

Forecasting is not a new problem, it has exercised the minds of management of different disciplines for decades. The accuracy of the forecasting method depends upon the purpose it has to serve, thus, the method has to be reviewed time after time. The rapidly escalating cost of purchased energy makes accurate forecasting of energy requirements a more urgent matter for the Chemical and Process industries. However, a large number of variables affect the specific energy consumption for a given plant, and engineers have traditionally had difficulty in providing accurate (or, in many cases, credible) forecasts.

The present forecasting method for space heating plants, the degree day method, is one example. This method assumes that the energy consumed by the space heating plant depends upon the degree days alone, but a large number of variables such as wind speed, rainfall, solar radiation physical properties of the building, the habits of the occupants also affect it. In addition there are energy gains from occupants, internal appliances and lighting (2, 12, 27, 39, 49, 96).

More than 40 years ago when this method was introduced, the cost of energy was very low in the sense that the expected 25% errors in the predicted results were tolerable. With the depleting energy sources and their increasing cost trends, this can no longer be comfortably accomodated. Thus a more accurate energy forecasting method is desirable. The objective of the present work is to improve the degree day method and make it a credible tool for energy managers to control the energy consumption without involving too much cost.

3.1 Forecasting Techniques

Forecasting can be defined as estimating the demand over the future time intervals or over given operating conditions, and are generated using the demands from the past (39, 49, 62, 114).

Forecasting techniques can be categorised into three groups (114), viz:

(a) Qualitative; Sometimes called subjective. All information and judgement relating to an item are used to forecast the item demand. This technique is often used when little or no historical data are available.

(b) Causal ; Sometimes called multivariate or prediction. The cause - and - effect type of relationship is sought. Here, the forecaster seeks a relationship between item's demands and other factors. For example the relationship between the energy consumption and production rate, or degree days is sought. The relationship is then used to predict the energy demand for given operating conditions.

(c) Time - series analysis; Sometimes called univariate or projection. A statistical analysis on the past demands is used to generate the forecasts. A basic assumption used is that the underlying trends of the past will continue into the future.

Our work is primarily concerned with forecasting as it relates to the causal approach. Although in practice a forecasting procedure may involve a combination of the approaches.

The forecasting process begins by developing a statistical type of fit through the historical data. For instance, energy consumption and production rate for a process plant. The fit that is sought may vary depending on the pattern of historical demands. Four demand patterns can be distinguished, viz:

(i) The constant demand pattern: of the form

$$y = C$$

where: C = constant

(ii) Trend demand pattern : of the form

$$y = C_1 + C_2X$$

(iii) Quadratic demand pattern : of the form

$$y = C_1 + C_2X + C_3X^2$$

(iv) Trend seasonal demand pattern : these are of two forms

(1) Multplicative type : $y = (C_1 + C_2X)\mathcal{S}_x$

(2) Additive type : $y = C_1 + C_2X + \mathcal{S}_x$

where: $\mathcal{S}_x$ = Seasonal influence

Thus the prevailing demand pattern can be detected by plotting the historical data on either a linear or logarithimic graph paper. The plot will also reveal the wild observations, which do not appear to be consistent with the rest of the data. In statistical terms these are often called OUTLIERS. The reasons explaining their occurance should be found, and if justified they should be discarded completely.

3.2 Linear Regression Analysis

Assume that the relationship between the energy consumption and the production rate vary according to the following equation:

$$y = C_1 + C_2X \quad \ldots\ldots\ldots\ldots \quad 3.1$$

Where y = energy consumption, kJ/day

C_1 = base load, kJ/day

X = production rate, kg/day

Then, the historical energy demand and production rates are used in the least square method with equal weights to seek the estimates of C_1 and C_2 in equation 3.1. With this procedure the estimates of C_1 and C_2 so obtained, yield residual errors with the property that the sum of the squares of the residual errors is minimum (22, 23, 38).

If we denote the predicted value by $\hat{y}$ then the residual error will be

$$e = y - \hat{y} \quad \ldots\ldots\ldots\ldots \quad 3.2$$

and the sum of squares of the residual error will be

$$S(e) = \sum_{i=1}^{n}(y - \hat{y})^2 \quad \ldots\ldots\ldots \quad 3.3$$

The estimates of C_1 and C_2 that will make equation 3.3 minimum are given by;

$$C_1 = \frac{(\Sigma y)(\Sigma X^2) - (\Sigma X)(\Sigma Xy)}{n(\Sigma X^2) - (\Sigma X)^2} \quad \ldots\ldots\ldots \quad 3.4$$

$$C_2 = \frac{n(\Sigma X y) - (\Sigma X)(\Sigma y)}{n(\Sigma X^2) - (\Sigma X)^2} \quad \ldots\ldots\ldots\ldots \quad 3.5$$

3.2.1 Regression Analysis

Predictions based upon least - square method, should not be looked upon as perfect, indeed, to make such predictions meaningful they should be interpreted as expectations (36, 38, 114, 120). The values obtained for $\hat{C}_1$ and $\hat{C}_2$ by the method of least - squares are estimates of C_1 and C_2. In linear regression analysis, it is assumed that the X's are constants, i.e. not values of random variable, and that for each value of X the variable to be predicted has a certain distribution as shown in Fig. 3 - 1, furthermore, that, these distributions are all normal curves having the same standard deviation, σ (38). If the X's are also looked upon as values of random variables the treatment of the data is referred to as correlation analysis, a subject appearing in the following subsection. Thus, in our work we will consider X's as values of random values.

The standard error of estimate is then given by:

$$Se = \left[\frac{(\Sigma y^2 - C_1(\Sigma y) - C_2(\Sigma Xy)}{n - 2}\right]^{1/2} \quad \ldots\ldots \quad 3.6$$

Where: n - 2 = the degree of freedom.

3.2.2. The Correlation Coefficient

This is the criterion of comparing the sum of the squares of the vertical

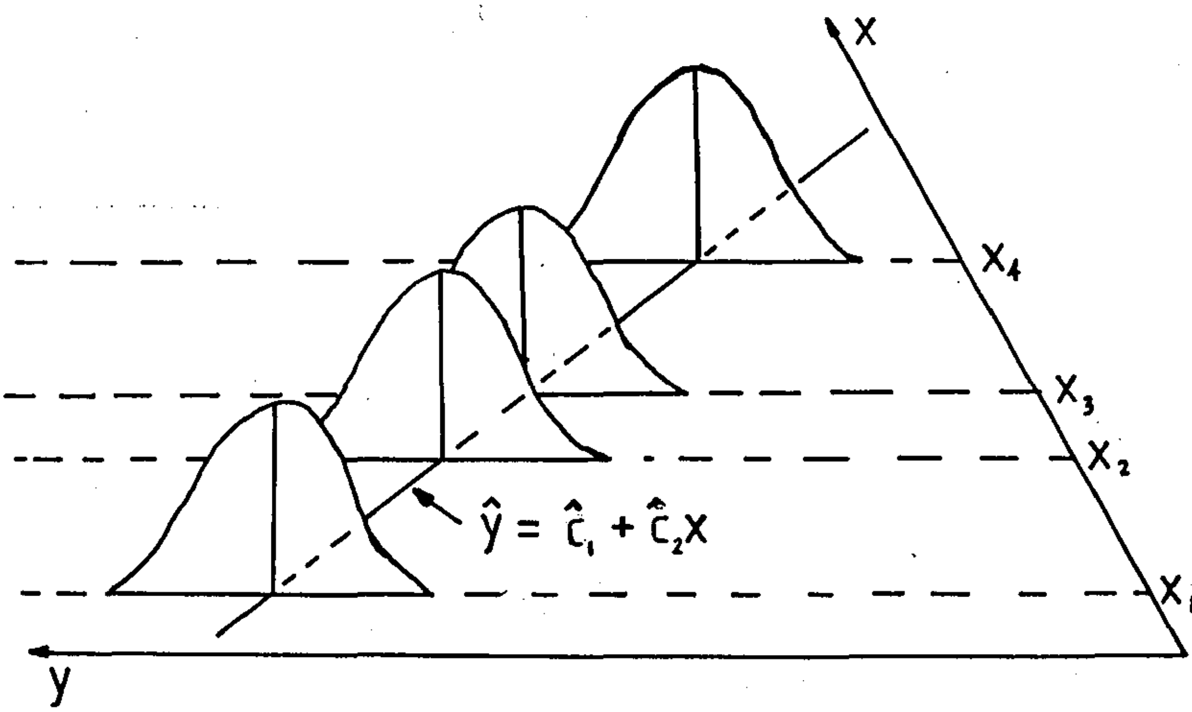


Fig. 3-1: Distribution of the y's for various values of x (38).

deviations from the least square line, $\sum(y - \hat{y})^2$, with the sum of the squares of the deviations of the y's from their mean, $\sum(y - \bar{y})^2$ (36, 38). The correlation coefficient can be defined as

$$r = \pm\left(1 - \frac{\sum(y - \hat{y})^2}{\sum(y - \bar{y})^2}\right. \quad \ldots\ldots\ldots\ldots \quad 3.7$$

Therefore, if the fit is poor, $\sum(y - \bar{y})^2$ and $\sum(y - \hat{y})^2$ are almost the same and thus the correlation coefficient is close to zero. If the fit is good, $\sum(y - \hat{y})^2$ is much smaller than $\sum(y - \bar{y})^2$, therefore their ratio is close to zero, and the correlation coefficient is close to $\pm$ 1. The correlation coefficient measures the strength of linear relationship between the variables. It indicates the goodness of the fit of a line fitted by the method of least square method, and this is the criterion that will be used in this work.

3.3 The Forecasting Equation

3.3.1 Allocating Energy Consumption

The energy consumption can basically be separated into two categories, viz:

(a) That required for production purposes

(b) That required for space heating in temperate countries.

However, there are other utilities and services such as hot water, lighting and cooking, and several losses such as convection and radiation losses from the boiler shell,steam and condensate lines, which occur continuously whether there is production, space heating or none, provided the boiler is on. These have to be accounted for, and they give rise to the third category of energy consumption, termed as the constant or base load (2, 12, 43).

(i) Base load

The base load consists of;

(1) Fuel used in start up of boilers, if it does not operate continuously, for example, preheating the steam recirculation lines.

(2) Supply of hot water for personnel use in washing and cleaning.

(3) Energy used in cooking and washing in canteens.

(4) Lighting the factory premises.

(5) Steam mains and condensate system losses.

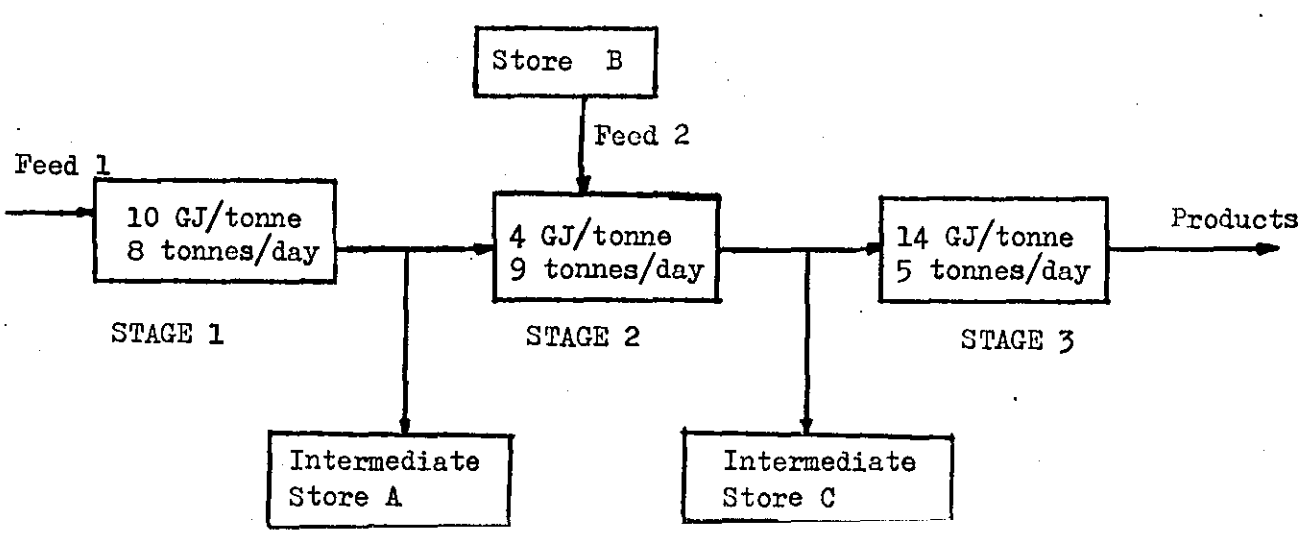
(6) Energy used to overcome losses from boiler shell, blow down, flash steam, oil storage tanks etc.

The steam and condensate system losses are encountered when the lines are live for a fixed period each day regardless of production level. Wherever the production rate is constant and variations in input are achieved by changing the periods of operation for the whole system, then these two items would be included in the base load (2). Care must be taken when dealing with continuous production units, because in some instances the actual energy input to the product may be much smaller than recorded. For example, a radiant dryer, could use almost the same energy whether at full load or at no load. Thus if the production output is varied by changing flow rate, then it is necessary to estimate the proportion used at no flow and add it to the base load term (2).

(b) <u>Production based:</u>

This is the amount of energy consumed by the processing equipment in transforming the raw material into the finished product. Complications in assessing this component arise:

(i) When several stages of operation, with each stage having a different throughput are involved. This is demonstrated in the hypothetical example below, for a particular day.



Total energy consumption per tonne = 10 + 4 + 14 = 28 GJ

Thus the total energy used = 8 x 10 + 9 x 4 + 5 x 14
= 186 GJ.

And the "equivalent" production for the whole process will be 186/28 = 6.6 tonnes.

(ii) When the raw material is processed several days in advance, is processed and scrapped or is partly or completely recycled, the production records will be completely different from that actually processed per day.

Thus, the production records, which are often based on the quantities dispatched may not be suitable for comparing the energy consumption and production level. Besides, it is a common practice to stock the products before sales. The best alternative therefore, is to quote the specific fuel consumption, which is given by:

$$\text{S.F.C.} = \frac{\text{Weekly fuel consumption}}{\text{Weekly production level}} \quad \ldots\ldots\ldots\ldots \quad 3.8$$

Equation 3.8 does not take into account the base load and thus the S.F.C. will tend to rise as the production throughput falls (2).

(c) Air conditioning as applied to buildings in commercial and manufacturing plants relates to the control of the working environment to maintain temperature and humidity within units appropriate to type of activity being carried out. It is also necessary in plants where the particular process being carried out necessitates temperature and humidity control. In many situations, air conditioning is limited to winter heating in temperate countries.

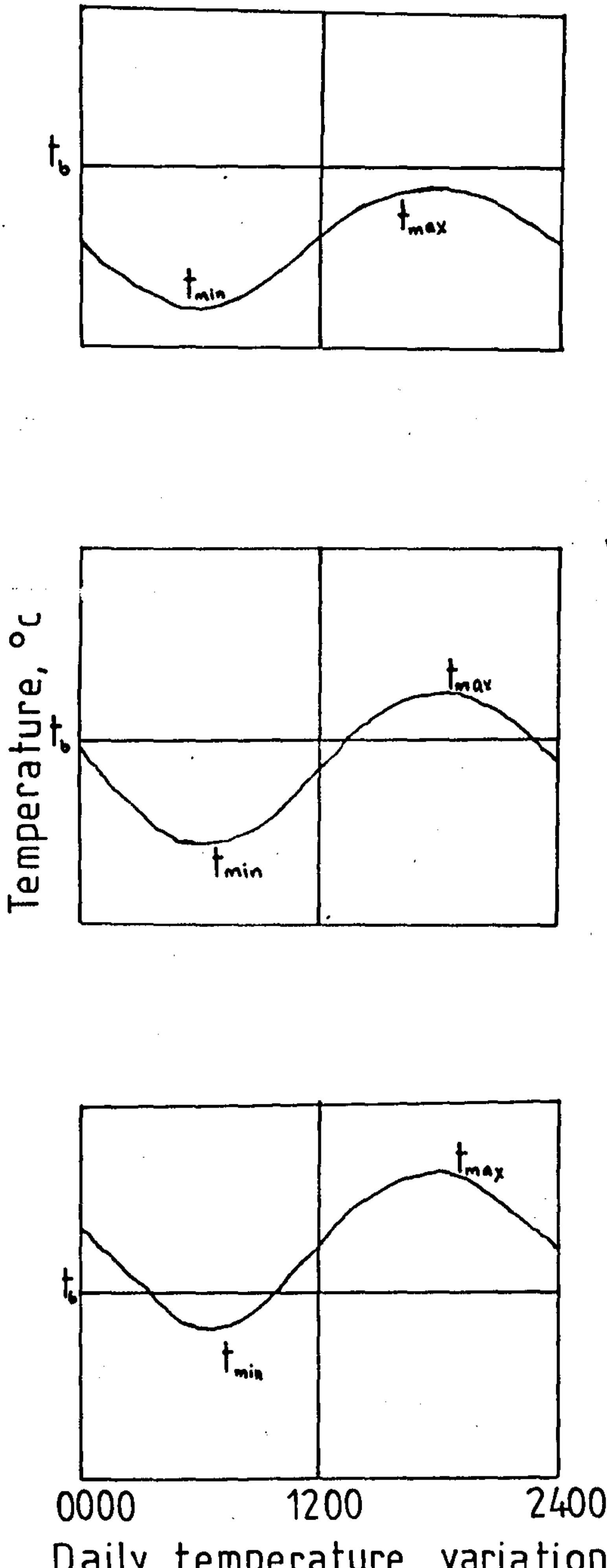
The current degree day method used to calculate the variations in energy for space heating plants, is discussed briefly in the following subsection.

3.3.2 Current Degree Day Method for Predicting Energy Consumption and Its Limitations.

Degree days can be defined as equivalent to the number of degrees(°C or °F) that the 24 - hour mean outside ambient temperature lies below a selected base temperature. In the U.K. this is usually 15.5 °C (60 °F) (2, 12, 27, 39, 43, 44, 47, 62, 96, 98, 99). In effect, the base temperature is the mean outside temperature at which inside temperatures are accepted with no heating. Thus, a 24 - hour period in which the mean outside temperature is 13 °C will have 2.5 degree days. Degree days are recorded during the 24 - hour period, which if temperature rises or falls rapidly may not be the true mean temperature such variations are assumed to even out over a period, however, when the outside temperature exceeds the base temperature for the part of 24 - hour period, special formulae are used for calculating the degree days, see Fig.3 - 2. (2, 27, 43).

The total number of degree days over a period provides a numerical indication of the average coldness of the locality, to which it is assumed that, the amount of energy needed to maintain a controlled inside temperature will be approximately proportional. There are a number of shortcomings to this assumption. No allowance is made for variation in plant efficiency with load, although it is reasonable to expect that energy consumption per degree day will be less in very cold weather when the heating plant is operating at high load than during warmer weather when it may be at low load for long periods (90, 98). In addition energy requirements may not be proportional to degree days when buildings are not continuously heated (27). It is often difficult to segregate energy used to provide hot water, especially when callorifiers are used, although an appropriate deduction can sometimes be determined based on summer consumption where there is no space heating.

Incidental heat gains from the sun, lighting plant, occupants and so on may also affect the figures. It is assumed that, these gains JUST make good the difference between the selected base temperature (15.5 °C) and the internal"comfort" temperature (e.g. 18.3 °C), but they may provide



when $t_{max} < t_b$

$$DD = t_b - \frac{t_{max} + t_{min}}{2}$$

when $t_{max} > t_b$ and $t_{av} < t_b$

$$DD = \frac{t_b - t_{min}}{2} - \frac{t_{max} - t_b}{4}$$

when $t_{max} > t_b$ and $t_{av} > t_b$

$$DD = \frac{t_b - t_{min}}{4}$$

Degree days

Fig. 3-2: Formulae for calculating degree days.(27)

more or less than this.

Wind speeds have a significant and variable effect on energy consumption especially in poorly draught - proofed buildings even if they are well insulated. The physical properties of the building also have a significant effect on energy consumption, since the fluctuation in internal temperature in a room (i.e. the temperature swing) is controlled by the admittance of the room (8, 9, 11, 25, 52, 72, 79). The greater the admittance the smaller the temperature swing, hence less energy input, see chapter 5.

The degree days are then used to predict the energy consumption for space heating, and the cumulative sum technique, outlined in section 3.3.3 can then be used to monitor the performance of the space heating plant.

3.3.3 The Cumulative Sum Technique

The ultimate goal of the energy predictions is to monitor and control the energy consumption. The cumulative sum technique, hereafter referred to as CUSUM, can be regarded as a useful tool in this respect. This technique involves subtracting the predicted value from corresponding actual figure in the series and accumulating the difference as each additional figure is introduced (2, 56, 86). For example, if a series of points $y_1, y_2, \ldots, y_n$ is being measured and if $\hat{y}_1, \hat{y}_2, \ldots, \hat{y}_n$ is the series of corresponding predicted points, then their cumulative sum is developed as;

$$S_1 = y_1 - \hat{y}_1$$

$$S_2 = (y_1 - \hat{y}_1) + (y_2 - \hat{y}_2) = S_1 + (y_2 - \hat{y}_2)$$

$$\vdots$$

$$S_n = S_{n-1} + (y_n - \hat{y}_n)$$

When the CUSUM values are calculated and plotted as they occur a CUSUM CHART is produced. Changes in mean level of the consumption, will be shown up as a change in slope on the CUSUM chart.

Fig.3 - 3 show an example of a typical CUSUM chart for an energy consuming plant, the vertical axis represent the cumulative sum of the differences between predicted energy consumption (obtained by relating energy consumption and production level, for example), and the actual consumption, while the horizontal axis represent the period, during which this trend was observed. It should be noted that, suitable scale of the CUSUM chart should be selected to avoid the danger of exaggerating chance variations and suppressing genuine changes. A scale of 45^{o} representing a change of 10% to 20% has been suggested (2, 56, 86). On the other hand, if there is no change in the mean consumption from the original data then the graph should alternate fairly randomly about a value of zero.

When using the regression equation to predict the energy consumption care should be taken, because, any slight error in the coefficients of the formula may produce a gradual rise or fall in the CUSUM slope. When a successful conservation measure has been adopted, or when there is an addition of energy consuming plant there should be a significant change in the slope of the CUSUM, and the date of the intersection of consecutive trend lines should correspond to identifiable causes. General trends should be sought so as not to attach unwarranted significance to short term variation. A period not less than 10 months has been suggested (2, 56, 86).

The following numerical example illustrate the procedure involved in formulating the equation for predicting energy consumption and constructing the energy monitoring system as outlined in the previous chapter.

3.3.4 Numerical Example

Assume the following energy consumption and production rate of a certain processing industry.

Production Tonnes/day X	108	90	121	115	105	130	118	95	143	160	102	80	90	130
Fuel Gallons/day	143	130	165	145	132	163	152	144	168	180	134	128	150	183

Production Tonnes/day X	105	105	95
Fuel Gallons/day y	152	155	167

We want to determine an equation for predicting energy consumption for a given production rate, and to construct an energy monitoring system by using the theories outlined in chapter 3.

If we assume that the line which fit in the above data is of a linear form then the regression equation is given by equation 3.1.

i.e. $\hat{y} = C_1 + C_2X$

The regression coefficients, C_1 and C_2 are calculated by using equations 3.4 and 3.5 respectively. The evaluation procedure is outlined in table 3.1. Thus, substituting the known values in equations 3.4 and 3.5 we have

$$C_1 = 84.16 \text{ gallons}$$
$$C_2 = 0.61 \text{ gallons/tonne}$$

Therefore, the regression equation is

$$y = 84.16 + 0.61\ X \text{ gallons.}$$

Where 84.16 gallons is the base load. The correlation coefficient is calculated by using equation 3.7, which when expanded give an equation of the form:

$$r = \frac{n\Sigma Xy - \Sigma X\,\Sigma y}{\sqrt{n(\Sigma X^2) - (\Sigma X)^2}\ \sqrt{n(\Sigma y^2) - (\Sigma y)^2}} \quad \ldots\ldots\ldots\ldots 3.9$$

substituting the values from table 3.1 it is found that,

$$r = 0.75$$

This correlation coefficient suggest that the relationship between fuel consumption and production rate is not VERY good. Most probably, the fuel consumption does not depend solely on the production rate.

The standard deviation is calculated from equation 3.6. Substituting the values,

S_e = 13.8 gallons or 9% of the mean consumption. This is termed as the standard error of estimates.

In table 3.1, for each given value of X (column 1) we have an observed y (Column 2) and a calculated value $\hat{y}$ (Column 6) obtained by substituting X in $\hat{y} = 84.16 + 0.61$ X. The differences (i.e. y - $\hat{y}$) (column 7) are accumulated (Column 8) and plotted as they occur to form a CUSUM chart, as portrayed in Fig.3.3, for monitoring purposes. It can be seen that there is a change in the slope of the CUSUM chart. This change in real terms, represent an increase of about 10% in the mean energy consumption, and principally, it should correspond to identifiable cause, such as an addition of an energy consuming unit in a process line or a severe energy loss is taking place.

Table 3 - 1 : Evaluation Procedure of the regression equation and the CUSUM

Column 1	2	3	4	5	6	7	8
Production, X(Tonnes)	Fuel, y (Gallons)	X^2	Xy	y^2	Predicted fuel, $\hat{y}$ (gallons)	$y - \hat{y}$	CUSUM $S_n = S_{n-1} + (y - y_n)$
108	143	11664	15444	20449	150.04	- 7.04	- 7.04
90	130	8100	11700	16900	139.06	- 9.06	-16.10
121	165	14641	19965	27225	157.97	+ 7.03	- 9.07
115	145	13225	16675	21025	154.31	- 9.31	-18.38
105	132	11025	13860	17424	148.21	-16.21	-34.59
130	163	16900	21190	26569	163.46	- 0.46	-35.05
118	152	13924	17936	23104	156.14	- 4.14	-39.19
95	144	9025	13680	20736	142.11	+ 1.89	-37.30
143	168	20449	24024	28224	171.39	- 3.39	-40.69
160	180	25600	28800	32400	181.76	- 1.76	-42.45
102	134	10404	13668	17956	146.38	-12.38	-52.83
80	128	6400	10240	16384	132.96	- 4.96	-59.79
90	150	8100	13500	22500	139.06	+10.94	-48.85
130	183	16900	23790	33489	163.46	+19.54	-29.31
105	152	11025	15960	23104	148.21	+ 3.79	-25.52
105	155	11025	16275	24025	148.21	+ 6.79	-18.73
95	167	9025	15865	27889	142.11	+24.89	+ 6.16
$\sum X=1892$	$\sum y=2591$	$\sum X^2=217432$	$\sum Xy=292572$	$\sum y^2= 399403$			

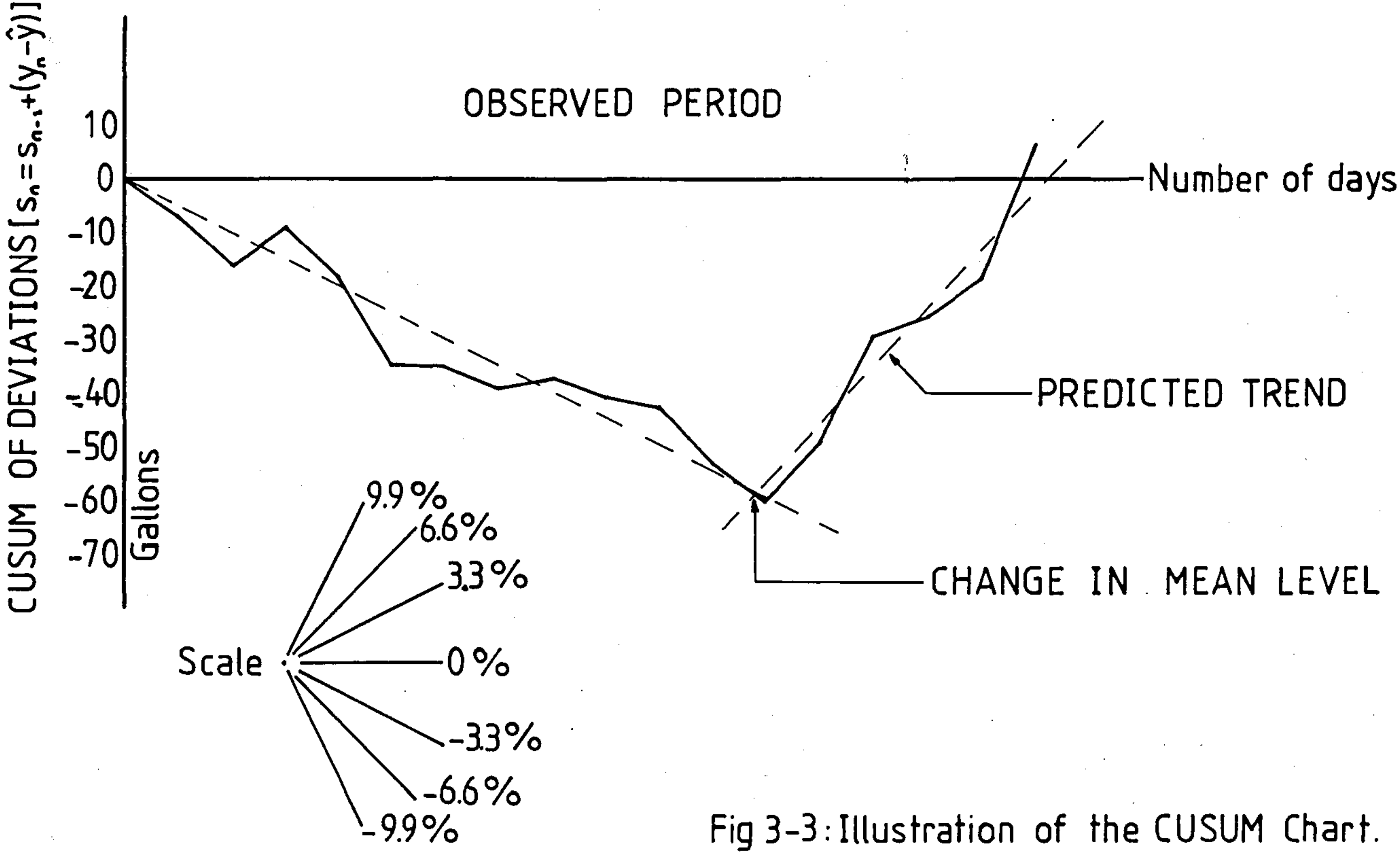


Fig 3-3: Illustration of the CUSUM Chart.

CHAPTER 4

THE EFFECT OF WIND SPEED UPON HEAT REQUIREMENTS

According to Miller (80), "When considering the overall increase in heat loss from an enclosure due to the influence of wind speeds in excess of those specified at a particular design condition, it is necessary to examine the relationship between the ventilation and structural losses in order to assess the additional thermal loads imposed".

The thermal load imposed by the infiltration of cold outside air into the interior of a heated building is a function of the prevailing wind speed and the difference between the internal and external air temperatures, which parameters also affect the rate of heat flow through the building fabric. Consequently, the total heat requirement with a strong wind and a moderate external temperature may be equal or exceed that with a light wind and a lower external temperature (60, 61, 62).

The estimation of air infiltration rates, however, is a complex task and at present only the broadest of approximations are used. One method of estimation is based on measured leakage rates of building components such as windows and doors with respect to the pressure difference across them. This is known as the "Air change" method involves the assumption of a certain number of air changes per hour for each room, depending on its location, number of exposed sides, window area, frequency of opening the doors, living habits of the occupants and other similar criteria.

Air infiltration is estimated to account for between 25% and 50% of the space heating demand for both domestic and commercial buildings (49, 59, 62, 71, 80). However, it is worth noting that the percentage effect of infiltration loss, to large extent, depend upon the level of insulation and area and type of glazing. That means, if the building is well insulated, with weather stripped double glazed windows, a substantial part of fuel consumed in space heating plant will be due to infiltration loss. Thus care should be taken in interpreting the resultant percentage effects.

For example, if a formally uninsulated building is insulated to reduce fabric heat loss, the infiltration loss will remain more or less constant. Consequently the percentage effect of the infiltration loss will increase. In the preceding sections the effect of wind speed on both infiltration and fabric heat losses will be discussed.

4.1 Infiltration Loss

Air infiltration can be defined as the fortuitous leakage of air through a building due to imperfections in the structure, mainly as cracks around windows and doors. It depends upon the air tightness of, and the pressure differential across, the containing structure, and the areas and resistances of the openings and interstices to air flow (13, 17, 18, 24, 26, 54, 59, 60, 61, 71, 80, 95, 103, 104, 113).

The pressure differential is caused either by the wind in which case air enters through cracks and apertures in the windward side leaves through similar openings in the leeward side, or due to the buoyancy of the air commonly termed as the "stack effect", caused by the difference in temperature and hence density, between similar columns of air inside and outside the building.

Wind speed and direction are very dependent upon local topography, for instance the general effects of a built-up area and the particular effects of the pressure of neighbouring buildings, trees, hills and valleys upon the pressures over the facades of a structure may be so complex as to be actually unpredictable. These parameters and those mentioned in previous section explain the reason for having a number of diverse mathematical models for estimating the air infiltration rates put forward by different research workers. Since we were not in position to validate any mathematical model for this particular site, a study of these models was necessary.

4.1.1 Mathematical Models for Estimating Air Infiltration Rate

Typically, each mathematical model presented in this sub-section was a result of field measurements conducted in 1 to 24 residential houses, of different height, age and topographical location, which parameters affect the wind speed and direction, leakage characteristics of the building and consequently air infiltration equation.

Coblentz et al in 1963 (24) proposed a **linear** relationship between air infiltration rate and wind speed and temperature difference as follows:

$$I = C_1 + C_2 V + C_3 \Delta T \quad \ldots\ldots\ldots\ldots \qquad 4.1$$

Where:

I = air infiltration, air change/hr

C_1 = air change rate with no wind or temperature difference, air change/hr

V = Wind speed, m/s

T = temperature difference, ^{o}C

C_2, C_3 = the increase in the air change rate per unit increase in wind speed and temperature difference respectively.

Hunt in 1980 (55) gave a similar relationship with some modifications as follows:

$$I = C_1 + C_2 V + C_3 (\Delta T)^{0.5} \quad \ldots\ldots\ldots \qquad 4.2$$

Equations 4.1 and 4.2 suggests that the effect of wind speed and temperature induced convection act independently, i.e. they can be simply added. This might be approximately true for single unit houses, from which most of the studies on infiltration rate were made. For tall buildings these parameters are complementary at the lower part of the building and they act contrary to one another at the upper part of the building (i.e. while the wind drives air inside the building the stack effect tend to cause an outward movement). This means, that equations 4.1 and 4.2 will give an exaggerated effect.

However, if we could leave the temperature induced term for the latter section, the heat loss due to air infiltration is given by $Q = C_p \rho I \Delta T$ (where I is in m^3/s) thus $Q \propto I$. Since $I \propto V$, then $Q \propto V$. This is demonstrated in table 4.1 for wind speeds of 3, 6 and 9 m/s relative to a velocity of 2 m/s.

A different approach was taken by Blomsterberg et al (13), Jackman (60, 61, 62), Lidament (72) and Miller (81). This approach is based on pressure induced infiltration due to wind speed.

4.1.2 Pressure Due to Wind Speed

Wind effects, based on mean wind speed over and around a building cause a pressure difference from inside to outside. This wind pressure on a building can be calculated from Bernoulli's equation (4, 13, 59, 60, 61, 62, 93).

$$\Delta P_w = \frac{1}{2} C \rho V^2 \quad \ldots\ldots\ldots\ldots \quad 4.3$$

Where: ΔP_w = wind pressure, N/m^2

ρ = air density, kg/m^3

C = pressure coefficient depending on the form of the building and the exposure.

V = wind speed at a height equal to that of the building, m/s.

The British Meteorological offices records hourly wind speeds measured at a height of 10 m above an open level site. However, it has been observed earlier that, local factors such as topographical features, height of the building and interaction with other buildings, affect the wind speed and the manner in which it acts upon a given building and as a consequence must be taken into account. The Building Reseach Establishment digest 210 (18), gives a simple formula for different types of terrain as follows:

$$V = V_m \, K Z^n \quad \ldots\ldots\ldots\ldots \quad 4.4$$

Where: V = local wind speed, m/s

V_m = Meteorological wind speed at height of 10m

K = Coefficient depending on terrain

n = exponent relating wind speed to height depending on terrain

Z = height, m

For a particular site however, the K, Z and n are constant. Thus $V = C\ V_m$ where C is a constant including K, Z and n. Therefore if one is looking for the percentage effect exerted by the wind speed, the Meteorogical wind speed can be used instead of local wind speed without causing significant errors. Thus, in this work the Meteorological wind speeds will be used instead of local wind speeds.

When wind blows at right angles to a building it is stopped by the face of the building and as a result produces a pressure on the windward face. The wind is deflected over the top and around the sides of the building which gives suction on these other faces as indicated in Fig.4.1 It is stated in reference 18 that the wind pressure tends to be greatest near the centre on the windward face, with the most severe suction at the edges and corners.

The IHVE Guide (58) states that pressure on the windward side varies from 0.5 to 0.8 times the velocity pressure of the free wind, while on the leeward side the suction or negative pressure is from 0.3 to 0.4 times the velocity pressure of the free wind. Somewhat similar values are given in the ASHRAE Handbook of fundamentals (4).

From equation 4.3 it can be seen that an increase in wind speed produces a considerable increase in suction. Building Research Establishment Digest No.119 (17) draws attention to this and shows how the channelling of wind between two buildings can result in an increase in speed and hence a large suction effect on the sides facing the gap, Fig. 4.2. In addition, the interaction of high and low rise buildings Fig.4.3 can cause a considerable wind turbulence producing a speeding up of the wind with resulting large increase in the wind pressures.

Fig. 4 - 4a and 4 - 4b show the aerial photographs of the company and it

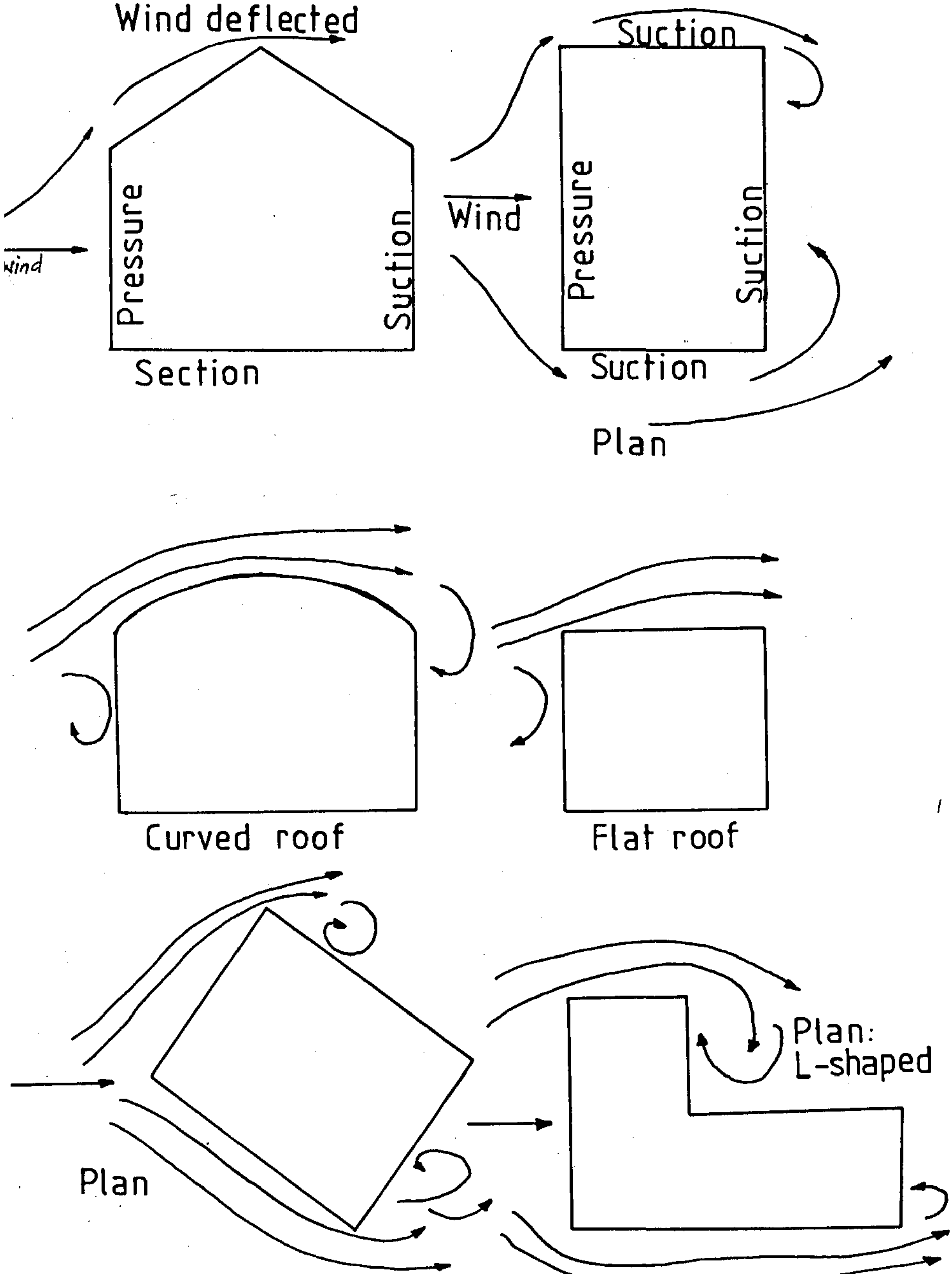


Fig. 4-1: Irregular wind flow around different building structure (18)

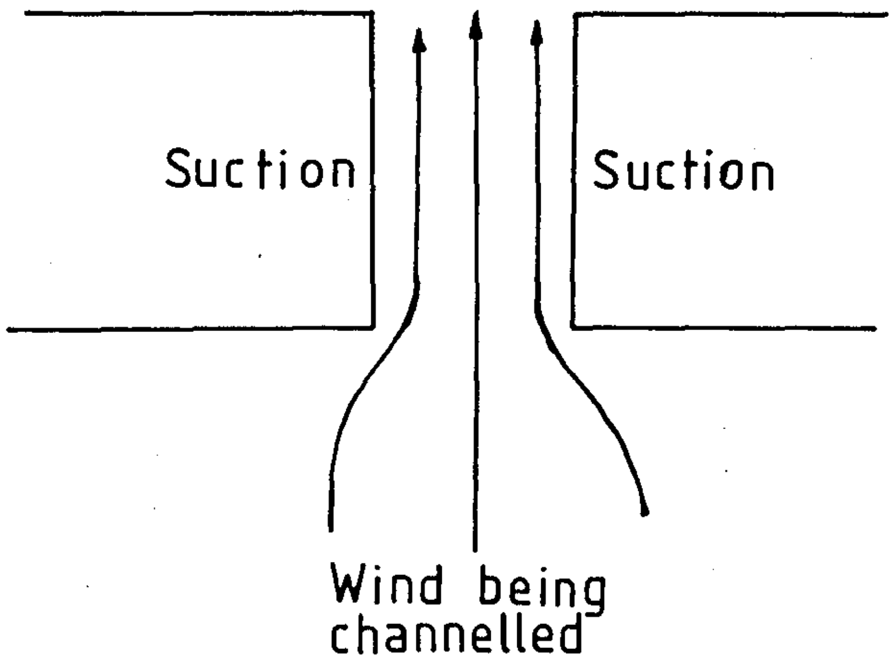


Fig. 4-2: Channelling of wind. (18)

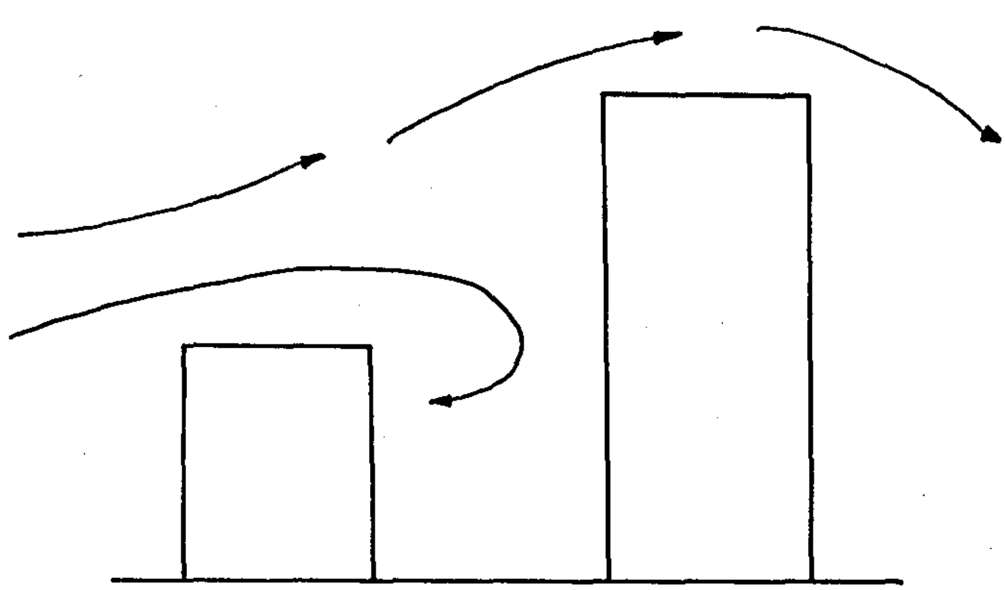


Fig. 4-3: Interaction of low-and high-rise buildings.(18

Fig. 4-4a: Aerial photograph of the company.

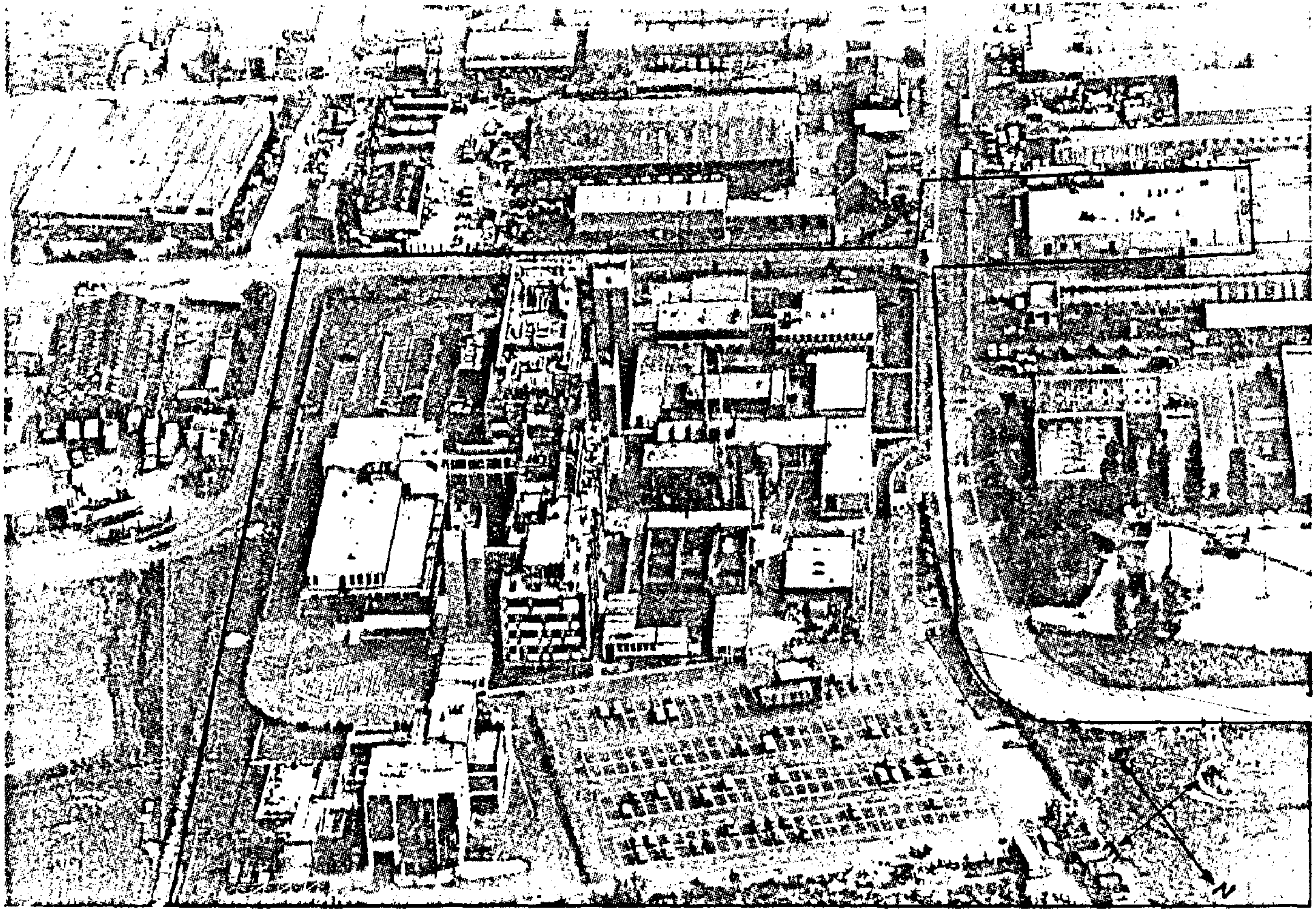
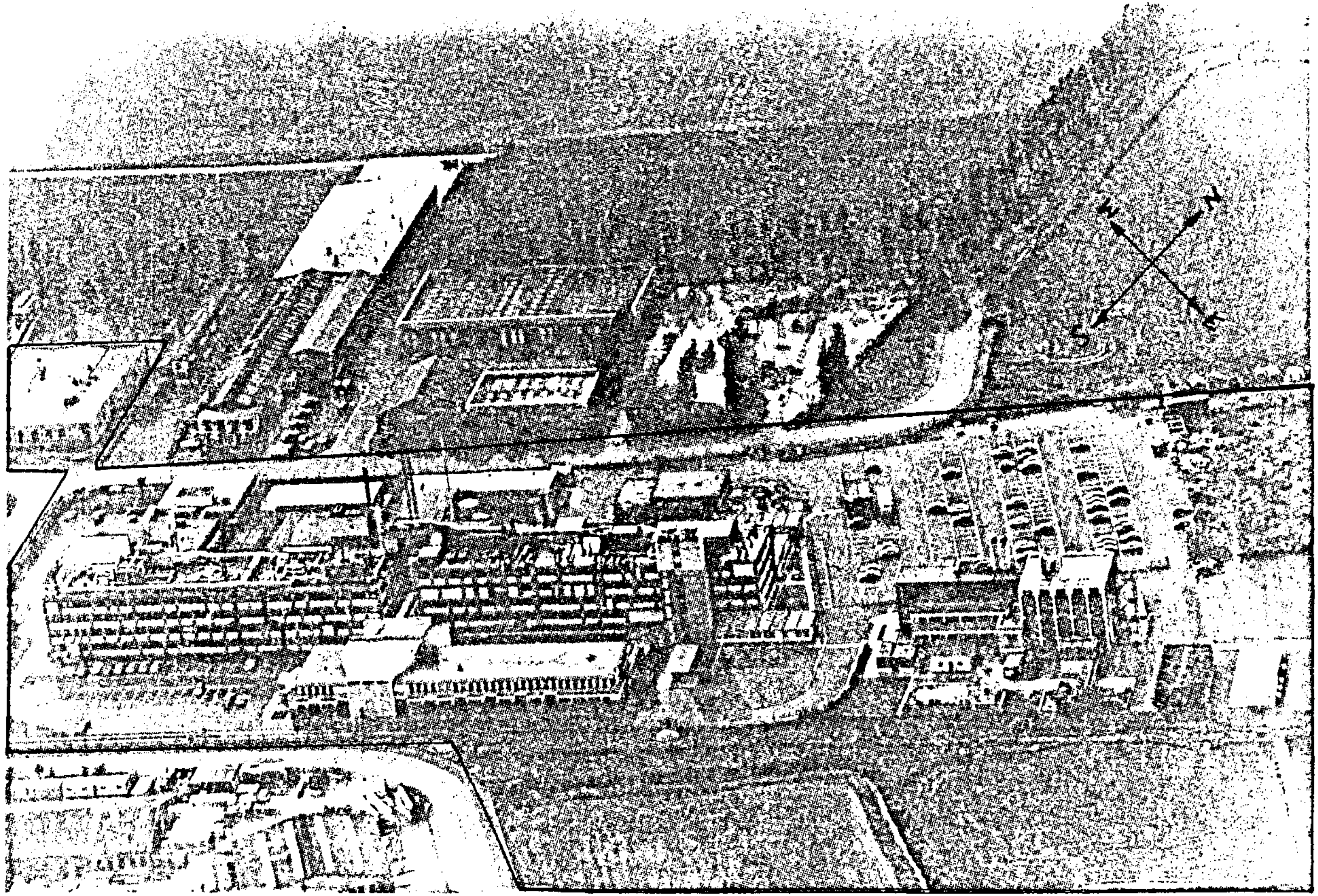


Fig. 4-4b : Aerial photograph of the company.



can be seen, that it consists of high and low rise buildings, and in addition it is situated in a relatively open country. Thus, it may be anticipated that the buildings are exposed to severe wind speed, consequently, high suction and high infiltration losses. This was found not to be the case, see chapter 6.

4.1.3 Pressure Induced Infiltration

Tests carried out by the Building Research Establishment, revealed that the rate of air flow is approximately proportional to the square root of the pressure difference and the IHVE Guide gives the following expression:

$$I = 0.827\,A\,(\Delta P)^{0.5} \quad \ldots\ldots\ldots \quad 4.5$$

Where: A = area of the openings, m^2

However, it is stated that in practice infiltration does not as a rule take place through a single opening, but that it is more likely that there will be a number of openings. Where the openings are in parallel, equation 4.5 is changed to the form:

$$I = 0.827\ \Sigma A\ (\Delta P)^{0.5} \quad \ldots\ldots\ldots \quad 4.6$$

and where the openings are in series the expression becomes:

$$I = 0.827\left[(A_1/A_2)/(A_1^{\,2} + A_2^{\,2})^{0.5}\right]\Delta P^{0.5} \quad \ldots\ldots\ldots\ldots \quad 4.7$$

Nevertheless, combining equations 4.3 and 4.5, it is possible to obtain an expression for the flow of air in terms of the wind speed. Thus

$$I = \text{constant} \times A \times V \quad \ldots\ldots\ldots\ldots \quad 4.8$$

The ASHRAE Handbook of fundamentals gives a similar equation, viz:

$$I = \varepsilon\,A\,V \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad 4.9$$

Where: ε = effectiveness of opening, having a value between 0.5 and 0.6 for the case where the wind is acting at right angles to the face of the opening.

However, equation 4.9 applies for the case where the area of inlet is the same as the area of the outlet. When they are not the same, the IHVE Guide gives the correction factors.

Basically equations 4.8 and 4.9 are similar to equations 4.1 and 4.2 as far as the wind speed is concerned, in the sense that they all state that the infiltration rate increases with wind speed (i.e. $I \propto V$).
The thermal load resulted from air infiltration is given by;

$$Q = C_p \rho I \Delta T \quad \ldots\ldots\ldots\ldots\ldots \quad 4.10$$

Where : C_p = specific heat capacity of air, kJ/kgK
ρ = density of air, kg/m^3
I = infiltration rate, m^3/s
ΔT = indoor - outdoor temperature difference, K

Hence, for a given room and temperature difference, the thermal load is proportional to the infiltration rate and thus, to the wind speed. Several other research workers have shown similar relationships with some minor modification. For example:
Jackman in 1969 (60) proposed that

$$I = C_L (\Delta P)^{1/n} \quad \ldots\ldots\ldots\ldots\ldots \quad 4.11$$

where:

C_L = kl = leakage factor
k = leakage coefficient, m^3/s per meter of gap length at ΔP of 1 N/m^2.
l = total gap length, m
$1/n$ = exponent $(1.4 \leqslant n \leqslant 1.6)$

The values of k varry with the type of the window fitting and are provided in reference 4 and 58.

Thus, when n = 1.4, the heat loss due to infiltration Q, varies exponentially with wind speed i.e. $Q \propto V^{1.43}$, and when n = 1.6 $Q \propto V^{1.25}$. This is demonstrated in table 4.1 for wind speeds of 3, 6 and 9 m/s relative to a speed of 2 m/s.

Jackman in 1970 (61) gave another equation

$$I = kl\ (\Delta P)^{0.625} \quad \ldots\ldots\ldots\ldots\ldots \quad 4.12$$

Equation 4.12 however, is included in the range given in equation 4.11.

Shaw in 1979 (103) suggested that:

$$I = C_j \; A_j \; (\Delta P_j)^{0.65} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 4.13$$

Which means that, $Q \propto V^{1.3}$ and this is within the range given in equation 4.11.

Tamura in 1979 (113) proposed another expression as follows:

$$I = 0.24 \; l_1/l_2 \; C_x x \; A \; F^{0.65} \; (e \; V_m)^{1.3} \quad \ldots\ldots\ldots \quad 4.14$$

Where:

F = pressure difference coefficient

l_1 = lenth of the wall facing front or back of house, m

l_2 = length of the side wall of house, m

A = total exterior wall area, m^2

C = exterior wall coefficient

e = Constant depending on location

V_m = Meteorological wind speed

For a particular building however, l_1, l_2, C_x and F are constant and thus equation 4.13 can be reduced to $C_i A \; (e \; V_m)^{1.3}$. If we could substitute the local wind speed for $e \, V_m$, we have $I = C_i \; A \; V^{1.3}$ which is similar to equation 4.12.

4.1.3.1 Quantitative Analysis of Heat Loss Due to Infiltration

Assume : A certain building in the East Midlands with a brick cavity wall, light weight plaster on internal surface. The building is a rectangular block with dimensions of 5 x 7 x 10 m. The U - value of such construction is approximately 1.37 W/m^2K (27) for the opaque fabric and 6.88 W/m^2K for the glazed area. The glazed area is said to be approximately 30% of the total external wall area. The crack length could be taken as 0.25 m/m^2 of the total external wall area.

We want to quantify the effect of wind speed on heat loss by using different theories, assuming wind speed of 2, 3, 6 and 9 m/s.

- The thermal load, from equation 4.10 is given by:

$$Q = C_p \rho I \Delta T$$

Assume:

- The average outside temperature $= 8\ ^\circ C$
- The average inside temperature $= 20\ ^\circ C$
- Design wind speed $\hat{=} 2$ m/s
- Air density at mean temperature = 1.23 kg/m^3 (93)
- Specific heat capacity at mean temperature = 1 kJ/kgK
- Amount of heat loss by conduction (assuming that the conduction heat does not vary with wind speed) is given by:

$$Q = \sum_{i=1}^{n} U_i A_i \Delta T$$

$$= U_g A_g \Delta T + U_{op} A_{op} \Delta T$$

$$= 6.88 \times 63 (20 - 8) + 1.37 \times 147 (20 - 8)$$

$$= 7.62 \text{ kw.}$$

(i) From equation 4.1

$$I = C_1 + C_2 V + C_3 \Delta T$$

Coblentz et al gave the values of constants as follows:

$$I = 0.15 + 0.013 V + 0.005 \Delta T$$

where: I = infiltration, Air change/hr

let $L = \dfrac{V\ I}{3600}$

Where: L = volumetric infiltration rate, m^3/s

V = volume of the heated building, m^3 substituting the known values

$$L = \frac{350\ I}{3600} = 0.097I$$

Leaving the temperature induced effect for the latter section,

$$Q = C_p \rho \ L \Delta T = 1 \times 1.23 \times L \times 12$$
$$= 1 \times 1.23 \times 12 \times 0.097\ (0.15 + 0.013\ \ )$$
$$= 0.215 + 0.019V \quad kW \ldots\ldots A$$

(ii) From equation 4.6

$$I = 0.827\ A\ \Delta P^{0.5}$$

and from equation 4.3

$$\Delta P = \frac{1}{2}\ C \rho V^2$$

thus, $I = 0.827\ A\ (\frac{1}{2} C \rho V^2)^{0.5}$

Where: A = area of the openings, m^2

C = pressure coefficient depending on the location and number of exposed sides. We will take a value of 0.8 for this particular case (61)

The crack length = 0.25 x 210 = 52,5 m

For demonstration purposes, we will assume an opening width of approximately 2 mm

then
$$A = 52.5 \times 0.002 = 0.105\ m^2$$
$$I = 0.827 \times 0.105\ (\frac{1}{2} \times 0.8 \times 1.23\ V^2)^{0.5}$$
$$= 0.061\ V\ \ m^3/s$$
$$Q = 1 \times 1.23 \times 12 \times 0.061\ V$$
$$= 0.90\ V \quad kW \ldots\ldots\ldots\ldots B$$

(iii) From equation 4.9

$$I = \varepsilon\ A\ V$$

Where : ε = effectiveness of opening having the values between 0.5 and 0.6 depending on the location of the building. We will take a value of 0.55 for our work.

thus, $I = 0.55 \times 0.105\ v = 0.058\ v$
$$Q = 1 \times 1.23 \times 12 \times 0.058\ v$$
$$= 0.86\ v \quad kW \ldots\ldots\ldots\ldots\ldots\ldots C$$

(iv) From equation 4.11

$$I = C_1 \ (\Delta P)^{1/n}$$

Where: C_1 = leakage factor, $m^3/h\ m\ mmH_2o$

Normally the value of C_1 vary from 0.1 $m^3/h\ m\ mmH_2o$ (for an excellently fitting window) to 15 $m^3/h\ m\ mmH_2o$ (for a worst fitting window) (61) However, for the type of building under consideration the window quality is generally somewhat better than average and C_1 values may be assumed to vary between 0.1 to 5 $m^3/h\ m\ mmH_2o$. For the illustration purposes a C value of 1 $m^3/h\ m\ mmH_2o$ is selected to represent a well fitting window.

Therefore, $I = 1\ (\frac{1}{2} \times 0.8 \times 1.23\ v^2)^{\frac{1}{n}}$

$$= (0.492\ v)^{\frac{1}{n}}$$

Where: $1.4 < n < 1.6$

thus, $0.642\ v^{1.25} < I < 0.603\ v^{1.43}$

and $Q = 1 \times 1.23 \times 12 \times 0.642\ v^{1.25}$

$$= 9.48v^{1.25} \quad \ldots\ldots\ldots\ldots \quad D(i)$$

$$Q = 1 \times 1.23 \times 12 \times 0.603\ V^{1.43}$$

$$= 8.9v^{1.43} \quad \ldots\ldots\ldots\ldots \quad D(ii)$$

Table 4.1 show the heat loss as influenced by the wind speed.

It should be noted that equations A, B, C and D were driven from experiments performed at different buildings with different leakage characteristics, and situated in different types of terrain. Thus, the different values of heat loss, when these equations are applied to the same building, should be anticipated.

However, the percentage heat consumption increase ranges between 7.5% and 75%, depending on the factors mentioned in section 4.1. Although we are not in position to state categorically the mathematical model (for air infiltration), suitable for this particular site, the possibility of the effect of wind speed being significant can not be ruled out.

TABLE 4 - 1 : Quantitative analysis of heat loss as influenced by the wind speed

Wind speed m/s	HEAT LOSS									
	EQUATION A		EQUATION B		EQUATION C		EQUATION D(i)		EQUATION D(ii)	
	Q kW	Increase %	Q kW	Increase %	Q kW	Increase %	Q kW	Increase %	Q kW	Increase %
2	0.253	Basis	2.00	Basis	1.72	Basis	22.5	Basis	23.98	Basis
3	0.272	7.5	2.70	35	2.58	50	37.43	66.0	41.82	74.4
6	0.329	30.0	5.40	170	5.16	200	89.02	294.8	115.38	381.2
9	0.386	52.6	8.1	305	7.74	350	147.78	555.3	206.04	759.2

4.1.4 Temperature - Induced Infiltration

The reason for the warm air rising is due to the fact that heated air is less dense than cold air, the variation in air density being inversely proportional to the absolute temperature of the air (13, 18, 54, 59, 60, 61).

Since the temperature inside a building is likely to be different from that ouside, this implies that the internal and external air will have different densities. Where there are openings in the building at different levels this variation in air density can produce a flow of air.

The general formula that governs the pressure drop is

$$\Delta P = (\rho_o - \rho_i)\ g\ Z \quad \ldots\ldots\ldots\ldots \quad 4.15$$

Where:

ρ_o = cold outside air density, kg/m^3

ρ_i = warm inside air density, kg/m^3

g = gravitational constant, m/s^2

Z = height between inlet and outlet, m

Equation 4.14 can be expressed in temperature form as; (80, 90, 92, 94, 95)

$$\Delta P = 3462\ Z\ \left(\frac{1}{T_o} - \frac{1}{T_i}\right) \quad \ldots\ldots\ldots \quad 4.16$$

Where : T_o and T_i = outside and inside temperatures respectively, K

4.1.5 Combined Wind and Stack Effect

As it was mentioned in section 4.1.1 when the wind and stack effect act together the resultant flow rate through the opening is not the sum of the two infiltration rates. The reason for this being that the two forces are acting contrary to one another at the upper level, on the windward faces, (i.e. while the wind forces the air inward through openings the stack effect produces an outward air movement), at the lower level of the building they are complimentary to each other as they both cause an inward air movement. Thus, it is widely accepted that the overall infiltration rate is generally governed by the effect with higher value when acting alone (61). Therefore at low wind speed when the stack effect is predominant, the rate of infiltration may be calculated on the basis of temperature difference alone. At high wind speeds only the effect of wind speed need to be considered.

We will make a further simplification in this work as follows; since company buildings are fitted with corridor/stairwell doors, to comply with fire regulations, which have a restricting effect on air flow between corridors and stairwell at each floor level, this will reduce considerably the infiltration caused by stack effect. Thus, we will assume that, the infiltration due to the action of the wind mainly on horizontal direction, will be substantially greater than that caused by the stack effect, and the combination of two motive forces, produces an infiltration rate approximately the same as that due to wind speed alone.

4.2 The Effect of Wind on Structural Heat Losses

Transmission losses occur from room air to and through exterior surfaces, by radiation, conduction and convection. The average rate of heat flow through an exposed building element is given by; (5, 9, 34, 76, 81, 93)

$$Q = U A \Delta T \qquad 4.17$$

Where : Q = rate of heat flow, W

U = overall heat transfer coefficient, W/m^2K

A = area of the element m^2

ΔT = indoor - outdoor temperature difference, K

The overall heat transfer coefficient for each element being calculated from:

$$U = \frac{1}{R_{si} + \sum R + R_{so}} \qquad 4.18$$

Where : $\sum R$ = sum of thermal resistances of the components of the building, $m^2\ K/W$

R_{si}, R_{so} = inside and outside thermal resistances respectively, m^2 K/W, given by

$$R_{si} = \frac{1}{1.2\ E_t h_{ri} + h_{ic}} \quad \ldots\ldots\ldots \quad 4.19$$

$$R_{so} = \frac{1}{E_t h_{ro} + h_{oc}} \quad \ldots\ldots\ldots\ldots \quad 4.20$$

Where:

E_t = long wave emissivity factor = 0.9 for normal building materials

h_{ri}, h_{ro} = inside and outside radiation coefficients, W/m^2K : $h_{ro} \triangleq 5.04 W/m^2K$ at a mean surface temperature of 8^oC.

h_{ic}, h_{oc} = inside and outside convection coefficients, W/m^2 K

The speed of the prevailing wind will affect the thermal resistance of the boundary layer at the outside surface, which in turn will affect the overall value of the transmittance coefficient thereby increasing the rate of heat loss from the structure for higher wind speed. Several formulae have been suggested for calculating h_{oc}, for example:-

Roux in 1959 (100) suggested that the natural convection heat transfer coefficient, h_{nc} can be determined from

$$h_{nc} = C\ (\Delta T)^{3/4} \quad \ldots\ldots\ldots \quad 4.21$$

Where : C = constant depending on direction of heat flow.

ΔT = temperature difference between outside surface and outside air, ^oC

and that, the forced convection, h_{fc}, can be given by

$$h_{fc} = \frac{0.2275\ C_p \rho\ v}{\ln\ (Re)^{2.58} (Pr)^{0.67}} \quad \ldots\ldots\ldots \quad 4.22$$

Where : C_p = specific heat capacity, kJ/kg K

Re = Reynold's number

Pr = Prandtl's number

He derived these equations from experiments on a flat plate and thus, these may not apply in practice as other factors have an influence on the heat transfer process. For example, it has been stated earlier, that the building shape and orientation have an influence to wind speeds and flow characteristics in its vicinity. On the windward faces the flow accelerates towards the corners while on the leeward side the flow pattern will be determined by the separation eddy. For these conditions, with strong pressure gradients and separated flows, the above theoretical equations do not apply.

On the windward faces, the impinging flow causes more heat transfer, but on the leeward faces, the mean speeds will be lower and it would be expected that there would be less heat transfer. At certain locations on the leeward face, however, turbulent fluctuations can be very large. Instanteneous high turbulent fluctuations close to the wall surface may produce mixing that will exceed the flat plate steady flow. Hütte in 1955 (55) proposed two relationships depending on the wind speed as follows:

$$h_{oc} = 6.2 + 4.2\,v \quad \text{for } v \leq 5 \text{ m/s} \quad \ldots\ldots\ldots\ldots \quad 4.23$$

$$h_{oc} = 7.5\,v^{0.8} \quad \text{for } v > 5 \text{ m/s} \quad \ldots\ldots\ldots\ldots \quad 4.24$$

Gerhart in 1967 (40) gave the following relationship

$$h_{oc} = 8.7 + 2.6\,v \quad \ldots\ldots\ldots\ldots\ldots \quad 4.25$$

Miller in 1978 (80), ASHRAE Hand book of Fundamentals and the IHVE Guide gave a relationship similar to 4.23 and 4.25 but with different coefficients i.e.

$$h_{oc} = 5.8 + 4.1\,v \quad \ldots\ldots\ldots\ldots\ldots \quad 4.26$$

The following numerical example illustrates the different effects (exerted by wind speed) on fabric heat loss, calculated by using different theoretical mathematical models above.

4.2.1 Quantitative Analysis of Effect of Wind Speed on Structural Heat Loss

Assume that we have the same building as in example 4.1.3.1.

Table 4 - 4 show the numerical variation of structural heat loss with wind speed. Equations 4.17 to 4.20 are used with

$h_{oc} = 8.7 + 2.6\ v$ . . . . A

$h_{oc} = 5.8 + 4.1\ v$ . . . . B

$h_{oc} = 6.2 + 4.2\ v$ for $v \leq 5$ m/s C(i)

$h_{oc} = 7.5\ v^{0.8}$ for $v > 5$ m/s C(ii)

and following values have been used

$E_1 = 0.9$ (80)

$h_{ri} = 5.7$ $W/m^2\,K$ for a mean temperature of 20 °C

$h_{ro} = 5.04$ $W/m^2\,K$ for a mean temperature of 8 °C

$h_{ic} = 3$ $W/m^2\,K$ for horizontal flow through walls where air movement does not, for practical purpose, occur over the surface (80).

Generally speaking equations 4.23 to 4.26 are more or less the same, as they all give almost similar structural heat, losses. The difference in constants being, probably, due to the difference in experimental set up and conditions. The percentage increase vary between 1.7 and 19.5%.

It can be concluded, therefore, that the effect of wind speed on both infiltration and structural heat loss may be significant depending on the type of building fabric, its height and location. And the thermal load imposed by wind speed increases with wind speed as follows:

$$Q \propto v^m \quad \text{where} \quad 1 \leq m \leq 1.43$$

However, we still have to find out, what form does heat loss take.

i.e. if $Q = C_1 \Delta T + C_2\, v^m$ (i.e. additive type)

or $Q = C_1 \Delta T\, (1 + C_2\, v^m)$ (i.e. multiplicative type)

Where: $C_1 \Delta T$ = the component of heat flow due to inside - outside temperature difference (degree days)

$C_2 v^m$ = the component of heat flow imposed by wind speed.

TABLE 4 - 2 : Variation of structural heat loss with wind speed

Wind Speed m/s	R_{so}, m^2 K/W				R_{si} m^2 K/W	R m^2 K/W	U_{opaque}, W/m^2 K				U_{glass} W/m^2 K	Heat Loss							
	Eq. A	Eq. B	Eq. C (i)	Eq. C (ii)			Eq. A	Eq. B	Eq. C (i)	Eq. C (ii)		Equation A		Equation B		Equation C (i)		Equation C (ii)	
												Q,kW	Incre-ase,%	Q,kW	Incre-ase,%	Q,kW	Incre-ase,%	Q,kW	Incre-ase,%
2	0.054	0.054	0.052	-	0.109	0.73	1.120	1.120	1.120		6.10	6.587	Basis	6.587	Basis	6.791	Basis	-	-
3	0.048	0.044	0.043	-	0.109	0.73	1.127	1.133	1.134		6.49	6.894	4.7	6.905	4.8	6.907	1.7	-	-
6	0.035	0.029	-	0.028	0.109	0.73	1.144	1.152	-	1.153	7.25	7.499	13.8	7.513	14.1	-	-	7.515	10.7
9	0.027	0.021	-	0.021	0.109	0.73	1.155	1.163	-	1.169	7.69	7.851	19.2	7.865	19.4	-	-	7.865	15.8

CHAPTER 5

THE EFFECT OF SOLAR RADIATION UPON HEAT REQUIREMENTS

Modern buildings, very often have large areas of glazing and use lightweight construction materials. The latter warms up quickly while the former admits large solar heat gains, particularly so for days of highest solar radiation. This type of construction, apart from causing summer overheating also has a two fold contradicting effects on the assessment of energy consumption during heating periods, viz:

(a) Although the winter solar radiation is generally small, it admits solar heat gains which can influence the quantity of energy used for space heating.

(b) It also increase the overall heat transfer coefficient for the building, due to its low thermal resistance, and hence increases structural heat losses.

Comparatively, the vast majority of the heating effect is due to the solar heat gain through glazed window areas on each wall of the building rather than the building fabric (72, 74, 112). But, in this chapter, we will consider both components (i.e. solar heat gain through glazing and that through solid fabric). The solar heat gain through a building structure depends primarily upon the intensity of the sunrays, which in turn depends upon the angle of incidence of the sun's rays to the particular facade. Since the sun changes its position continuously throughout the day the intensity of the sun rays on any particular facade is variable. In addition there is also the variation of height of the sun for different months of the year and the duration of the sunshine. These factors makes the heat gain quantification a bit complicated.

Before quantifying the heat gain due to solar radiation for space heating, an understanding of the mechanism governing the heat flow through the wall is essential.

5.1 Influence of Solar Radiation

Before considering how the building reacts to solar radiation, it is logical to take into account the manner in which the radiation will enter the building. Radiation striking a surface may be :

(a) Absorbed

(b) Reflected

(c) Transmitted

If α = proportion of radiation absorbed

γ = proportion of radiation reflected

τ = proportion of radiation transmitted, then

$$\alpha + \gamma + \tau = 1.0 \quad \ldots\ldots\ldots\ldots\ldots \quad 5.1$$

The values of α, γ and τ depend upon the physical properties of the surface receiving the radiation, the wavelength of the radiation and the angle of incidence of the sun rays (4, 8, 9, 25, 37, 58).

5.1.1 Absorption

A body which is a perfect emitter of radiation is termed as a "black body". A body which absorbs radiation will also emit radiation and the relationship between the radiation and the wavelength emitted by a blackbody is shown in Fig. 5 - 1 (93). The area under the curve is equal to the total power radiated, and can be expressed by Stefan Boltzman's law, as;

$$P_o = \sigma A T^4 \quad \ldots\ldots\ldots\ldots\ldots \quad 5.2$$

Where; A = area, m^2

T = absolute temperature, K

σ = Stefan Boltzman's constant = $5.7 \times 10^{-8} W/m^2 K^4$.

It can be seen from the curves that, the wavelength at which the maximum radiation occurs, changes with temperature rises. Hence a high temperature source will radiate at a short wavelength and conversely a low - temperature source will radiate at a long wavelength.

5.1.2 Emissivity

At a given wavelength, the maximum amount of radiation which can be emitted will be that from a black body i.e. emissivity of a black body = 1.

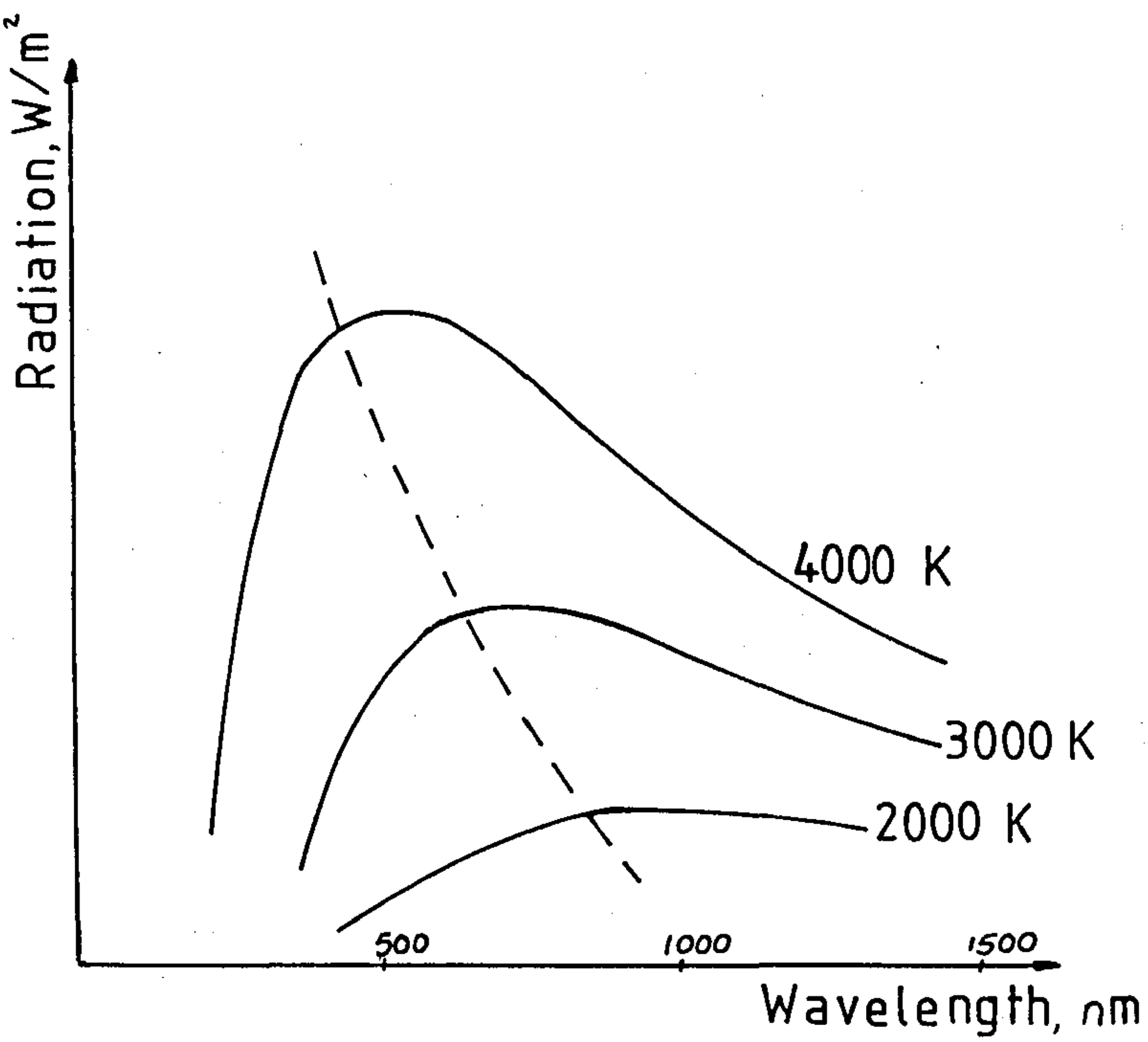


Fig. 5-1: Spectoral distribution of black-body radiation (112).

It follows that a non-black body will have an emissivity less than one, thus emissivity of any body at a particular wavelength is given by;

$$E_\lambda = \frac{\text{Total power emitted per unit area of body}}{\text{Total power emitted per unit area of a black body}}$$

Within the temperature range normally met in practice most building materials have a high emissivity i.e. 0.9 to 0.95, whereas polished metals have a low emissivity, e.g. polished Aluminium has a value of 0.05 (4, 58, 93).

Two important aspects regarding wavelength have been mentioned, viz:

(a) The values of the absorptivity and emissivity are influenced by the wavelength.

(b) The wavelength at the peak radiation depends upon the temperature of the radiation source.

Thus, the sun, being a high-temperature source will emit radiation in the short wave band, while a building being a low-temperature source, will radiate at a long wavelength.

5.1.3 Sol-air Temperature

When the surfaces of a building are subjected to solar radiation, a rise in internal temperature is produced. A similar rise in internal temperature could occur if there was no solar radiation but if the external air temperature was increased. This increased external temperature which is producing the same internal temperature rise as was obtained with solar radiation acting in conjuction with actual external air temperature is termed as the "sol-air temperature" (4, 8, 9, 72, 73, 85).

Let, the heat flow at a surface fabric due to temperature difference per unit area be given by (8, 66, 72);

$$Q_1 = \frac{1}{R_{so}} (T_{ao} - T_{so}) \quad \ldots\ldots\ldots\ldots \quad 5.3$$

The solar radiation absorbed by fabric per unit area be given by (72, 79);

$$Q_2 = a\, I_G \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 5.4$$

the heat loss by long - wave re-radiation per unit area of the building fabric be given by(72, 79);

$$Q_3 = E\ I_L \quad \ldots\ldots\ldots\ldots\ldots \quad 5.4$$

then, the net heat flow per unit area of the building fabric at the surface, is given by;

$$Q = \frac{1}{R_{so}} (T_{ao} - T_{so}) + a\ I_G - E\ I_L \quad \ldots\ldots \quad 5.5$$

Where:

R_{so} = external surface resistance, m^2K/W

T_{ao} = external air temperature, K

T_{so} = external surface temperature, K

a = solar absorptivity of surface

I_G = global solar irradiance on surface fabric, W/m^2

E = emissivity

I_L = long - wave radiation, W/m^2

But the net heat flow per unit area is equal to the heat flow due to sol-air temperature (according to the definition) thus,

$$Q = \frac{1}{R_{so}} (T_{ei} - T_{eo}) \quad \ldots\ldots\ldots\ldots \quad 5.6$$

It can be seen that the first term in equation 5.5 depends upon the prevailing wind speed, since R_{so} decreases with increasing wind speed, see section 4.2. Thus the heat flow into the building due to sol-air temperature increase with wind speed.

The IHVE Guide (58), gives a value of 100 W/m^2 for the long-wave radiation from horizontal roof at a clear sky. Since we are interested in winter periods when there are clouds and precipitation almost throughout the day, it is anticipated that this value will be lower, besides, for a multistorey building the outer roof area is small compared to the external wall area.

In the case of verticle surfaces, the Guide states that $E\ I_L$ can be taken as zero, since it is assumed that the long-wave radiation which a wall emits is approximately balanced by that which it receives from the ground.

The sol-air temperature on a vertical surface consists of two components, viz:

(a) External air temperature, T_{ao}, which is independent of orientation, with a peak value in afternoon.

(b) Temperature increase over external temperature due to solar radiation = $R_{so}\ I_{GV}$, where the time of the peak depends upon orientation.

These two components suggests that the time when the maximum value of the sol-air temperature occurs on a particular orientation may not be the same as that for maximum external air temperature.

Since the effect of sol-air temperature is a combination of the effects of solar radiation and indoor - outdoor temperature difference, it should be borne in mind, that this effect is dealt with, indirectly, by taking into account the effects of solar radiation and indoor - outdoor temperature difference, hereafter.

5.1.4 Heat Flow Through Building Fabric Due to Solar Radiation

For steady state conditions i.e. the situation where the temperature difference between the internal and external temperature is constant, the rate of heat flow through a wall is given by (66, 93);

$$Q/_A = U\ (T_{ei} - T_{ao}) \ldots\ldots\ldots\ldots \qquad 5.7$$

Where:

T_{ao} = the external air temperature, ^{o}C

The external air temperature, however, does not remain constant; on contrary, it varies with time, hence the temperature difference varies with time.

The time lag between the instant the external air temperature changes to the instant when this change has been transmitted through the fabric will depend upon the type of structure. For a heavy weight building, any change in outside temperature take a long time to effect the inside temperature. Fig. 5 - 2, show a hypothetical variation of internal temperature due to the change in external temperature. It can be seen from Fig. 5 - 2 (b) that for the heavy weight construction the time lag for an outside change to effect the internal temperature is increased but

Fig.5-2 :Effect of (a) lightweight (b) heavyweight structu on internal temp. variations.

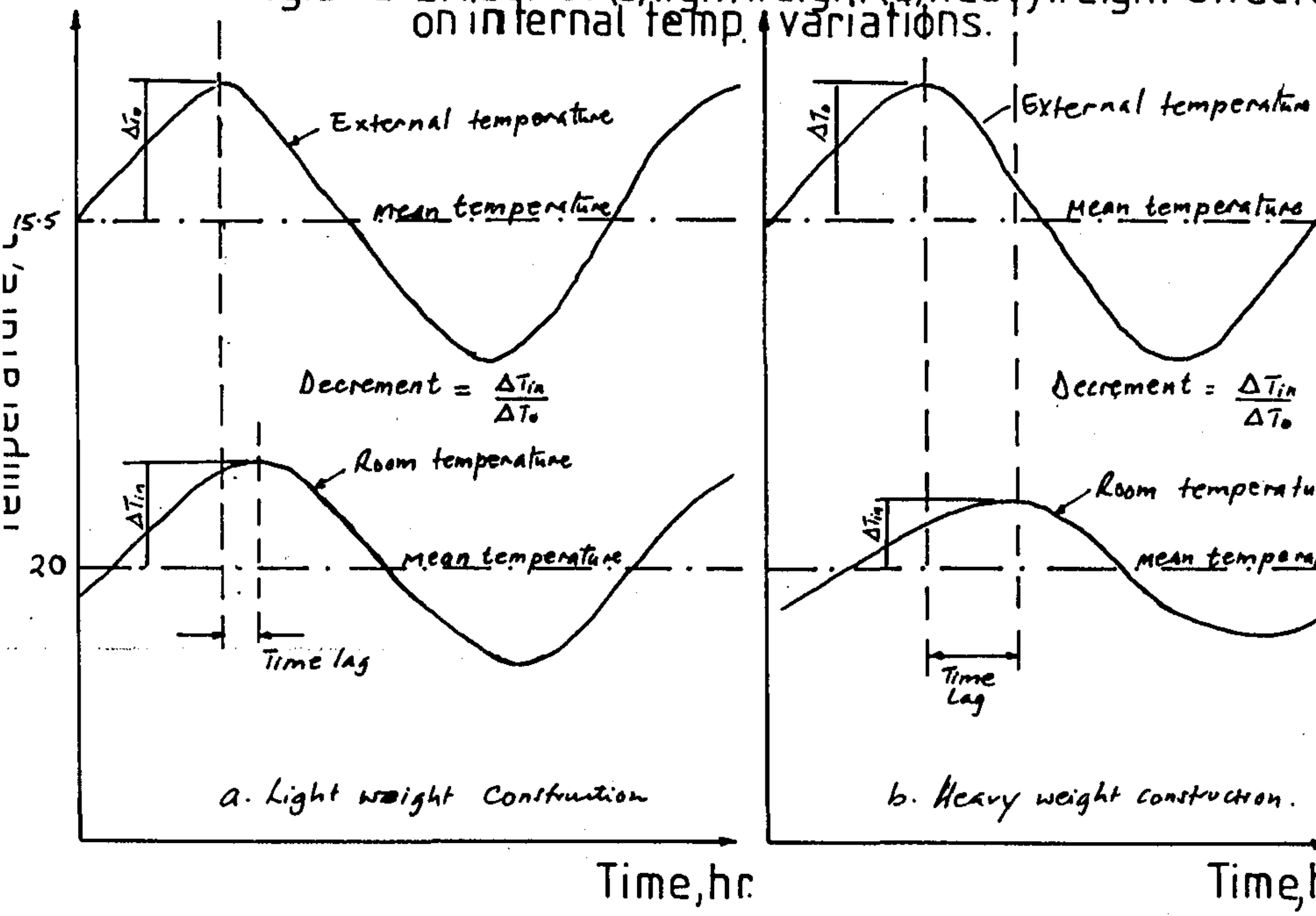
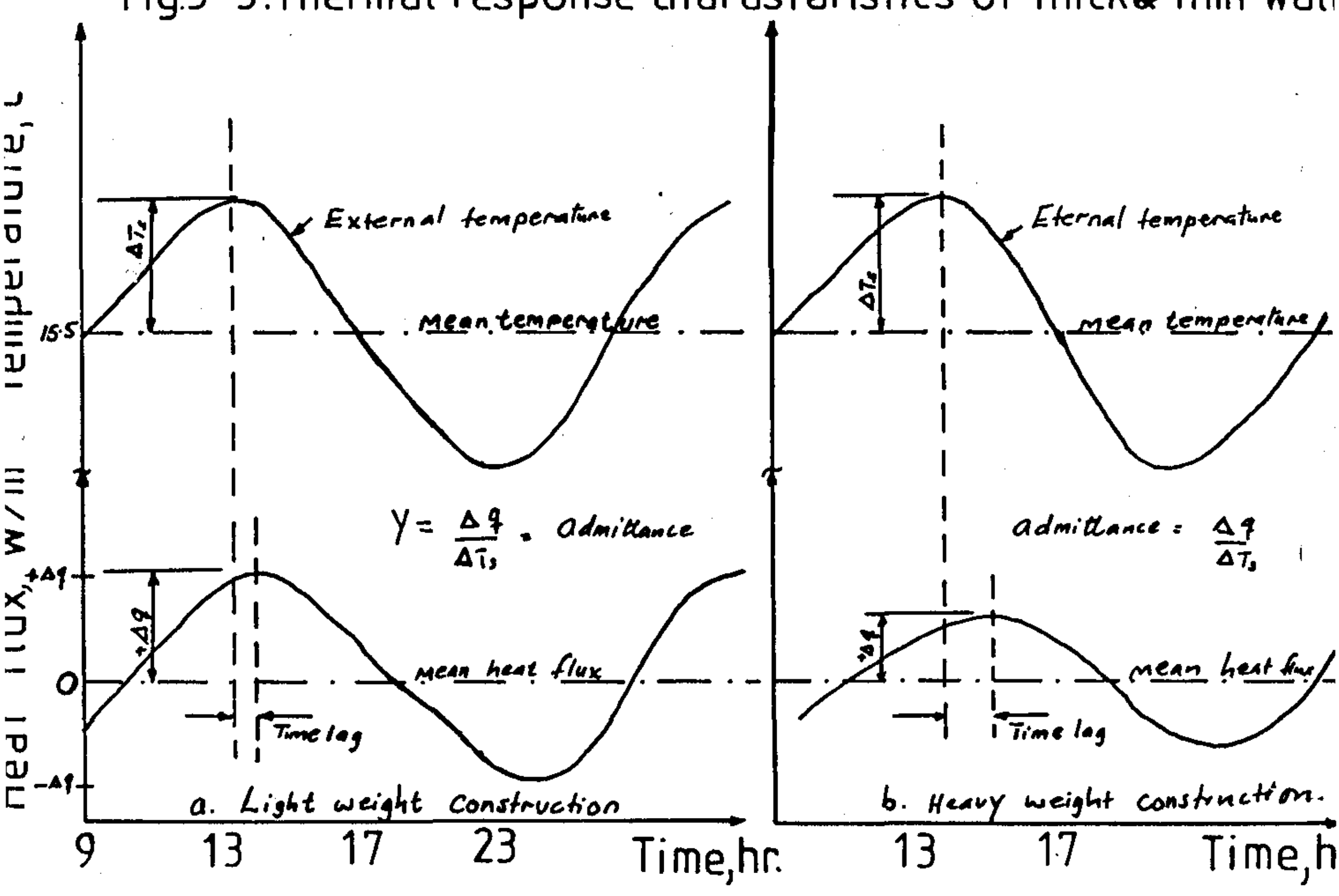


Fig.5-3:Thermal response charastaristics of thick& thin wall



the amplitude or "swing about the mean internal temperature" is reduced or damped. This is termed as "decrement". This increase in time lag and damping effect is related to heat storage effect of the building fabric as it will be seen later.

The ability to store heat is very important when considering situations where there are variations in temperature difference or changes in heat input, and it has to be taken into account when evaluating the rate of heat flow due to solar radiation. This gives rise to the concept of thermal admittance.

The property of thermal admittance of a wall is a measure of the ability of the wall to absorb and store heat during one part of cycle (day time) and then release the heat back through the same surface during the second part of the cycle (night time) (25, 72, 73, 74, 79). Thus it is coupled to the cyclic nature of the give and take of heat at the surface of the wall. Technically the thermal admittance is the ratio of the amplitude of sinusoidal wave of heat flow to the amplitude of the corresponding sinusoidal wave of surface temperature (79).

Thus,

$$Y = \frac{\Delta q}{\Delta T_s} \qquad \text{5.8}$$

Where;

Δq = amplitude of sinusoidal wave heat flow, W/m^2

ΔT_s = amplitude of sinusoidal wave of surface temperature, K

Y = admittance, W/m^2 K.

The factors influencing the admittance value of a particular building are thermal diffusivity ($^k/C_p \rho$) and the thickness of the building materials, see equation 5.10 - 5.16.

Fig.5 - 3 show the thermal response characteristics of building construction and it can be seen that the magnitude of temperature cycle is $\Delta T_s/2$ so that the wall surface temperature varies from $\bar{T}_s + \Delta T_s/2$ to $\bar{T}_s - \Delta T_s/2$ (where $\bar{T}_s$ = Mean surface temperature). And the magnitude of the heat flow is Δq so that the heat flow varies from $+ \Delta q/2$ (into the wall) to $-\Delta q/2$ (out of the wall). It should be noted that the two sinusoidal waves are slightly out of phase. Typically the temperature wave leads the heat wave

by one eighth of a cycle (45 angular degrees) for a thick wall and one fourth of a cycle (90 angular degrees) for a very thin wall (72, 73, 74, 79).

Generally we are most concerned with cyclic oscillations which have a period of 24 hours (i.e. charging the wall during the day and a discharging at night). The fact that the wave is not trully sinusoidal is considered not to be of great consequence. Very often, First order effects have been assumed by considering pure sine waves (25, 72, 73, 74, 79), even though, the cyclic heat gain are still comlicated to quantify as it is illustrated in the following example.

Example

Consider the solar radiation falling on a flat roof, one way of quantifying the cyclic heat gain is as follows:

Assume;

(1) The construction materials are isotropic and have physical properties (i.e. k, ρ, C_p) that are independent of temperature.

(2) The absorption of radiant energy below the surface, may be characterized by a Lambert's decay - law expression of the form (93);

$$I_{GX} = I_{GO}\, e^{-\mu X}, \quad W/m^2 \qquad \ldots\ldots\ldots\ldots \qquad 5.9$$

Where; I_{GX} = intensity at any depth X, W/m^2

I_{GO} = intensity of unreflected radiation at surface, W/m^2

μ = decrement factor

(3) The initial temperature of surface is T_i

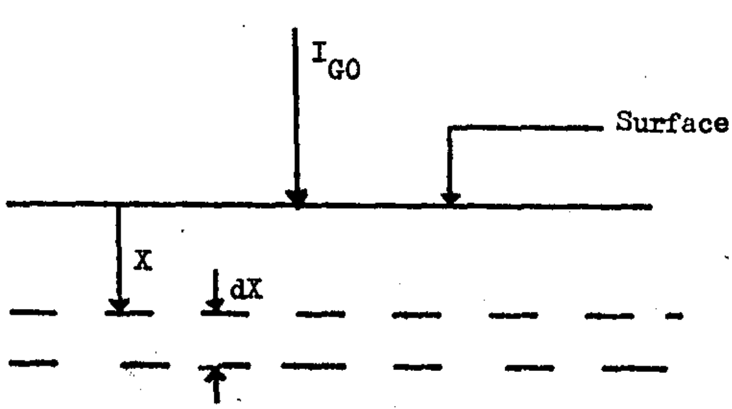


Fig. 5 - 4 Solar heating of a flat roof

Consider a section of thickness dX and of unit area located X mm from the surface of the roof. Applying the first law of thermodynamics:

$$\text{Rate of energy input} = -k\,\frac{\delta T}{\delta X} + I_{GX} \quad \ldots\ldots \quad 5.10$$

Rate of energy output

$$= -k\left(\frac{\delta T}{\delta X} + \frac{\delta^2 T}{\delta X^2}\,dX\right) + \left(I_{GX} + \frac{\delta I_{GX}}{\delta X}\,dX\right) \quad \ldots \quad 5.11$$

$$\text{Rate of energy accumulation} = \frac{\delta}{\delta t}\left(c_p T\, dX\right) \quad \ldots\ldots \quad 5.12$$

The applicable differential equation is

$$\nu\,\frac{\delta^2 \theta}{\delta X^2} + \beta\,\bar{e}^{\mu X} = \frac{\delta \theta}{\delta t} \quad \ldots\ldots\ldots\ldots \quad 5.13$$

Where;

$$\nu = \frac{k}{\rho c_p} = \text{diffusivity} \quad \ldots\ldots\ldots\ldots \quad 5.14$$

$$\beta = \frac{\mu I_{GO}}{\rho c_p} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 5.15$$

$$\theta = T - T_0 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 5.16$$

For simplification, following boundary conditions are assumed:

(i) $\theta = 0$ at $t = 0$ for all X

(ii) $\theta = 0$ at $X = \infty$ for all X thus,

$$\frac{\delta \theta}{\delta X} = 0 \quad \text{at } X = 0 \text{ for all } t.$$

The third boundary condition results from the following considerations. Examine a slice of material of infinitesimal thickness at the surface of the roof Fig. 5.5

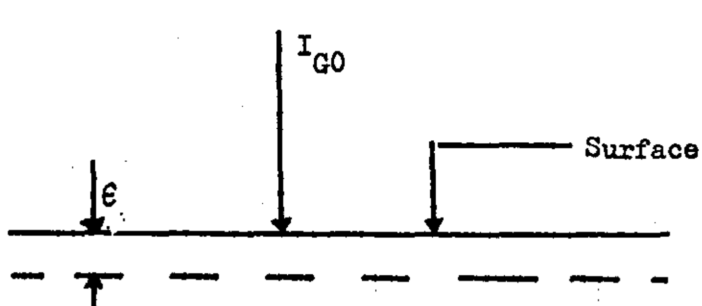


Fig. 5 - 5 Boundary condition at solid surface

Rate of energy input $= I_{GO}$ 5.17

Rate of energy output $= -k(\frac{\delta I}{\delta X})_{X=\epsilon} + I_{GO}\bar{e}^{\mu\epsilon}$ 5.18

Rate of energy accumulation $= \frac{\delta}{\delta\epsilon}(\rho\,\epsilon\,C\,T)$ 5.19

Thus

$$(\frac{\delta T}{\delta X})_{X=\epsilon} = I_{GO}(\bar{e}^{\mu\epsilon} - 1) + \rho\,C\,\epsilon\,\frac{\delta T}{\delta t} \quad \ldots \quad 5.20$$

Now let $\epsilon \longrightarrow 0$ corresponding to $X \longrightarrow 0$ then the right hand side of equation 5.20 vanishes and there results

$$(\frac{\delta T}{\delta X})_{X=0} = 0 \quad \ldots\ldots\ldots \quad 5.21$$

Taking the Laplace transform of equation 5.13 we have

$$\nu\int_0^{\infty}\bar{e}^{pt}\frac{\delta^2\theta}{\delta X^2}\,dt + \beta\int_0^{\infty}\bar{e}^{pt}\,e^{\mu X}\,dt = \int_0^{\infty}\bar{e}^{pt}\frac{\delta\theta}{\delta t}\,dt \quad \ldots \quad 5.22$$

equation 5.22 may be integrated if it is noted that θ (i.e. $T - T_0$) is a function of two independent variables, namely, thickness and time of oscilation and that the limits of integration are independent of X.

These factors complicates the whole exercise of quantifying the solar heat gain. Besides the anticipated outcome makes equation 5.22 less attractive in industrial applications where quick and simple estimation procedure is needed . Thus, for demonstration purposes a simplified method outlined in reference 9 will be adopted. The method involves multiplying the solar intensity by an absorpttance of the building material. However, the main handicap of this method, like many other methods, is that, it does not take into account the fact that the solar intensity is varying with time of the day. Otherwise, if we assume an average insolation, I_G, the heat gain is given by:

$$Q = a\,I_G \quad \ldots\ldots\ldots\ldots \quad 5.23$$

Where; a = absorption coefficient ranging from 0.2 (for a white wall) to 0.8 (for a dark wall).

I_G = insolation, W/m^2

The absorption of solar energy on the outer wall of a building increases the average wall temperature given by (9);

$$\Delta T_s = \frac{a\, I_G}{h_{oc}+U} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 5.24$$

Where;

h_{oc} = external heat transfer coefficient (14 $W/m^2 K$ for an average wind speed of 2 m/s, see section 4.2)

U = U-value of the building wall, $W/m^2 K$

As long as the outside temperature remains to be lower than the room temperature, this temperature increase reduces thermal losses which are normally calculated on air temperature basis. Consequently heating demand of a building will decrease relatively. See numerical example at the end of the chapter.

5.1.5 Transmission of Solar Radiation Through Glass

Glass transmits radiation within the range of 300 to 2800 nm. Radiation emitted by the sun and received by the glass is of short wavelength and the glass is transparrent at this wavelength, and the radiation is transmitted into the interior of the structure. This energy is absorbed by the surfaces, inside the building which in turn raise their temperature and become low-temperature emitters. The radiation emitted by these surfaces will be of long wavelength to which the glass is opaque and as a consequence this energy is contained within the structure with a corresponding rise in temperature (72, 73, 74). It is this effect which can give rise to overheating in summer and lower energy consumption for space heating in winter, due to solar heat gains. This depends upon the transmission characteristics of the glass.

Transmission characteristics of glass

As it has been mentioned earlier, the radiant energy striking a surface could be transmitted, absorbed or reflected. Thus,

(a) The energy transmitted through a glass at any wavelength = (incident solar energy on glass) x (transmission coefficient)

$= \tau A I_G$

(b) The energy reflected by the glass

= (incident solar energy on glass) x (reflection coefficient).

The reflection coefficient changes with angle of incidence as illustrated in Fig. 5 - 6. The angle of incidence in turn depends upon the position and height of the sun in the sky. The position of the sun can be determined from a knowledge of the solar altitude and azimuth angles as potrayed in Fig. 5 - 7 and Fig. 5 - 8(by definition, the solar altitude is the angle in a vertical plane between the earth's surface and the sun's rays, whilst, the solar azimuth is that angle between the vertical plane of the sun's rays and due north).

During mid winter periods the solar altitude for the entire U.K. ranges between 6^{o} to 11^{o} thus, the radiation intensity is relatively low. From Fig. 5 - 6 the reflection coefficient is approximately 0.08, this means the reflection losses are about 8%. The absorption losses, (caused by the impurities in the glass mostly iron oxide), correspond to about 5 - 10% for normal, single pane window glass (90). Therefore the remaining 82 - 87% is transmitted to the building. However, the absorbed part is not lost as such, since it increases the window temperature and thereby reducing heat loss by conduction through the window glass. See the numerical example at the end of the chapter.

The solar heat gain therefore, is given by;

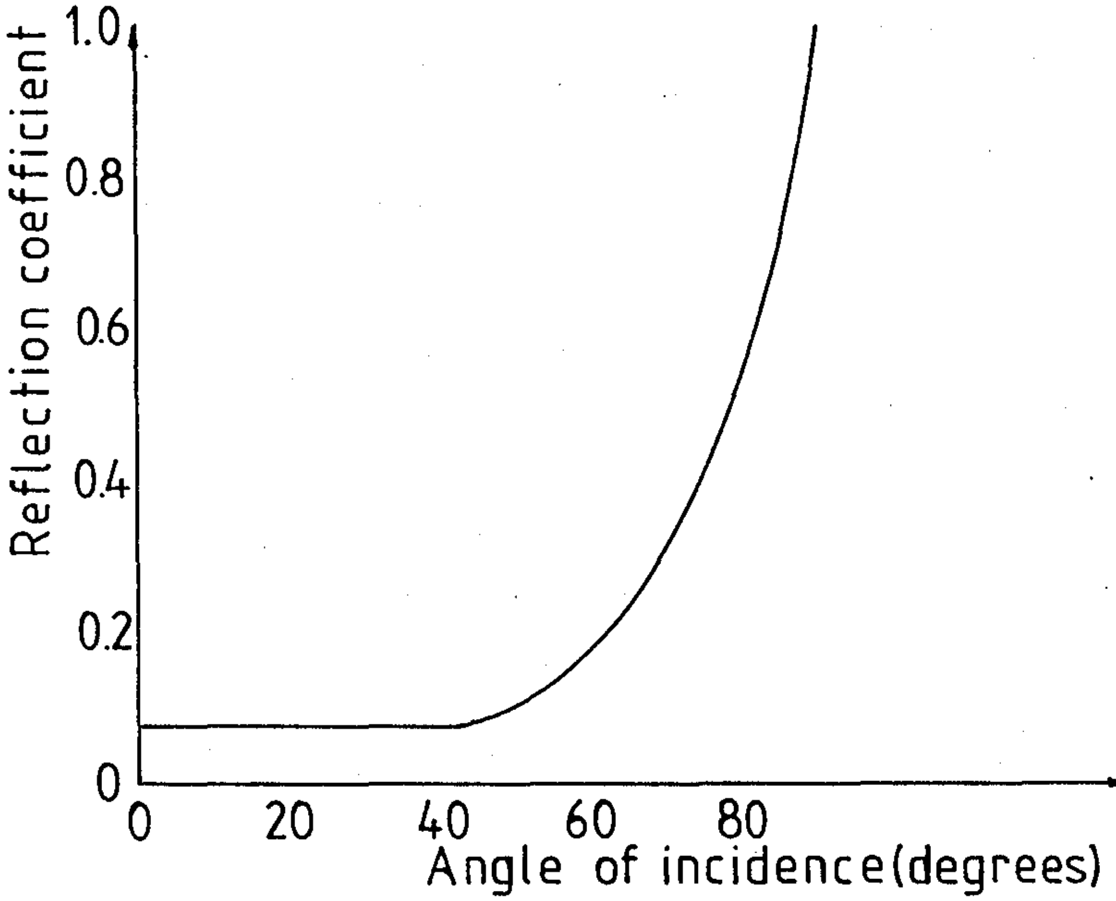
$$Q = \tau_g I_G A_g \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 5.25$$

Where; τ_g = window glass transmittance

A_g = area of glazing

Equation 5.25 gives a mean solar heat gain through the glazed area, however, if a more accurate value is needed, a method given by Pentherbridge (90)

Fig.5 -6 : The change of reflection coefficient with angle of incidence.(90)



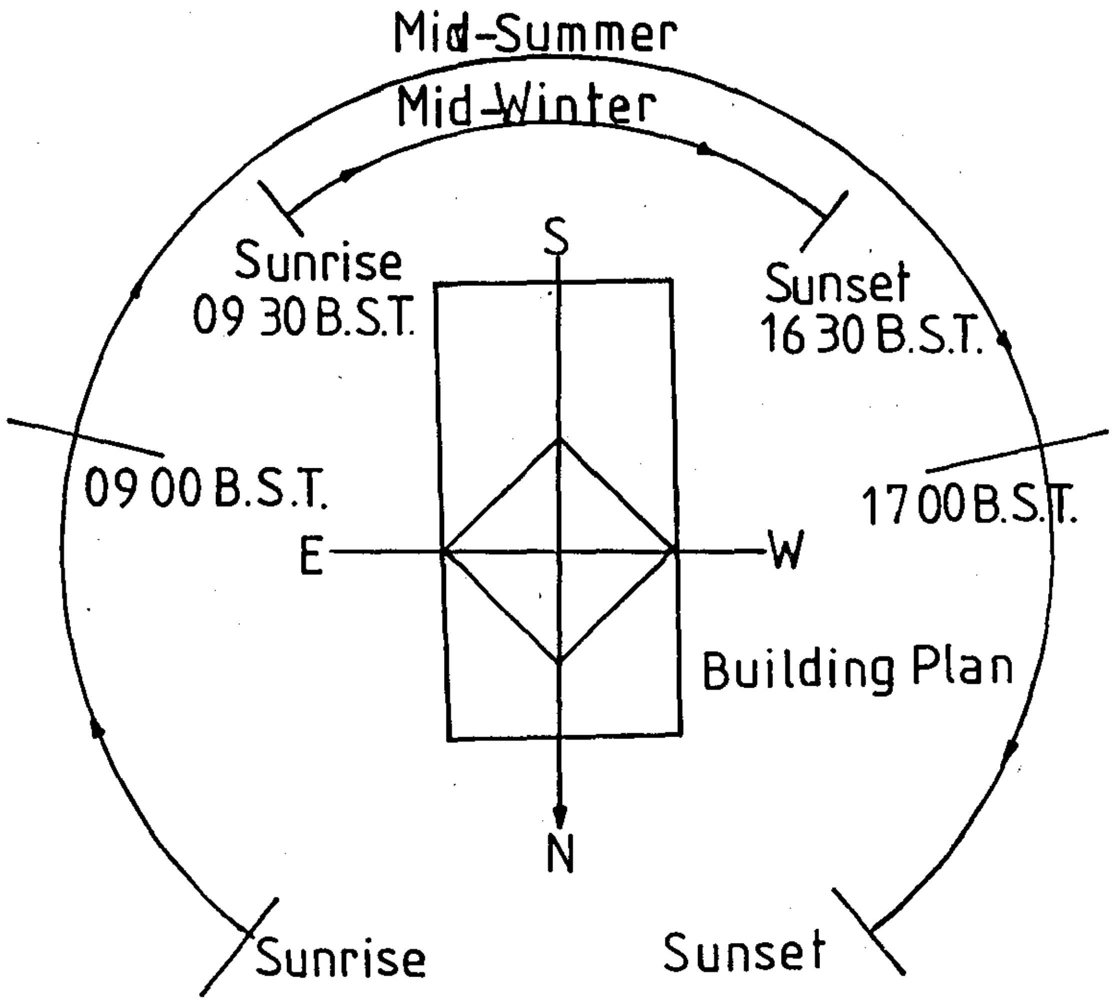


Fig.5-7: Solar Azimuth (90).

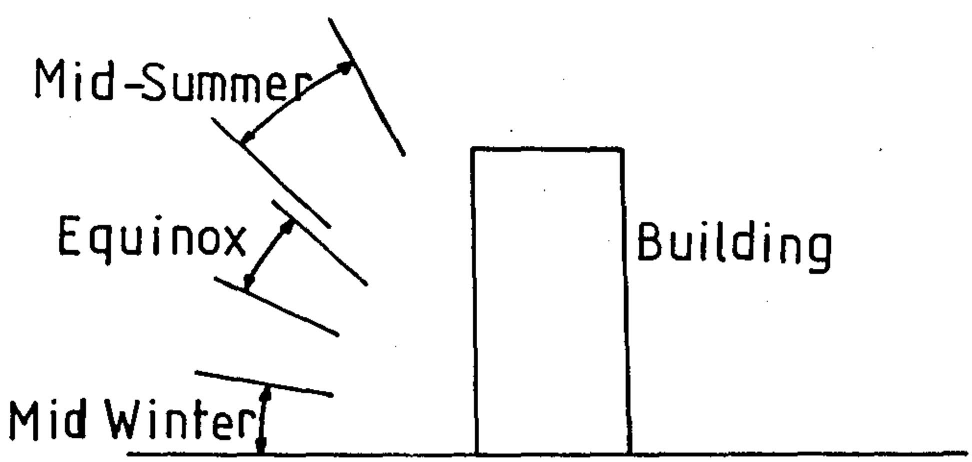


Fig. 5-8: Solar Altitude (90).

could be used. Pentherbridge gave a series of overlays which when used in conjunction with the stereographic sunpath projections, the hourly and daily total transmitted radiation can be computed. The procedure and use of these overlays are fully described in reference 90.

5.1.6 Numerical Example

Consider a certain building in the East midlands, having a floor area of about 200 m^2 and a volume of 1400 m^3. The window area is 103 m^2 which correspond to 30% of the outer wall area. The wall is made of a brick - brick cavity with lightweight plaster on internal surface. A typical monthly insolation (in a particular year) falling on 1-m^2 area of a wall (at different orientation) in a particular location, is shown in Fig. 5 - 9. We want to quantify the solar heat gains and their resultant effect on the energy used for space heating.

Solution:

From Fig. 5 - 9 the average monthly insolation during heating season are:

Southward wall	=	38 kWh/month.m^2 (158 W/m^2)
Westward wall	=	17.5 kWh/month.m^2 (73 W/m^2)
Northward wall	=	14.3 kWh/month.m^2 (53 W/m^2)
Eastward wall	=	24.3 kWh/month.m^2 (101 W/m^2)

The values in brackets have been worked out by assuming a period of 9 hours of sun shine per day.

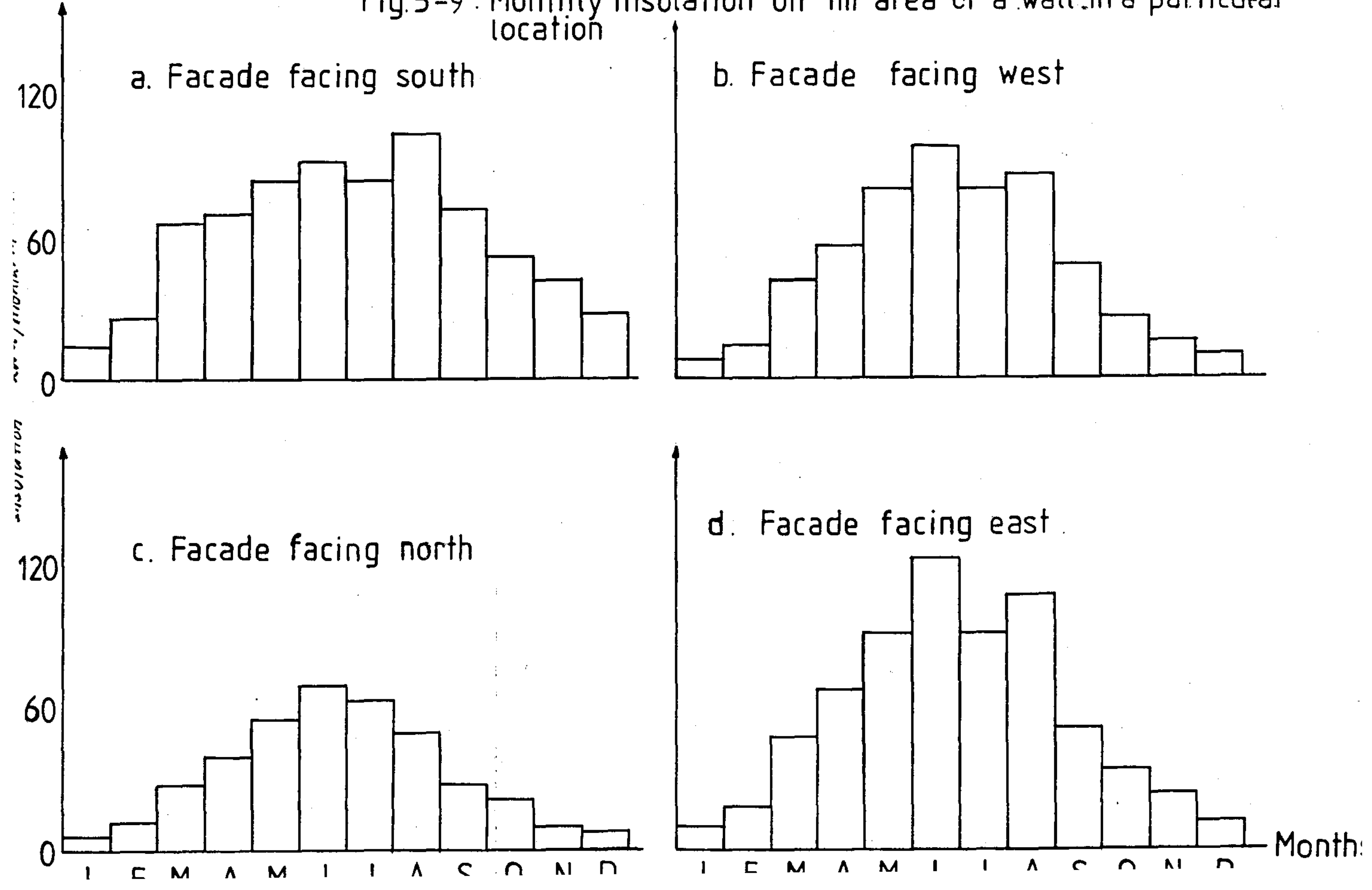
Assume:

- The average room temperature = 20 °C
- The average outside temperature = 8 °C
- The U-value for such construction = 1.37 W/m^2K
- The absorption coefficient of a solid wall = 0.5
- The transmittance coefficient of the glass = 0.85
- The absorption coefficient of window glass = 0.075

The total heat loss through solid fabric without taking into account the solar heat gains is given by;

$$Q/_A = U \Delta T = 1.37\,(20 - 8) = 16.44 \ W/_{m^2}$$

Fig.5-9: Monthly insolation on 1m² area of a wall in a particular location



And that through the glazed area = 6.10 (20 - 8) = 73.2 W/m^2

Where: 6.10 W/m^2 is the U-value of the window glass at an average speed of 2 m/s.

The total heat loss = 89.64 W/m^2.

The amount of heat loss through the building fabric after taking into account the effect of solar radiation is shown in table 5 - 1. It can be seen that the heat loss from the building is reduced by about 15.4% as a result of this effect. On the other hand the solar heat gains through the windows can be computed from equation 5.25 as follows;

$$Q/_A = \tau I_G$$

Thus; Southward wall, Q = 0.85 x 158 = 134.30 W/m^2
Westward wall, Q = 0.85 x 73 = 62.05 W/m^2
Northward wall, Q = 0.85 x 53 = 45.05 W/m^2
Eastward wall, Q = 0.85 x 101 = 85.85 w/m^2

Average heat gain per square metre of glazing = $\frac{134.30 + 62.05 + 45.05 + 85.85}{4}$

= 81.8 W/m^2(271 l/week assuming 9 hrs. of sun shine per day)

This value (which has been calculated after a lot of simplification), is around 1.1 times the heat loss, which is quite big. It is anticipated that in actual practice the amount of heat gain will be lower than this value. However, the example indicates that insolation on an external wall may have a considerable influence on the heating requirement of a building.

Table 5 - 1 : Heat loss from the building after taking into account the solar heat gains.

	Southward Wall				Westward Wall				Northward Wall				Eastward Wall			
	ΔT_s °C $\left(\frac{aI}{h_o+U}\right)$	T °C $(T_o+\Delta T_s)$	Heat loss W/m² $(U(T_{in}-T)$	Decrease in heat loss, %	ΔT_s °C	T °C	Heat loss W/m²	Decrease in heat loss %	ΔT_s °C	T °C	Heat loss W/m²	Decrease in heat loss %	ΔT_s °C	T °C	Heat loss W/m²	Decrease in heat loss %
Solid wall	5.1	13.1	9.45	42.5	2.3	10.3	13.2	19.5	1.7	9.7	14.2	14.2	3.2	11.2	12	27.0
Window glass	1.9	9.9	61.35	16.2	0.9	8.9	67.7	7.5	0.7	8.7	69.2	5.4	1.2	9.2	56.4	24.0
Total			70.8	21.0			80.9	9.8			83.3	7.1			68.4	23.4

$$\text{Average total heat loss} = \frac{70.8 + 80.9 + 83.3 + 68.4}{4} = 75.85 \text{ W/m}^2$$

$$\text{Average reduction in heat loss} = \frac{89.64 - 75.85}{89.64} = 15.4\%$$

CHAPTER 6

DATA ANALYSIS AND SUMMARY OF THE RESULTS

Energy monitoring is an essential aspect in assessing the efficiency of fuel consumption in any plant or in assessing the success of any conservation programme. The ultimate goal of the monitoring being to improve the control of the overall energy consumption.

At the beginning of this work, the objective was to devise a monitoring system for a Chemical and Process industry i.e. the system incorperating the production rate as well as the space heating component. Thus, the ideal place for such a study was a plant with a multitude of energy consuming unit operations such as distillation, evaporation and drying, in addition to space heating. Several attempts were made to secure such a place but the responce was rather poor. However, a local pharmaceutical Research and Development firm, which uses energy almost entirely for space heating, offered their site for this study. This firm is located in the outskirts of Loughborough. Fig. 4 - 4a and 4 - 4b show the photographs of the site. Thus, the company refer to in this chapter, is this site.

6.1 The Company

This Company, covers an approximate built area of 19045 m^2. It comprises extensive office and laboratory blocks with a considerable number of plant rooms containing heating and ventilation plants and domestic hot water calorifiers.

The large steam raising boiler plant has two modern Ruston boilers, (Capacity- aproximately 0.85 MW each), burning heavy fuel oil, 43,000 sec, and supplies a considerable demand for hot water and heat of air conditioning plant throughout the year. There are no large individual process steam using plants except for a pilot plant in the Pharmaceutical Research and Development Unit (P R D). The schematic steam distribution for the entire site is shown in Fig. 6 - 1.

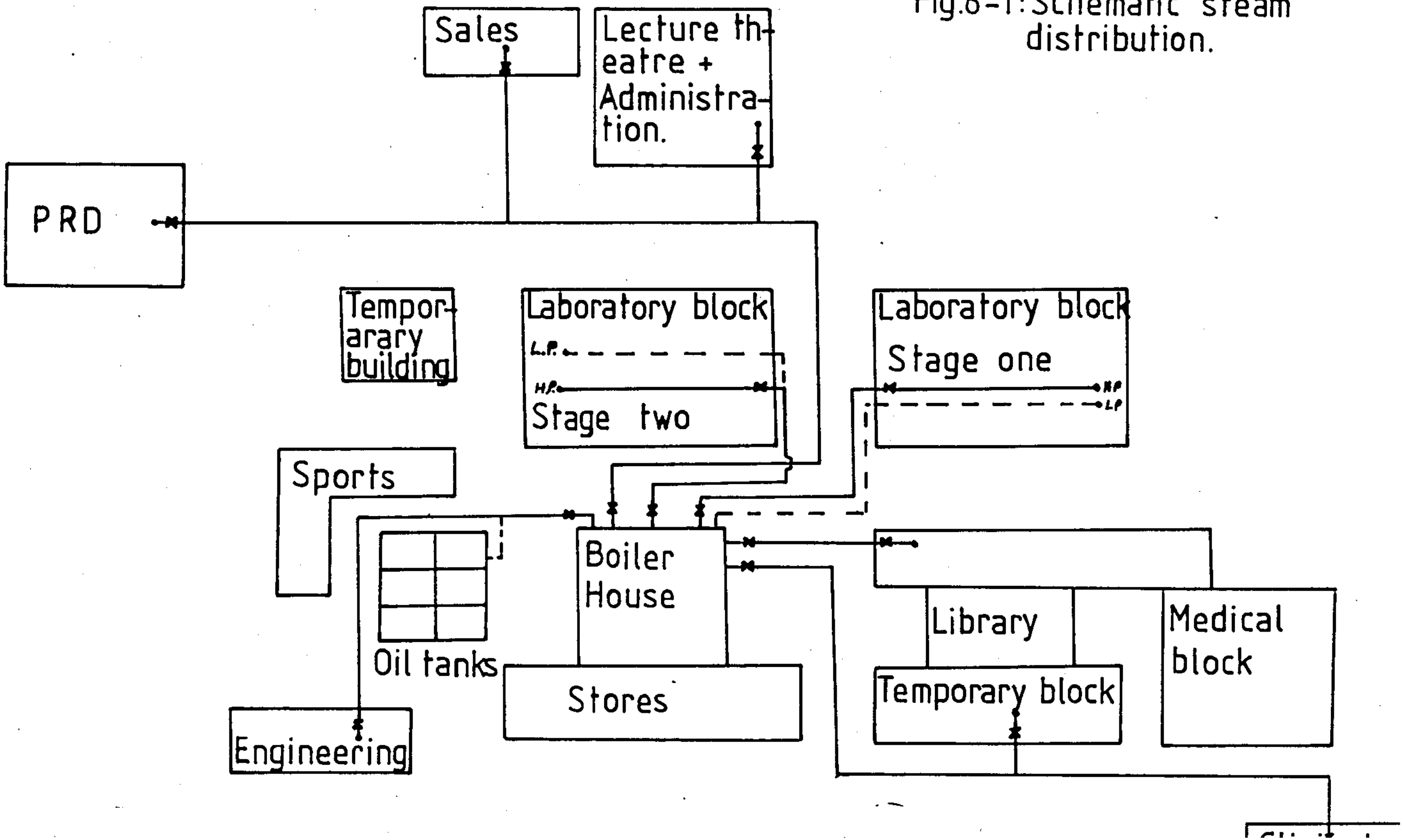


Fig.6-1: Schematic steam
distribution.

Electricity is mainly used for air conditioning plant, fans and refrigeration compressors, air compressors, some extraction systems, lighting and a limited application in space heating for the temporary buildings.

Gas is mainly used for cooking in canteens and for Bench laboratory work. The historical data for the fuel oil consumption, electricity and gas for the period starting from 1975 up to 1983 was available. However, the period starting 1975 to April 1979 was dropped, since the weather data obtained, starts from May 1979 (see section 6.1.1).

The project was confined to the assessment of the effects exerted by the weather parameters such as wind speed, and solar radiation on heavy fuel oil consumption, since a large proportion of electricity and gas used does not depend on these parameters.

6.1.1 Weather data

Historical weather data was supplied by Sutton Bonington Meteorological Office, located less than fifteen miles from Loughborough. It is assumed, therefore, that the measured weather parameters at the Meteorological office, can be taken as being similar to those prevalent at the Company site. The weather data of the period from May 1979 to December 1983 on the following, was obtained:

(a) Daily mean run of wind, km/day
(b) Daily maximum and minimum outdoor temperatures, ^{o}C.
(c) Daily mean Global radiation falling on a horizontal surface, mWh/cm^2.

Although it was mentioned earlier that, rainfall may have effect on energy consumption, it was found by different research workers for example, 4, 58 and 96, that the effect of rainfall is very small and can be neglected without causing significant changes in energy consumption calculations. Thus, in this project we accepted their approach and we neglected this effect.

6.2 Development of a Modified Energy Predicting Method

Before developing a new method, it was logical to investigate the accuracy of the method for predicting energy consumption from the degree day data, so that the accuracy of the new method could be compared with it.

6.2.1 Predictive Accuracy of the Degree day Method

The degree day method assumes, that the energy consumption for a space heating plant is directly proportional to the degree days. Thus, it does not take into account the effects exerted by other weather parameters. In order to assess the accuracy of this method the weekly degree day totals will be obtained by processing the daily weather data in a computer programme, which is discussed under section 7.3. The degree days will be plotted versus the weekly fuel consumption on a linear graph. The regression curve will be fitted, to the points plotted, by using the "least square" method outlined in section 3.2. It is anticipated that most of the points will lie above or below the curve i.e. some weeks will have over estimated fuel consumption, while others will have under-estimated fuel consumption. Thus, the percentage errors will be calculated for each week. These errors should however, be normally distributed with a mean of zero.

The standard error of estimates, herein referred to as "standard errors" will effectively be regarded as a reasonable estimate of the accuracy of the degree day method to predict the fuel consumption.

Finally the regression equation will be used to estimate the fuel consumption based on degree day totals. Thence, the CUSUM technique outlined in section 3.3.3, will be used to monitor the fuel consumption.

6.2.2. Base Temperature

Because of the nature of the work carried out by the company which will not be discussed in detail here, the indoor temperatures in some of the rooms are kept above the value of 18.3 °C, recommended for normal office occupation. Table 6 - 1 shows a crossection of the indoor temperatures in offices and laboratories, measured on 23rd May 1984. Unfortunately, these temperatures cannot be regarded as a representative of winter temperatures, because, the outside temperature on that particular day was rather high i.e. 17.8 °C, producing higher room temperatures than normal. Nevertheless, the verbal confirmation from technicians, operators and people in the offices, indicated that the indoor temperatures are always kept between 19 and 21 °C, and this was enough to facilitate the move for assessing the effect of changing the base temperature on the energy equation. Thus, several base temperatures will be assumed and their contribution to the accuracy of energy estimation equation will be assessed.

6.2.3 New Energy Prediction Method as a Function of Wind and Solar Radiation

Effects of wind speed and solar radiation will be considered individually first and their combined effect will be assessed.

As it was stated in section 4.1.1 and 5.1.6 the Meteorological wind speed and Global radiation will be used throughout the calculations. From chapter four and five following mathematical models will be considered:-

For wind speed:

$$Q = C_1 \Delta T + C_2 v^n$$

$$Q = C_1 \Delta T (1 + C_2 v^n)$$

For solar radiation:

$$Q = C_1 \Delta T - C_2 I_G^m$$

Combined effect:

$$Q = C_1 \Delta T + C_2 v^n - C_3 I_G^m$$

$$Q = C_1 \Delta T(1 + C_2 v^n) - C_3 I_G^m$$

Where; Q = heat loss from the building, W

ΔT = degree days, °C

v = wind speed, m/s

Table 6 - 1 Company indoor temperatures

Two thermometers (electronic and mercury thermometer) were used for measurements. Only the average readings appear in this table.

Building	Temperature readings in °C				
		1	2	3	AVERAGE
Stage 1	ground floor	21	21	21	21
	1st floor	23	23	22	22.7
	2nd floor	23	22	23	22.7
	3rd floor	19	22	23	21.3
Stage 2	G. floor	20	20	21	20.3
	1st floor	22	22	23	22.3
	2nd floor	21	23	21	21.6
	3rd floor	20	21	21	20.7
Library		19	20	21	20.0
Medical Service		19	20	20	19.7
Admin.		19	18	20	19
C S U		25	24	24	24.3
P R D		21	23	24	22.7
Engineering		25	23	23	23.6

Average 21.8 °C

I_G = Global solar radiation, mWh/cm^2

The mathematical models will then be used to estimate the energy consumption for space heating, under stated degree days, wind speed and solar radiation. The regression equation for each model will be determined by the least square method.

The standard deviation of the errors will be calculated and compared with those obtained by using the degree day method. For an improved energy predicting equation the standard errors should be lower than those obtained by using the degree day method.

It will be assumed that the improvement in the energy predicting method reflects the effect of the parameter(s) in question on the energy consumption, i.e. if the new method, including the wind speed say, is 10% more accurate than the degree day method, we will assume that 10% of energy consumption is attributed by the wind speed.

Finally the predicted energy consumption will be subtracted from the actual energy consumption in the same way as discussed in section 3.3.3 and 6.2.1 to build up a cumulative sum for energy monitoring purposes.

6.3. Computer Programmes

Two computer programmes were written and the listings of each,are presented below.

Program 1

The program reads the daily maximum and minimum outside air temperature and calculates the weekly degree day totals by using the procedure outlined in section 3.3.2. Before executing the program the user must type in the year, month, date and the day(when the first data was collected) in sequential order in the program, and then the base temperature.

PROGRAM 1

```
5 REM ** PROGRAM FOR CALCULATING THE DEGREE DAYS
10 REM ** WRITTEN BY KATIMA
15 DIM A$(35),A(35),C(35),D(35),W(36),R(35)
20 FOR I=1 TO 1000:NEXT
30 INPUT"YEAR";YY
40 INPUT"MONTH";MM
50 INPUT"DAY";D
60 INPUT"DAY OF WEEK";D$
70 INPUT "BASE TEMP";T1
80 REM**W=RUN OF WIND,B=MAX TEMP,C=AV. TEMP,D=AV.TEMP DIFF
85 REM**A=MIN TEMP,R=SOLAR RADIATION
90 REM**N=NO. OF DAYS IN THE MONTH
100 GOSUB 470
105 PRINT
110 READ N
115 IF N=99 THEN PRINT"END OF DATA":STOP
120 FOR J=1 TO N
130 READ W(J),A(J),B(J),R(J)
140 C(J)=(A(J)+B(J))/2
145 IF B(J)>T1 THEN D(J)=0:GOTO 225
150 IF A(J)>T1 GOTO 180
160 D(J)=T1-C(J)
170 GOTO 225
180 IF C(J)<T1 GOTO 210
190 D(J)=0.25*(T1-B(J))
200 GOTO 225
210 D(J)=0.5*(T1-B(J))-0.25*(A(J)-T1)
225 IF WD=1 THEN S=0:S1=0
230 S=S+C(J)
240 S1=S1+D(J)
250 IF WD=7 GOTO 280
260 GOTO 300
280 PRINT"WEEK DEG DAY";S1
290 PRINT
300 F=29
310 IF YY/4=INT(YY/4) THEN F=30
315 D=D+1
320 IF D<F GOTO 450
330 IF MM=2 GOTO 380
340 IF D<31 GOTO 450
350 IF MM=4 OR MM=6 OR MM=9 OR MM=11 GOTO 380
360 IF D=32 GOTO 380
370 GOTO 450
380 MM=MM+1
390 D=1
400 IF MM>12 GOTO 420
410 GOTO 450
420 MM=1
430 YY=YY+1
450 WD=WD+1
455 IF WD=8 THEN WD=1
460 NEXT J
465 GOTO 110
```

```
470 FOR I=1 TO 7
480 READ A$(I)
490 NEXT I
500 FOR I=1 TO 7
510 IF A$(I)=D$ THEN WD = I:GOTO 540
520 NEXT I
530 GOTO 60
540 RETURN
550 DATA"MON","TUE","WED","THU","FRI","SAT","SUN"
```

Program 2

The program reads the weekly actual fuel consumption, weekly degree days, weekly average wind velocity and weekly average solar radiation. Then it calculates the mean values of the fuel consumption, degree days, wind velocity and solar radiation, the regression coefficients and prints out the actual fuel consumption, estimated fuel consumption, their differences and the cumulative sum of these differences. Also it computes the correlation coefficient and standard error as percentage of the mean fuel consumption. The number of independent variables have to be typed in before excuting the program.

6.4 Observations

6.4.1. Correlation of Degree Days With Fuel Consumption

The graph of weekly actual fuel consumption versus weekly degree days for the period of May 1979 to December 1983 is shown in Fig.6-2. The degree days were calculated by using a 15.5 ^{o}C base temperature. As expected the straight line fitted by the "Least square" method show a reasonably good correlation of 0.9175, between the fuel consumption and degree days.

The line with the slope of about 237.5 litres per week per degree day, cuts the vertical line at the value of 13000 litres per week. This, was defined as the base load, see section 3.3.3, and it constitutes the energy used to supply hot water and energy used to overcome losses from the steam system as discussed previously.

From appendix A.1, it can be assumed that the distribution loss due to convection and radiation from steam system, leaks and faulty steam traps, for the entire plant is uniform, and stands at about 25% of the heat supplied.

Therefore, energy loss $\hat{=}$ 0.25 x 23332 $\hat{=}$ 5833 litres/week
Boiler losses contributes to this amount of base load, however, it is difficult to evaluate the boiler inefficiency as it depends upon the boiler load (which is not constant), with maximum efficiency obtainable

```
PROGRAM2
5 REM**PROGRAM WRITTEN BY KATIMA
10 INPPUT"NUMBER OF INDEPENDENT VARIABLES";N
15 REM**D=NUMBER OF LINES OF DATA
20 READ D
25 IF D<N GOTO 10
30 DIM X(N,D),Y(D),S(N),B(N),P(N),A(N,N)
35 PRINT:PRINT
40 FOR K=1 TO D
45 READ Y(K)
50 FOR J=1 TO N
55 READ X(J,K)
60 S(J)=S(J)+X(J,K)/D
65 NEXT J
70 SO=SO+Y(K)/D
75 NEXT K
80 PRINT"MEAN VALUES ARE"
85 PRINT SO
90 FOR J=1 TO N
95 PRINT S(J)
100 NEXT J
110 FOR K=1 TO D
111 Y(K)=Y(K)-SO
114 FOR J=1 TO N
115 X(J,K)=X(J,K)-S(J)
120 NEXT J
125 NEXT K
130 FOR J=1 TO N
135 FOR K=1 TO D
140 B(J)=B(J)+Y(K)*X(J,K)
145 NEXT K
150 FOR I=1 TO N
155 FOR K=1 TO D
160 A(J,K)=A(J,I)+X(I,K)*X(J,K)
165 NEXT K
170 NEXT I
175 NEXT J
180 FOR I=1 TO (N-1)
185 Z=ABS(A(I,I))
190 L=I
195 IF I=N GOTO 230
200 FOR M=(I+1) TO N
205 IF ABS(A(M,I))>Z GOTO 215
210 GOTO 225
215 Z=ABS(A(M,I))
220 L=M
225 NEXT M
230 IF Z=0 GOTO 560
235 IF I=L GOTO 280
240 FOR M=I TO N
245 Q= A(I,M)
250 A(I,M)=A(L,M)
255 A(L,M)=Q
260 NEXT M
```

```
265 Q=B(I)
270 B(I)=B(L)
275 B(L)=Q
280 IF I=N GOTO 325
285 FOR K=(I+1) TO N
290 M=-A(K,I)/A(I,I)
295 FOR J=(I+1) TO N
300 A(K,J)=A(K,J)+M*A(I,J)
305 NEXT J
310 B(K)=B(K)+M*B(I)
315 NEXT K
320 NEXT I
325 FOR I=N TO I STEP -1
330 S=B(I)
335 IF I=N GOTO 355
340 FOR K=(I+1) TO N
345 S=S-A(I,K)*P(K)
350 NEXT K
355 P(I)=S/A(I,I)
360 NEXT I
365 S1=S0
370 FOR I=1 TO N
375 S1=S1-P(I)*S(I)
380 NEXT I
385 PRINT:PRINT
390 PRINT"REGRESSION EQUATION IS"
395 PRINT
400 PRINT" Y = ";S1
405 FOR I=1 TO N
410 PRINT"    +(";P(I);")*X(";I;")"
415 NEXT I
420 PRINT
425 PRINT"TYPE CONT FOR CUMULATIVE SUM"
430 STOP
435 W=0:Q=0
440 PRINT
445 Z=0
450 F=0
455 PRINT"ACTUAL     PREDICTED";
460 PRINT"DIFFERENCE     CUSUM"
465 FOR K=1 TO D
470 W=W+Y(K)*Y(K)
475 C=0
480 FOR J=1 TO N
485 C=C+P(J)*X(J,K)
490 NEXT J
495 Q=Q+C*C
500 F=F+Y(K)-C
510 PRINT Y(K)+S0,INT(C+S0),
514 PRINT INT(Y(K)-C),INT(F)
515 Z=Z+(Y(K)-C)^2
520 NEXT K
525 R=SQR(Q/W)
530 PRINT
```

```
535 PRINT"REGRESSION COEFFICIENT=";R
540 Z=SQR(Z/(D-N-1))/SO
541 PRINT"STANDARD ERROR=";
545 PRINT INT(1000*Z)/100;"% OF MEAN"
550 GOTO 565
560 PREINT "CAN'T BE DONE"
570 END
```

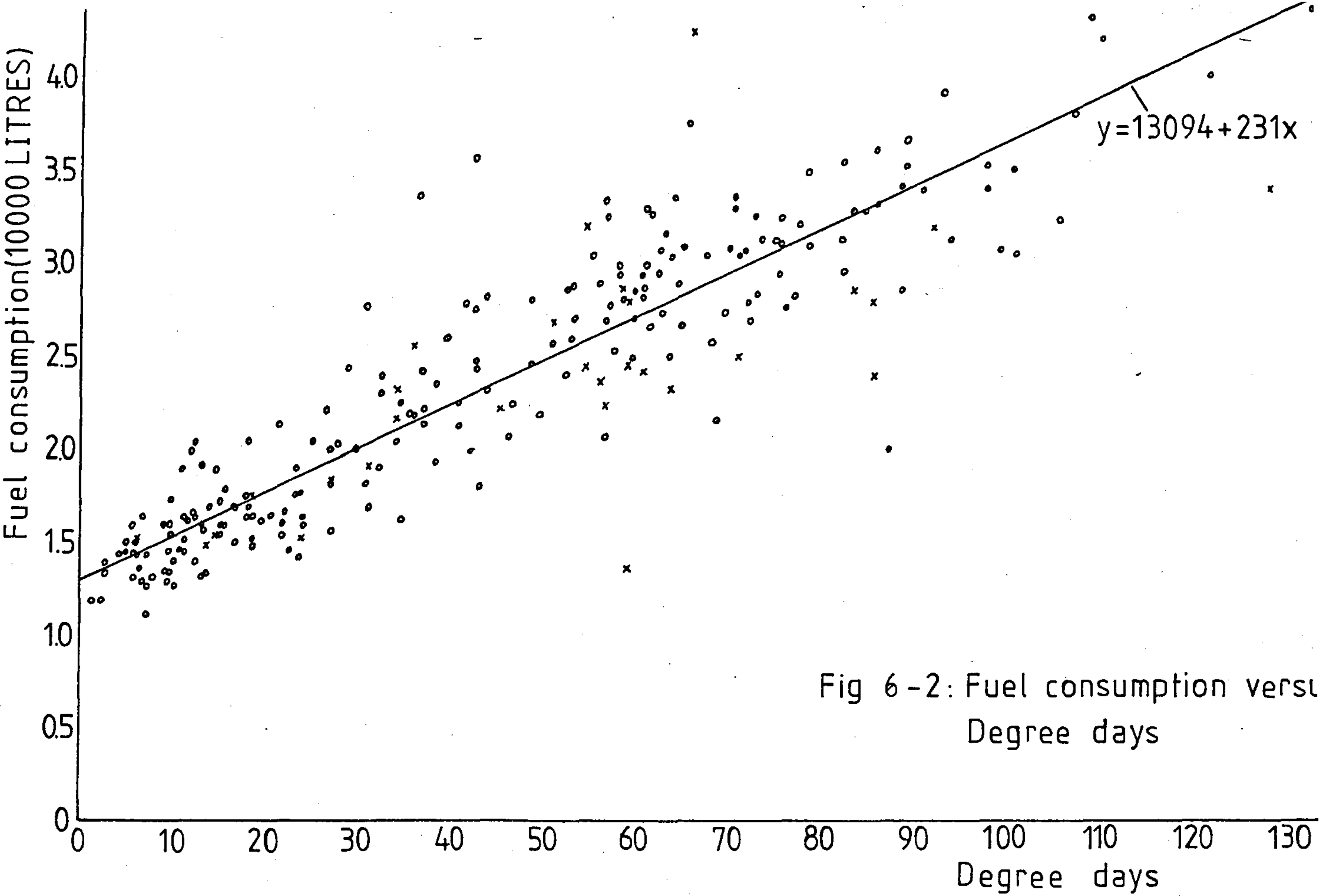


Fig 6-2: Fuel consumption versu
Degree days

at full load, see section 2.1.1.2. Thus, for simplicity we will assume that the boiler was operating at full load throughout and hence maximum efficiency, of around 80% can be used.

Therefore, the average energy loss due to boiler inefficiency (based on average weekly fuel consumption of 23332 litres)

$$\hat{=} \ 0.2 \times 23332 = 4666 \text{ litres/week}$$

There was no available data for hot water consumption and because the company uses hot water calorifiers (which complicates the quantification of energy used to supply hot water), we had to approximate the amount of hot water as follows:

- The dishwater in the canteen(serving approximately 600 staff), operating for around 2500 hours per year is consuming approximately 1.4 m^3/h of hot water at arround 60 ^oC.

- Hot water for other uses (e.g. for hand washing in Laboratories, canteen and toilets) is approximately 10 litres per person per day.

Thus total volume = $1.4 \times 2500/52 + 600 \times 10 \times 10^{-3} \times 7 = 109.3 m^3$/week

Assuming an average winter temperature of about 8 ^oC and an error of 50% in estimating the hot water consumption, the energy used to raise the water to around 60 ^oC is given by:

$$Q = V \rho C_p \Delta T = (109.3 \times 1.5)\, 1000 \times 4.2\, (60-8)$$

$$= 35.8 \times 10^3 \text{ MJ/week (871 l/week)}$$

Boiler blowdown losses can be substantial, because the blowdown is carried out manually for 10 seconds (which is subjective), three times a day. Therefore, a 5% blowdown loss will be assumed (see section 2.1.1.2).

Thus, blowdown loss $\hat{=}$ 23332 x 0.05 = 1167 l/week

Heat used to maintain the oil storage tanks at 40-50 ^oC $\hat{=}$ 96 l/week, see appendix A.2, however, this value might be a bit higher during winter periods as the heat loss from the tank depends solely on the outside temperature (the outside temperature was high when the measurements were made) and the wind speed. From the rough calculations above, the base load of about 12633 litres/week is anticipated. This value, within practical limits, is in agreement with the 13,000 litres/week obtained from Fig.6 - 2.

OUTLIERS

It can be seen from Fig. 6 - 2, some of the points lie either very far above or very far below the regression line. Suprisingly enough some of the points lying above the line correspond with weeks containing Bank holidays, Christmas and Easter holidays, hereafter referred to as "holidays", (marked by "X" on Fig.6 - 2). Because, energy, is only used for air conditioned rooms, it is expected that less energy consumption should have been recorded, hence the points with holidays should appear below the regression line. This suggests that all the weeks with holidays in them should be treated with some doubt. Thus it was decided that these weeks should be discarded.

Other points which lie very far above and below the regression line, may be due to errors in meter reading. It was gathered that meter reading is carried out by a single person (not particularly the same), at 12 pm of each Sunday, and there is no mechanism of counterchecking the readings. It has happened sometime that the man responsible can unintentionally fall asleep and take the readings after 12 pm or can take the readings before 12 pm or can read the meter incorrectly. However, there is no evidence to support these verbal arguments, therefore these observations will be left in the data as they appear.

Following, are the regression equations, Correlation coefficients and standard errors for data points with and without holidays.

Table 6.2 : Predictive accuracy of the degree day method

Situation	Regression equation	Correlation Coefficient	Standard error (% of mean)
With holidays	y = 13094 + 231 DD	0.9175	13.15
Without holidays	y = 12933 + 239 DD	0.9392	11.51

Table 6.16 : Standard errors for the Ratcliffes' degree day method

Situation	Independent variables	Standard errors (% of mean)		Ratcliffes' Modified degree day method			
				Standard errors (% of mean)		Improvement over linear relationships (%)	
		15.5°C	16.5°C	15.5°C	16.5°C	15.5°C	16.5°C
May 1979 - Dec.1983 with holidays	Degree days	13.5	12.34	13.5	12.34	0	0
May 1979 - Dec.1983 without holidays	Degree days	11.51	11.34	11.51	11.34	0	0
May 1979 - Dec.1983	Degree days wind velocity Global radiation	10.65	10.63	10.77	10.55	-1.12	0.75
28th week, 1980 - Oct.1983 without holiday	Degree days	9.98	9.77	9.98	9.77	0	0
28th week 1980 - Dec.1983 without holidays	Degree days wind velocity Global radiation	9.10	9.10	9.3	9.10	-2.20	-0.11

Note: a negative sign means "Less accurate"

Where; y = fuel consumption, litres per week

DD = degree days

Billington 1966(12) reported that errors in estimating fuel consumption based on monthly degree day totals, could be as much as ± 25%. Ratcliffe, 1981(96) reported the values of ± 28% and 48% for Hawker Siddley and Princess buildings respectively whilst the IHVE Guide estimates the errors to lie between ± 15% and ± 25%. However, the Department of Energy (27) gives the values of about ± 5% to ± 10%.

Therefore, it is apparent that errors in predicting energy consumption by using the Degree day method depends upon the situation it is applied, the period to be assessed and the accuracy of the energy data available, and thus, it can be as low as ± 5% and as high as the errors in data collection and inefficiency in energy consumption can dictate. The value of 13.5% in table 6 - 2 is within a reasonable range (for the degree day method), however, probably it can be much better, if a longer period of one month, for example, is used (27).

The standard error of 11.51% obtained after discarding the weeks with holidays represent a difference of about 12.5% in the accuracy of the same method. This is quite substantial, thus, unless specified, the weeks containing holidays are not included in the data used in the discussions hereafter.

6.4.2 Effect of Weather Parameters on Fuel Consumption

Program 2, presented in section 6.3, was used to calculate the effect exerted by the wind speed and solar radiation on fuel consumption. A summary of the results is given in table 6.3.

Table 6.3 : Effect of weather parameters on energy equation (Base temperature 15.5 °C)

Independent variables	Regression equation	Correlation coefficient	Standard error,% of mean	Improvement (%)
Degree days	$y = 12933 + 239\ DD$	0.9392	11.51	Basis
Degree days and Wind velocity	$y = 10338 + 235\ DD + 12\ RW$	0.9461	10.88	5.5
Degree days and Global radiation	$y = 15909 + 212\ DD - 7\ GR$	0.9445	11.03	4.2
Degree days, wind velocity and global radiation	$y = 1294 + 251DD + 10RW - 5GR$	0.9486	10.65	7.5

The equations appearing in table 6.3 indicate that the heat loss, hence fuel consumption, increases with degree days and wind speed and decreases with increasing solar radiation.

Based on the assumptions given in section 6.2.1, that the improvement in energy equation's accuracy (i.e. the percentage decrease in standard error of estimates), reflects the effect of that particular parameter on energy consumption, then ,

The effect of wind speed on energy consumption is about 5.5% and that of solar radiation is about 4.2%. It is worth noting that the overall effect i.e. 7.5% is not the sum of the effects exerted by the individual parameter. This could mean that these parameters do not act independently. There is a possibility that one effect may influence another.

For example it was shown in section 5.1.4, that the absorptivity of solar energy on the outer wall of a building increases the average surface temperature of the wall by

$$\Delta T = \frac{a\ I_G}{h_{oo}+U}$$

Where a = absorption coefficient

I_G = Insolation W/m^2

Assuming that h_{oc} remains constant, the increase in surface temperature reduces the fabric heat loss (see section 5.1.6). But h_{oc} increases with wind speed (see section 4.2), thereby decreasing ΔT for a given insolation. Thus the wind speed reduces the effect of solar radiation to some extent.

6.4.3 Effect of Changing the Base Temperature on Energy Equation

Base period

Around the 28th week of 1980 there was a change in the trend of energy consumption i.e. there was a negative trend up to the 27th week of 1980 and a positive trend thereafter, see Fig B - 1. This will be discussed in details later in section 6.5. The positive trend was almost maintained until the end of 1983. Thus this period will be regarded as the representative of the present, actual fuel consumption. Therefore, the results given in table 6.4 are based on this period.

Table 6.4: Effect of changing the base temperature

Base Temperature °C	Regression equation	Correlation Coefficient	Standard error (% of mean)
15.5	y = 13760 + 222DD + 8RW - 6GR	0.9636	9.10
16.5	y = 12803 + 215DD + 8RW - 5GR	0.9637	9.09
17.0	y = 12332 + 213DD + 7RW - 5GR	0.9634	9.12
17.5	y = 11830 + 211DD + 7RW - 5GR	0.9629	9.19
18.5	y = 11044 + 206DD + 6RW - 5GR	0.9614	9.37

The 16.5 °C base temperature gives a regression equation with a slightly better correlation coefficient and a negligibly improved accuracy of around 0.11% over that based on the 15.5 °C base temperature. However, from the definition of the degree day, it is assumed that the difference between the selected base temperature, is compensated by the incidental heat gains from occupants, lighting and electrical appliances, and it was mentioned in section 6.2.2 that the company indoor temperatures are kept between 19°C and 21°C. If an average indoor temperature of 19.5°C is assumed, for example, the difference between this temperature and the 15.5°C base, may be too much to be offset by the incidental heat gains. Thus from a practical point of view the 16.5°C base temperature was considered to be suitable for this particular situation. Therefore the discussions based on the 15.5°C base temperature are given for comparison purposes.

A summary of the effects exerted by weather parameters based on the 16.5°C base temperature for the whole period, i.e. May 1979 to Dec.1983, is shown in table 6.5. While tables 6.6 and 6.7 show the same effects for the period starting on the 28th week of 1980, for the base temperature of 15.5°C and 16.5°C respectively.

Table 6.5 : Effect of weather parameters on energy equation (base temperature : 16.5°C)

Independent variables	Regression equation	Correlation Coefficient	Standard error(% of mean)	improvement %
Degree days	y = 11968 + 236 DD	0.9410	11.34	Basis
Degree days and Wind velocity	y = 9564 + 226 DD + 11 RW	0.9467	10.82	4.6
Degree days and Global radiation	y = 14702 + 207 DD - 6 GR	0.9452	10.96	3.4
Degree days wind velocity & Global radiation	y = 12009 + 209 DD + 9 RW - 5 GR	0.9488	10.63	6.3

Comparing table 6.3 and 6.5 it can be seen that there is a very slight improvement in the correlation coefficient and a very slight decrease in the standard error, although this improvement is too small to be of more than academic interest. However, the decrease in the effects exerted by the weather parameters, suggests that there is a danger of exaggerating the effect exerted by weather parameters, by selecting a wrong base temperature. That means the error in base temperature selection will feature in the error of estimates.

Table 6.6 Effect of weather parameters on energy equation (base temperature 15.5°C starting: 28th week 1980)

Independent Variable	Regression equation	Correlation Coefficient	Standard error (% of mean)	Improvement (%)
Degree days	y = 13273 + 247 DD	0.9554	9.98	Basis
Degree days and Wind speed	y = 10842 + 242 DD + 11RW	0.9608	9.41	5.7
Degree days and Global radiation	y = 16327 + 219 DD - 7 GR	0.9608	9.40	5.8
Degree days Wind speed & Global radiation	y = 13760 + 222DD + 8RW - 7GR	0.9636	9.10	8.8

Table 6.7 : Effect of weather parameters on energy equation (base temperature 16.5°C, starting: 28th week 1980)

Independent variables	Regression equation	Correlation coefficient	Standard error (% of mean)	Improvement(%)
Degree days	y = 12304 + 237 DD	0.9573	9.77	Basis
Degree days and Wind speed	y = 10174 + 233DD + 10RW	0.9615	9.32	4.6
Degree days and Global radiation	y = 15111 + 213DD - 7GR	0.9614	9.33	4.5
Degree days Wind speed & Global radiation	y = 12803 + 215DD + 7RW - 5 GR	0.9637	9.09	7.0

So far we have been looking on a linear heat loss equation. From the assumptions made earlier, that the period starting from the 28th week of 1980 represents the actual trend of the company fuel consumption; that the 16.5°C base temperature is suitable for this situation; that the improvement in the accuracy of predicting equation reflects the effect of the parameter in question, on energy consumption; then the percentage effect of wind speed on heat loss is about 4.6% while that of Global radiation is about 4.5%, and their combined effect is about 7.0%. That means the regression equation incorperating wind speed and solar radiation is 7% more accurate than the degree day method.

6.4.4 Other Mathematical Models

Tables 6.8 to 6.12 give the summary of the results obtained by using different mathematical models i.e. non-linear equations. The base temperature of 16.5 °C and the period starting from the 28th week of 1980 is used throughout the calculations.

Wind speed:

1st mathematical model:-

$$Y = C + C_1 DD + C_2(RW)^n \quad \ldots\ldots\ldots\ldots\ldots \quad 6.2$$

Table 6.8 : A summary of results for the 1st mathematical model

n	C	C_1	C_2	Correlation Coefficient	Standard error (% of mean)
0.5	8277	233	274	0.9608	9.41
0.8	9688	233	35	0.9612	9.36
1.0	10174	233	10	0.9615	9.32
1.5	10853	232	0.4	0.9622	9.24
2.0	11221	232	0.02	0.9628	9.17
2.5	11457	232	1×10^{-3}	0.9633	9.11
3.0	11622	232	5×10^{-5}	0.9637	9.06

Wind Speed :

2nd mathematical model :-

$$y = C + C_1DD\ (1 + C_2(RW)^n) \quad \ldots\ldots\ldots\ldots\ldots \quad 6.3$$

Table 6.9 : A summary of results for the 2nd mathematical model

n	C	C_1	C_2	Correlation Coefficient	Standard error (% of mean)
0.5	12253	186	3.4	0.9594	9.57
0.8	12252	201	0.44	0.9598	9.52
1.0	12252	207	0.13	0.9601	9.49
1.5	12251	214	5.8×10^{-3}	0.9607	9.41
2.0	12251	218	2.8×10^{-4}	0.9613	9.35
2.5	12250	221	1.4×10^{-5}	0.9618	9.29
3.0	12250	224	7.0×10^{-7}	0.9621	9.25

Comparing the respective n - values in table 6.8 and table 6.9, reveals that the variation in the accuracy of the equations is ranging from 1.5% to 2.0% in favour of the regression equations in table 6.8. Also table 6.8 show slightly better correlation coefficients. Therefore, basing our conclusion merely on standard errors and correlation coefficient, the multiple linear equations given in table 6.8, feature favourably than the multiple non - linear equations given in table 6.9.

However, although the values of n (in table 6 - 8) ranges from 0.5 to 3 (i.e. a 500% difference) they show a very slight difference in the standard errors and correlation coefficients of about 3.7% and 0.3% respectively. Besides, the values of $n > 1.5$ lacks the theoretical background, see section 4.1.3.1. That means, for this particular site, the exponent of the wind speed is of no significant importance in the energy equation. Therefore the heat loss is directly proportional to the wind speed, and the energy equation is of a linear form:

$$y = C + C_1DD + C_2\ RW$$

From table 6.7 the range of the improvements in energy equation's accuracy, suggest that the close infiltration rate equation (suitable for this site) is:

$$I = 4.92 + 0.43v \text{ (see section 4.1.3.1 et sec.)}$$

and that of h_{oc} is :

$$h_{oc} = 6.2 + 4.2\,v \quad \text{for} \quad v \leq 5 \text{ m/s}$$
$$h_{oc} = 7.5\,v^{0.8} \quad \text{for} \quad v > 5 \text{ m/s}$$

Table 6 - 10 show the total heat loss due to infiltration and fabric heat loss as influenced by the wind speed (see tables 4 - 1 and 4 - 2).

Table 6 - 10 : Total heat loss due to wind speed

Wind Speed m/s	H E A T		L O S S	
	Due to Infil. kW	Fabric kW	Total kW	Increase %
2	0.253	6.791	7.044	Basis
3	0.272	6.909	7.179	1.9
6	0.329	7.515	7.844	11.4
9	0.386	7.865	8.251	17.1

The percentage effect of 4.6% obtained in table 6 - 7 is more or less in a reasonable range of the percentage increase portrayed in table 6 - 10 for lower wind speeds.

The 4.6% improvement may sound negligible but, it is significant in real terms. For example, the company is currently using about £230000 per year to purchase heavy fuel oil. If the annual budget of the fuel consumption is based upon the degree day method (which is 4.6% less accurate than the method including the wind speed) the predicted fuel consumption will be 4.6% less than the actual fuel consumption. Thus, in monetary terms, the budget will suffer a deficit of about £10580, which is quite substantial.

Solar radiation;

3rd mathematical model :-

$$y = C + C_1\,DD - C_2(GR)^n \quad \ldots\ldots\ldots\ldots \quad 6.4$$

Table 6.11 : A summary of results for 3rd mathematical model

n	C	C_1	C_2	Correlation Coefficient	Standard error, % of mean
0.2	21392	210	− 2682	0.9622	9.24
0.5	16885	210	− 219	0.9621	9.25
0.8	15615	212	− 26	0.9618	9.29
1.0	15111	213	− 7	0.9615	9.33
1.2	14723	215	− 2	0.9611	9.38
1.5	14266	218	− 0.22	0.9605	9.44

Although the value of n = 0.2 gives a slightly better correlation and a slightly improved accuracy, it lacks theoretical background. Also the negligible difference between the correlation coefficients obtained by using equations of different n-values suggests that the exponent of the solar radiation intensity, is of no significant importance to heat gain due to solar radiation. Thus we will stick to the theories outlined in chapter 5, that the heat loss decreases with increasing solar radiation intensity (see section 5.1.6). The energy equation therefore, is of a linear form:

$$y = C_2 + C_2DD - C_3GR$$

The solar radiation effect of 4.5% (see table 6 − 7) is, somehow, realistic than the 15.4% obtained in table 5 − 1. The surface temperature increase ΔT (see section 5.1.6), was calculated by assuming an average insolation, but the insolation is never constant in nature. Also it was assumed that the wind speed does not affect the surface heat transfer coefficient, h_{oc}, which is not true, see section 4.2. Therefore, the reduction in heat loss potrayed in table 5 − 1 is exaggerated.

Wind speed and solar radiation

4th mathematical model

$$y = C_1 + C_2DD + C_3(RW)^m - C_2(GR)^n \quad \ldots\ldots\ldots\ldots \quad 6.5$$

Table 6.12 : A summary of results for the 4th mathematical model

n	m	C	C_1	C_2	C_3	Correlation Coefficient	Standard error (% of mean)
0.5	1.0	11573	215	202	- 5.2	0.9631	9.16
0.8	1.0	12498	215	27	- 5.0	0.9634	9.12
1.0	1.0	12803	215	7	- 5.0	0.9637	9.09
1.5	1.0	13213	216	0.3	- 4.8	0.9642	9.03
2.0	1.0	13428	216	0.02	- 4.6	0.9647	8.97
2.0	0.2	18278	213	0.02	- 2014	0.9653	8.89
2.0	0.5	14853	213	0.02	- 163	0.9652	8.4
2.0	0.8	13846	215	0.02	- 18	0.9649	8.94
3.0	1.0	13679	216	4.2×10^{-5}	- 4.5	0.9655	8.87

Wind velocity and solar radiation:

5th mathematical model;

$$y = C + C_1 DD\,(1 + C_2(RW)^n) + C_3(GR)^m \quad \ldots\ldots\ldots\ldots \quad 6.6$$

Table 6.13 : A summary of results for the 5th mathematical model

n	m	C	C_1	C_2	C_3	Correlation Coefficient	Standard error (% of mean)
0.5	1.0	14724	181	2.3	- 5.7	0.9623	9.25
0.8	1.0	14670	191	0.32	- 5.6	0.9626	9.22
1.0	1.0	14638	194	0.09	- 5.5	0.9628	9.19
1.5	1.0	14570	200	4.5×10^{-3}	- 5.3	0.9633	9.13
2.0	1.0	14521	203	2.3×10^{-4}	- 5.2	0.9638	9.08
2.0	0.2	19716	201	2.2×10^{-4}	- 2199	0.9644	9.01
2.0	0.8	14950	202	2.2×10^{-4}	- 21	0.9640	9.05
3.0	1.0	14472	207	5.8×10^{-7}	- 5.1	0.9645	8.98

Comparing the respective n - values in table 6.12 and 6.13, the equations in table 6.12 are 1.1% to 1.3% more accurate than those in table 6.13. Besides, table 6.12 give very slightly better correlation coefficient. From statistical point of view, the reliability of the regression curve varies, with correlation coefficient and inversely with the standard error of estimate. However, this criterion only, is not enough to conclude that, the value of n and m which gives a lower standard error gives the "real nature" of relationship.

However, it was shown in chapter four that, the relationship between the thermal load imposed by wind takes the form $Q \propto v^n$, where, $1 \leq n \leq 1.43$. Also it was shown in chapter five that, the relationship between heat gain and solar radiation intensity takes the form $Q \propto I_G^m$. In tables 6.8 to 6.13 the n and m took the values from 0.5 to 3.0 and 0.2 to 1.5 only to give trivial results. Therefore, as it has been stated earlier, the exponent values for the wind speed and solar radiation are not particularly important for this site. Thus the best energy equation will probably take the form;

$$Q = C_1 + C_2(DD) + C_3(RW) - C_4(GR) \quad \ldots\ldots\ldots\ldots \quad 6.7$$

From table 6 - 7 the standard error of about 9.09 which represent an improvement of about 7% is realised by incorporating the solar radiation and run of wind in the energy equation. This improvement is significant in real terms. For example if the energy consumption is under estimated by 7%, will mean a deficit of around £16100 per year on the annual budget (assuming the energy budget of about £230000). Whilst over estimation of about 7% can sometimes mean cutting down the same amount from the money available for other, equally important, activities such as Research and Development.

6.4.5 Ratcliffe's Modified Degree Day Method

Ratcliffe, 1981 (96) formulated the following degree day equation for building energy loss estimation.

$$DD^* = \sum \left(dd \left[1 + 0.08726 \left(\frac{RW}{86.4} \right)^{0.564} \right] - \frac{GR}{n} \right) \quad \ldots\ldots \quad 6.8$$

Where

dd = the daily degree day

n = dividing factor dependent upon month of year

The values of n given in his report are reproduced in table 6.14.

Table 6.14 : Values of n

Month	n
January	123
February	216
March	287
April	306
May	316
June	281
July	170
August	82
September	168
October	200
November	176
December	142

He reported an improvement in accuracy, on the old degree day method, of about 32% for Hawker siddley and 44% for Princess House. This modified equation was used with the company energy data to assess its accuracy. The values of n were incorporated in the weather data and the computer program 1 was used to evaluate the new degree day totals, for the base temperatures of 15.5°C and 16.5°C. Some weeks recorded negative degree day totals, and zero degree days was used for these weeks instead. Then the new degree days and the fuel consumption were processed in program 2 to compute the regression coefficients, correlation coefficients and the standard error.

As usual the standard errors were used as a measure of accuracy for the method. Table 6.15 gives a summary of the regression equations obtained.

Table 6.15 : Regression equations based on Ratcliffe's modified degree day Method.

Situation	Regression equations	
	15.5°C base temperature	16.5°C base temperature
May 1979 - Dec.1983 with holidays	$y = 15100 + 188\ DD^*$	$y = 14474 + 178\ DD^*$
May 1979 - Dec.1983 without holidays	$y = 14968 + 195\ DD^*$	$y = 14330 + 184\ DD^*$
28th week 1980 - Dec. 1983 without holidays	$y = 15245 + 202\ DD^*$	$y = 14604 + 191\ DD^*$

Table 6.16 gives a summary of the standard errors for the regression equations given in table 6.15 and their comparison with the respective regression equations obtained previously. From table 6.16 it can be seen that Ratcliffe's degree day method is less accurate, by a minor value of around 0.7%, than the regression equation outlined in section 6.4.3. Thus, it can be concluded that, the Ratcliffe's method can be applied to this data with more or less the same accuracy as the regression equations outlined before. It is worth mentioning that, in deriving this modified degree day, Ratcliffe assumed that the heat loss from a building fabric is proportional to the square of the wind velocity i.e. $Q \propto v^2$ and the standard errors obtained by using his equation are almost the same as those obtained by assuming a linear relationship between heat loss and wind velocity i.e. $Q \propto v$. This supports the argument given in section 6.4.4, that, the exponent values of wind speed and solar radiation are of no importance in this particular situation. The main handicap of the Ratcliffe's modified degree day method lies in the empirical constants. These constants depends upon the leakage characteristic of the specific building, the type of terrain, type of exposure the building is situated in and many similar characteristics, which influence ventilation losses and are particular to each building. The empirical constants in equation 6.8 were determined after the study performed on two buildings only. This method could be improved to give, statistically, more reliable results by making further studies, similar to those done by Ratcliffe, on several buildings to

obtain average values for these empirical constants.

6.5. Construction of a Monitoring System

The cumulative sum technique outlined in section 3.3.3, was used for this purpose. The estimated fuel consumption obtained by using some of the regression equations, discussed in the previous sections, were subtracted from the actual fuel consumption, and the differences were added together week by week. The sum of successive differences were plotted as they appear to form a CUSUM chart. As it was mentioned earlier, if there is no change in the mean consumption from the original data the graph is supposed to alternate fairly randomly about the zero axis or a line parallel to the zero axis.

It can be seen from Fig. B - 1 that there was a negative trend with a slope of about - 10% until around the 28th week of 1980. Then a positive trend started with a slope of about 2.5%. Thus the total change in fuel consumption is about 12.5%. This value is almost in agreement, within practical limitations, with the percentage increase of about 11% calculated from the mean fuel consumption of the two periods i.e. 21685 litres per week for the period ending on the 28th week of 1980 and 23988 litres per week for the remaining period.

Attempts were made to identify the causes of that change as follows;

Source of Information

(a) Boiler house log - sheets
(b) Minutes of the Groups' energy Managers' meetings
(c) Internal memorandum
(d) Verbal information from plant operators and technicians.

Several possible causes were revealed, viz;

(i) Starting April 1979 there was a world wide fuel deficit on the market because of the Iran uprising. Hence, the fuel allocation for the company, from their suppliers, was cut by 30%. Although this cut was not transmitted directly to the plant's fuel consumption, because of the substantial fuel stocks the company had at that particular time, as a precaution a tight control on energy consumption was exercised. For example, some services, such as comfort heating, background heating, steam to clinical supplies Unit (which was supposed to come on line almost at the same time) were switched off and heating system for the oil storage tanks was maintained on two tanks only.

(ii) Around September in the same year there was a strong energy conservation campaign. In addition to normal enforcing of good house_keeping practices, outlined in chapter 2, they added insulation to all pipe work carrying steam and condensate. This move could have resulted from the first cause.

(iii) Around the 30th week in 1980 the fuel situation was back to normal. Most probably the steam line to clinical supplies unit, which consumes around 150 lb/hr of steam, was in full operation. This amount resulted into approximately 4% change in energy consumption. Also other services such as back ground heating, were operating normally.

(iv) During the unspecified period of between 1980 and 1981, the low pressure hydrogenation unit, which is estimated to consume about 19 lb/hr of steam, came on line. This amount is approximately 0.5% change in fuel cnsumption.

(v) Towards the end of 1980, there was a switch from a heavy oil of 3500 sec to an even heavier fuel of 4500 sec. Since the heavier fuel has a relatively low calorific value, i.e. $\hat{=}$ 17300 Btu/lb, than the heavy one, i.e. $\hat{=}$ 18300 Btu/lb, there was an increase in fuel oil Consumption. This amount resulted into approximately 6% change in fuel consumption.

(vi) Change of the type of heavy oil, necessitated a change in storage temperatures of heavy fuel oil, i.e. from between 35 - 45 ^{o}C, for a 3500 sec. fuel oil, to between 45 - 55 ^{o}C, for a 4500 sec. fuel oil (information from the company's records). The resultant increase in fuel consumption due to this temperature increase (taking the datum of 35 - 45 ^{o}C storage temperature) is given by:

$$Q = m\,C_p\,\Delta T$$

Where: ΔT = change in storage temperature

From appendix A - 2, total volume of storage tanks is approximately 353 m^3. Assuming that the tanks are kept at 90% full, then the volume of fuel oil $\hat{=}$ 318 m^3.

The density of fuel oil = 998 kg/m^3

The specific heat capacity = 1.384 kJ/kgK

Therefore, Q = 318 x 998 x 1.384 x 5 = 2990 MJ
($\hat{=}$ 75 litres)

This amount resulted into approximately 0.5% change in fuel consumption.

Thus, the approximate accountable total change $\hat{=}$ 11%. The remaining 1.5% takes into account unaccountable energy consuming components and laxity in energy conservation ethics.

However, the above analysis has revealed that with this kind of historical information, it is not possible to specifically quantify the effect of individual cause. Therefore the period ending at 27th week of 1980 was dropped when assessing the effect of wind speed and solar radiation hereafter. It can be seen from Fig. B - 2 to Fig B - 9 that the graphs are alternating approximately randomly about, either the zero axis or the line parallel to the zero axis. This suggests that there was no change in the mean fuel consumption during the period starting from 28th week of 1980 to December 1983. Since we have chosen the 16.5 ^{o}C base temperature to be suitable for this situation the CUSUM charts based on 15.5 ^{o}C are given for comparison purposes.

It can also be seen from Fig. B - 1, that around the 16th week of 1982 there was a change in trend, which lasted for almost 27 weeks. It was mentioned in section 3.3.3, that only general trend should be considered i.e. not to attach unwarranted significance to short term variations, and the period of more than 10 months was recommended, thus, this change in trend was ignored.

The CUSUM chart has demonstrated its ability to detect the changes in energy consumption. If energy records are properly documented it is possible to deduce the cause of the changes, which was not totally possible in our case. However, by using following equation;

$$Q = 12803 + 215DD + 7RW - 5GR,$$

the company can be able to estimate its energy consumption with an accuracy of around 90 $\pm$ 10% (see appendix A.3 for significance interval). By constructing the CUSUM chart as potrayed in appendix B, they can be able to monitor and hence control their fuel consumption. The CUSUM chart will also help them to improve their energy documentation system, this is essential because, however small the cause may be should be documented, as it could be a big potential of energy loss. The CUSUM chart will also help them to assess the success or failure of the energy conservation programme if they will start one.

CHAPTER 7

DISCUSSION AND CONCLUSION.

7.1 The Degree Day Method.

The degree day method assumes, that energy transfer over time is linearly related to inside-to-outside temperature differences. It was shown, in section 6.4.1 et sec., that, in actual fact, there is a strong correlation between the degree days and the fuel consumption. However, any load calculation procedure using only daily mean temperature as a climate descriptor will not account for the temperature swings within the day.

It is apparent that the degree day method considers the temperatures that fall below the base temperature and says nothing about the influence of the temperatures that fall above the internal temperature upon the degree day figure. The heat gain due to such instanteneous temperature swing could be substantial, particularly so, when the area of glazing constitutes about 20% and above, of the external wall area.

Consider a typical schematic outside temperature profile (for a particular day within a heating season), Fig. 7.1. According to the definition of the degree day the energy consumed by a space heating plant will be proportional to the area under the base temperature line. Therefore for the portion ABDE no heat is supposed to be added into the building but since the outdoor temperature is still less than the indoor temperature there will be some heat flowing out of the building. The degree day method states that this difference in the internal and base temperature is compensated by the incidental heat gains. On the other hand for the portion BCD the outside temperature is greater than the indoor temperature, this will cause heat to flow into the building or rather, it will result into an increase in external wall temperature, thereby reducing the fabric heat loss. The amount being dependent upon the thermal properties of the building. In addition to that, since the offices in the U.K. open at 9.00 am. and close at 5.00 pm., according to Fig 7.1 the mean outside temperature is above the base temperature during official hours. Although the proper degree day total for such a day is not zero, in practice, there will be no space heating for the whole day. However, this fact is not taken into account by the degree day method.

Therefore, it can be anticipated that neglecting these factors probably reduces the accuracy of the estimation method.

Daily temperature swing may be accounted for by using the degree hours or degree days obtained by accumulating degree hours. It is true that normally this data will be difficult to determine, however, since the company has a micro-computer, it can be in position to measure it. This kind of analysis will automatically take into account the daily temperature swing with relation to occupation periods.

Also it will help in evaluation of the instanteneous heat gains through the building glazing resulting from the outside temperature being greater than the internal temperatures, and this should be included in the energy equation. The energy equation will probably be of the form:-

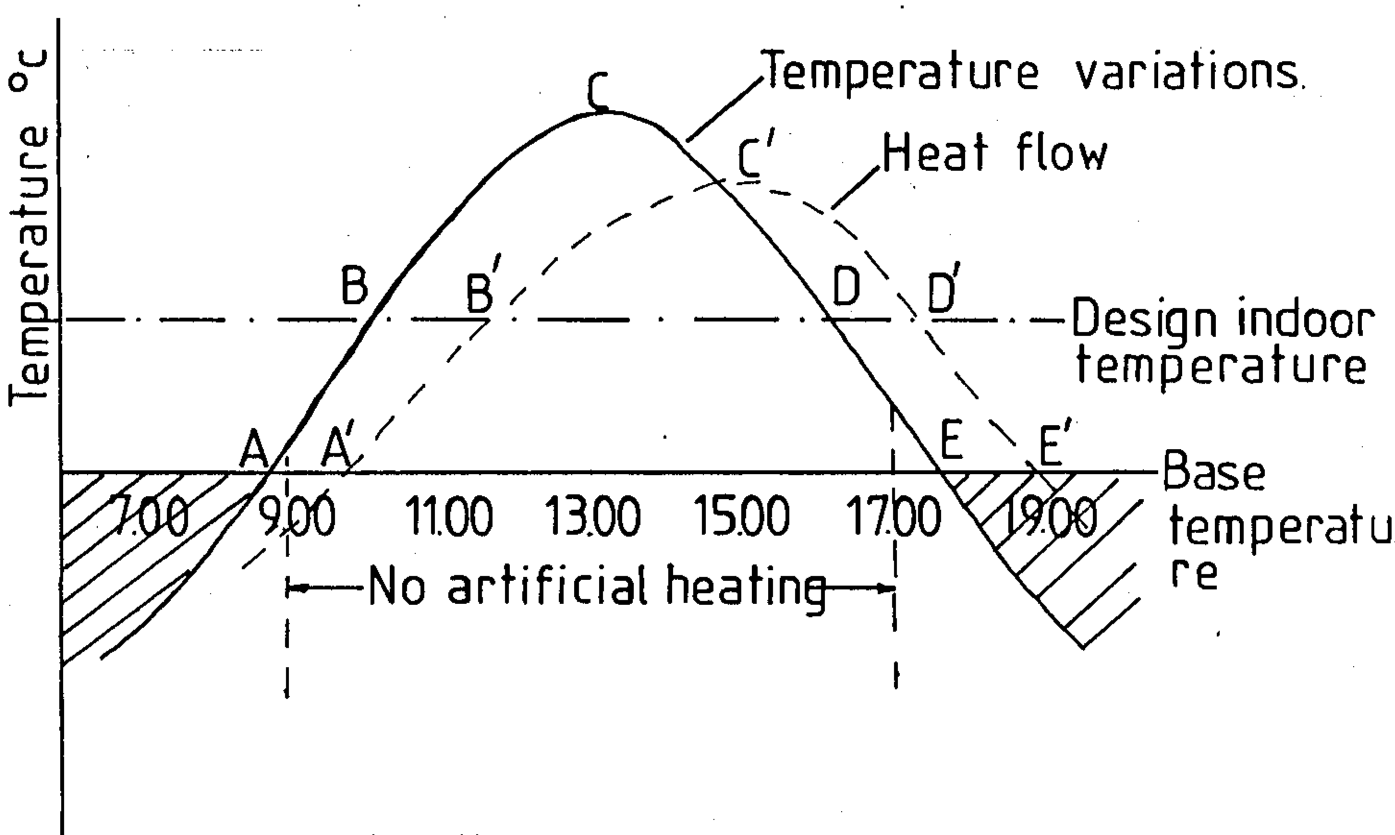


Fig.7.1: Oscillatory outdoor temperature profile.

$$Q = C_1 + C_2DD + C_3RW - C_3GR - C_4Q_1 \ldots\ldots\ldots \quad 7.1$$

Where: Q_1 = heat gain during the outside temperature being greater than the inside one.

Finally any method of estimating energy used based primarily on calculations of heat loss or gains through the building fabric, as is the case with degree day method, must of necessity, also take into account considerations of the efficiency with which the energy is used. The efficiency of utilisation, however, vary quite widely even in similar residence or plants in the same area. Thus a fixed value could be misleading because of the many factors affecting it. The efficiency of a system will depend not only on the operation of the equipment and the control arrangement but also on the operation as affected by people's habits, the net effect of infiltration, distribution pipe lengths and their level of insulation and the location of the heating unit. Very often these factors are important in determing the efficiency of utilisation for heating purposes and yet very difficult to assess. Although the efficiency of utilisation was not included in our assessment, it should be appreciated, intuitively, that the fuel consumption for space heating increases with increasing inefficiency of utilisation. Its omission, probably, has resulted into some errors in the estimation method.

7.2 Infiltration Losses and Global Radiation

Several factors, which affect the infiltration loss, have been listed, viz: building workmanship, height and structure, type of terrain, living habits of the occupants, wind speed and indoor-outdoor temperature difference. This demonstrates the immense complexity in formulating a common mathematical model for predicting the infiltration rate. All equations given in chapter 4, in one way or another give the relationship between the infiltration rate and wind speed alone. Ofcourse some other factors can not be quantified e.g. infiltration loss as related to living habits of the occupants, infiltration due to opening of doors in the morning (i.e. at the start of working session) and during lunch break, which could be enormous, and particular to each building. Thus, it is not likely in the near future to have one equation with common empirical constants, for predicting the infiltration rate, for all types of buildings. The simplification made in section 4.1.4 of ignoring the stack effect, could, to some extent, cause some errors, because the stair well doors are not perfectly sealed. However, the accuracy that could be anticipated by including the stack effect does not warrant the work involved in computing the infiltrated air due to this effect.

It has been shown that infiltration rate and hence heat loss is varying linearly with the wind speed, but the conclusion is based upon the standard error of estimates and correlation coefficients. The data that was available i.e. the energy data, the maximum and minimum outside temperatures and the run of wind, could not have been used otherwise to assess the "real nature" of the relationship between these parameters and the energy loss. However, the relationship obtained agrees with the theory that $Q \propto v^n$ where $1 \leq n \leq 1.43$. (n being equal to one in our case).

Although, the company is situated in open country, and consists of low and high rise buildings with single glazed windows which cover almost 30% of the outside wall area (which factors either increase the infiltration loss or increase heat loss from building enclosure), the effect of wind speed on energy consumption was lower than the reported values. For example Liddament (71) reported that "air infiltration is estimated to account for between 25% and 50% of the space heating requirements of both domestic and commercial buildings". The IHVE Guide and the ASHRAE book of fundamentals gave the range of between 20% and 30%. It is mentioned in the IHVE Guide that the trend towards higher levels of thermal insulation is resulting into a proportional increase in the air - infiltration component of a building heat loss. Thus, the lower values obtained in chapter 6 suggests that either this company is poorly insulated, hence energy loss is dependent almost entirely upon heat flow through building fabric due to inside-outside temperature difference or the company is to some extent perfectly built and sealed, with disciplined occupants, in the sense that the infiltration rate is negligible. The former sounds logical because of the large area of single glazing the company has i.e. $\hat{=}$ 30% of the outside wall area. Thus, it is possible that the heat loss through the glazed area is very large and this explains the reason for obtaining a small percentage effect i.e. 4.6%, in chapter 6, attributed by the wind speed. However, it was shown in section 6.4.4, that the 4.6% effect can be substantial in monetary terms.

The assumption that the weather parameters recorded at Sutton Bonington Meteorological Office are prevalent at the company site could be almost true for the wind speed, because the correction coefficients, as shown in section 4.1.2, are constant for a particular site, but for the Global radiation this assumption is not completely correct. This should be expected as the solar intensity depends upon the angle of incidence and height of the sun in the sky. These depend upon solar altitude and azimuth, which are not exactly the same for these two sites. However, there is no direct mathematical relationship between the Meteorological and local global radiation. Therefore, if the assumption is not correct the error should feature in the standard error of estimates. Otherwise, it was seen that the energy requirements vary, inversely with the global radiation.

Because of the unity admittance of the window glass, the heat gain due to solar radiation through opaque structure is very small compared with that through glazed structure, and this amount at times could be substantial. However, it was seen that the solar radiation raises the surface temperature thereby reducing the fabric heat loss, see section 5.1.6. Thus, the omission of heat gain through glazed area and the surface temperature rise, could cause errors in the equation for the energy estimation.

Therefore, if the insolation rises the external wall temperature by $\Delta T = \alpha I_G/(h_o + U)$ see section 5.1.4, this amount has to be included in the degree day calculation (i.e. the outside temperature should be $(To + \Delta T)$ rather than To). Since the global radiation is varying hour after hour, the degree hours or rather the degree days accumulated by calculating degree hours, will be a good alternative of taking care of the variation of the solar intensities.

7.3. Conclusion

The relationship between energy consumption and the weather parameters were studied in turn, and were compared with the degree day method. Different mathematical models were considered and the results show how difficult it is to assess the actual relationship between energy consumption and the weather parameters especially when the analysis is based on the weather parameters and historical energy consumption only.

However, using the old degree day method the regression equation was found to be;

$Q = 12304 + 237DD$ liters per week with a multiple correlation coefficient of 0.9573 and a standard error of 2339 litres/week (9.77% of mean weekly fuel consumption). Whereas, the resulting regression equation obtained by incorporating the weather parameters was found to be;

$$Q = 12803 + 215DD + 7RW - 5GR \text{ litres/week}$$

with a multiple correlation coefficient of 0.9637 and a standard error of 2176 litres/week (9.09% of mean weekly fuel consumption). This represent an improvement of about 7% over the old degree day method, and this improvement can be attributed to the inclusion of weather parameters in the energy equation.

However, the standard error of 2176 litres/week seem to be still high, this can be attributed by the omission of the effect of the physical properties of the building fabric- especially the glazed surface which respond rapidly, due to its unity admittance, to temperature swings or solar radiation which could be substantial at times, see section 5.1.6. Also it can be attributed by the omission of the efficiency with which the energy is utilised. This fact, although very difficult to quantify, could offset the effects exerted by the weather parameters upon energy consumption. However, it is worth noting that a 90% efficiency in energy consumption is regarded as an excellent target in an industrial atmosphere (27, 34, 48, 51).

The base load appearing in equation above, 12803 l/week, is almost in agreement with the estimated base load of 12633 litres/week, see section 6.4.1.

Ratcliffe's modified degree day method, on the other hand, gives the standard error of 2179 litres/week (9.10% of mean weekly fuel consumption). This is almost the same as that obtained by using a linear relationship above. Since the empirical constants in Ratcliffe's method are dependent upon the leakage characteristic of the building, further studies have to be carried up on several other buildings so as to obtain average values that could be used in estimation equation for all types of buildings.

It can be seen that prediction may well be improved if one considers additional relevant information in the energy equation, however, the more information added the more complicated the data becomes to handle. Thus, the degree day procedure probably will continue to be used because of its ease of application and suitability to quick estimates. With the present improvement of between 5% to 10% (although this is substantial in monetary terms see section 6.4.4) it will take a while for the people in industry to appreciate, that the results obtained are worth the efforts of collecting and handling the data that include weather data, run of wind, global radiation and physical properties. For the time being this improvement can be regarded as of no more than an academic interest.

For monitoring purposes the CUSUM technique has demonstrated its value in detecting the change in energy consumption. It can be seen that even with the gross inaccuracies,demonstrated by the mathematical models old and new alike, in assessing the energy consumption, it is still practicable to monitor the change in fuel consumption due to inefficient production, distribution and utilisation of that energy and the conservation measures. The CUSUM technique is a very simple and effective tool for communicating the energy consumption trend to the Management personnel.

8. REFERENCES

1. Aird, R.J. ; Bruffel, P.D.; Wooldridge, D.M. :"Energy consumption in dye Works", Report.

2. Aird, R.J. : "Process Energy", Lecture notes.

3. Angelino, G.; Morani, V. :"Perspective for waste heat recovery by means of organic fluid cycles", J. of Eng. for plant, April 1973, P 75.

4. ASHRAE Handbook of Fundamentals, New York, (1972).

5. Babckock, G.H.; Wilcox, S. : "Steam/its generation and use", Babckock and Wilcox Company, (1972).

6. Barber, R.E. :"Rankine cycle systems for waste heat recovery", Chem. Engineering. Nov.25, 1974, P 101.

7. Barker, A.H. : "The relative fuel economy of electricity, Gas, Oil and solid fuel as heating agents", JIHVE, Vol.1, 1933-34, P 344.

8. Basset, C.R. ; Pritchard, M. D.W. :"Heating", Longmans, (1968).

9. Billington, N.S.: "Building Physics: Heat", Pergamon press, (1967), P 62.

10. Billington, N.S. :"Thermal insulation of building", Building Services Eng., Vol.42, (1974), P 219.

11. Billington, N.S. :"Thermal insulation and thermal capacity of building", Building services Eng, Vol.43,(1963), P 226.

12. Billington, N.S. :"Estimation of annual fuel", JIHVE, Nov. 1966, P 253.

13. Blomsterberg, A.K.; Harrje, D.T. : "Approaches to evaluation of air infiltration energy losses in buildings" ASHRAE.

14. Boyen, J.L. : "Practical heat recovery", John Wiley and sons, (1975).

15. Briley, G.C. : "Conserve energy - refrigerate with waste heat", Hydro. Processing, May 1976.

16. The British Petroleum Company Ltd.: "Statistical Review of the world oil industry",(1965 - 1982).

17. Building Research Station digest: "The assessment of wind loads", Digest 119, July 1970.

18. Building Research Establishment Digest: "Principal of Natural ventilation", Digest 210, Feb. 1978.

19. Bull, G.M.; Freeman, W.; Garland, S.J. : "Specification for standard BASIC", Percy Brothers Ltd. (1973).

20. Butler, P.A. : "A showpiece to save money by promoting waste heat recovery", The Engineer, Vol.241, 23 Oct. 1975.

21. Chase, J.; Lucas, N.J. D.; Murgatroyd, W. : "Industrial energy Use - II, Energy use in a light engineering factory", Int. J. of energy research, vol.2, 1978, P 375.

22. Chatfield, C. : "The analysis of time series theory and practice", Chapman and Hall ltd., (1975).

23. Cleland A.C. ; Earle, M.D.; Brag, I.F. : "Application of multiple linear regression analysis of data from factory energy", J. of food Tech. Vol.16, 1981, P481.

24. Coblentz, C.W.; Achembach, P.R. : "Field Measurements of air infiltration in few electrically heated Houses" ASHRAE J., July 1963.

25. Danter, E. : "Heat exchange in a room and the temperature of the room", Building research establishment current paper, Feb.1975.

26. Degids, W.F. ; Mvanshijdel, L.L.; Ton, J.A. : "Wind funnel and site pressure distribution on a house and its effects on infiltration", ASHRAE, 1979.

27. Department of Energy : "Fuel Efficiency booklets", Booklets number 1 - 18.

28. Department of Energy: "Energy savings; the fuel industries and some large firms", Energy paper No.5, HMSO, 1975.

29. Department of Energy: "Energy research and development in the U.K.", Energy paper No. 11, HMSO, 1970.

30. Department of Energy: "ACEC industry Group", Report No.15, HMSO, 1976.

31. Diamant, R.M.E.: "Total energy", Pergamon Press Oxford, (1965).

32. Diamant, R.M.E. : "Energy storage part I" HVE Dec.1981, Jan.1982, P 14.

33. Diamant, R.M.E. : "Energy storage Part II", HVE, Feb.1982, P 18.

34. Dryden, I.G.C. : "The efficient use of energy", IPC Science and Technology press, (1977).

35. Edwards, R.A. : "The energy audit procedure and potential", Chart. Mech. Eng., Vol.26, n 6, June 1979, P 61.

36. Ezekiel, M.; Fox, K.A. : "Method of correlation and regression analysis", John Wiley and son Inc., (1959).

37. Filliben, J.F. : "Reducing Energy requirements in industry", ASHRAE, 1981, Part 1, P.717.

38. Freund, J.E. : "Modern Elementary Statistics", 3rd Edn. Prentice - Hall inc., Englewood cliffs, New Jersey, (1967), P.331.

39. Gerrard, A.M. : "Another look at forecasting energy requirements", Chem.Eng., May 30, 1983, P.71.

40. Gerhart, K. : "Model investigations of the distribution of convective heat transfer over building faces", Kaltechnik, Vol 5, 1967, P.122.

41. Goosens, G. : "Principles governing energy recovery", Hydro. Processing, Aug. 1978, P.133.

42. Govidan, T.S. : "The Management of energy conservation", Energy progress, Vol.4, n 1, March 1984, P.9.

43. Grant, C.O. : "Energy conservation in the Chemical and Process Industries", I Chem.E. (1979).

44. Grierson, R. : "Degree days - Proposed basis for estimating the consumption of energy in space heating installations",JIHVE, 1940 - 41.

45. Griffin, J.M. ; Steel, H.S. : "Energy economics and Policy" Academic Press Inc. (1980).

46. Grover, G.M. : "Heat pipes", Symposium - Efficient use of fuels in Process and Manufacturing industries, April 1974, P 189.

47. Guntermann, A. : "A simplified Degree day Method for Commercial and Industrial Buildings", ASHRAE TRANS., 1981, Part 1, P.461.

48. Haage, P.R. : "Improving energy usage", Energy progress, Vol.3, n 11, March 1983. P.20.

49. Hager, N.E. Jr. : "Estimation of heating energy requirements", ASHRAE J., Aug.1962, P.40.

50. Harris, J. : "The economic value of insulation", maintenance Engineering, April 1975, P.24.

51. Hardy, C.W.E. : "Efficient use of energy - combustion control", HVE, Feb. 1979, P.16.

52. Harrington - Lynn, J. : "The admittance procedure; intermittent operation", Building service Engineering, Vol.42, 1974, P.219.

53. Hopkins, P.S. ; Marriot, A.P. :"Program for energy conservation", Energy in the '80s, Ichem E symposium series, n 48, P.18.1.

54. Hunt, C.M. : "Air infiltration: a review of some existing measurement techniques and data", American Society for testing and materials, 1980 P.3.

55. Hütte : "The Engineers' Handbook, Vol.1 theoretical fundamentals", Berlin 1955, P.499.

56. ICI monographs n3. : "Cummulative sum Technique", Oliver and Boyd, (1964).

57. Institute of Fuel, "Waste heat recovery" Chapman and Hall Ltd., (1963).

58. IHVE Guide, Book A, (1970),

59. Jackman, P.J. : "Heat loss in building as a result of infiltration", Building Research Establishment, current paper, April 1974.

60. Jackman, P.J. : "Natural ventilation principles in design", Build.Serv. Res. and information, 1969.

61. Jackman, P.J. : "A study of the natural ventilation of tall office buildings", IHVE, Oct. 1970.

62. Jacobs, W.P. :"Forecasting energy requirements", Chem.Eng. March 9, 1978, P.97.

63. Jensen, J. :"Energy storage", Newnes - Butter-Worths and Co. (Publishers). (1980).

64. Kaiser, V. : "Energy Optimisation by exergy analysis", Chem.Eng. Feb.23, 1981, P.63.

65. Kantyka, T.A. : "Energy Conservation in Chemical industry", Dept. of industry energy seminar, New castle 1976.

66. Kern, D.Q. : "Process heat transfer", McGraw Hill, (1950).

67. Ladomatos, N.; Lucas, N.J.D.; Murgtroyd, W. : "Energy use in industry - V : Energy use in a heavy Engineering Workshop", Int. J. of energy research, Vol.14, 1980, P.377.

68. Lewis, G.E.H. : "Factory steam plant - a practical guide to fuel economy", Sir Isaac Pitman and sons Ltd., (1949).

69. Lewis, W.E. "Problem solving principles for BASIC programmes, Applied logic, Psychology and Grit", Hayden Book company Inc., (1981).

70. Legge, R.F.; Claudle, P.G. : "Energy conservation on Industrial based approach", Energy in the '80s, I Chem E. Symposium series n 48 P.20.1.

71. Liddament, M.W.: "The air infiltration - centres program of model validation", ASHARAE TRANS, 1983, Part 2 B.

72. Loudon, A.G.; Danter, E. : "Investigation of summer over-heating at the building research station", England.

73. Loudon, A.G. : "Window design criteria to avoid overheating by excessive solar heating gains", Building research station current paper, 4/68, 1968.

74. Loudon, A.G. : "Summertime conditioning", Buil.Res. station Current paper 47/68, 1968.

75. Lyle, O. "The efficient use of steam", 11th impression, HMSO, London, (1974).

76. Mai, K.L. : "Energy and Petrochemical feed - Stocks a view of the future", Energy Progress, viz. no.4 Dec.1983, P.206.

77. Mcdowall, D.; McCleary, R.; Meidinger, E.E.; Itay, R.A. : "Interupted time series analysis", series n.07 - 021, 1980.

78. Matson, J.V.: "Energy conservation through Cooling water treatment, Energy Progress, Vol.3 Sept. 83, P.182.

79. Milbank, N.O.; Harrington - Lynn, J. : "Thermal response and admittance procedure", Buil.res. Establishment, current paper 61/74, 1974.

80. Miller, L.M. : "The effect of wind speed upon heat requirements and internal temperatures", HVE, April 1978, P.5.

81. Monro, D.M. : "BASIC an Introduction to programming", Edward Arnold (Publishers) Ltd., (1978).

82. Monro, D.M. : "Interactive computing with BASIC A first course" Edward Arnold (Publishers) Ltd. (1974).

83. Monroe, E.S. : "Energy Conservation with Vacuum pumps", Chemical Engineering Progress, Oct. 1975, P.69.

84. Moon, P.W.H. ; Tasker, G.J.H. : "Continuous Monitoring of energy consumption in a larger Chemical Plant", Energy in the '90s Ichem E. symposium series n 48, P.8.1.

85. Morris, J.R.S. : "Organisation of energy conservation", Energy in the '80s I chem E. Symposium series n 48 P.15.1.

86. Murdoch, J. : "Control Charts", the Macmillan Press Ltd., (1979).

87. Northcroft, L.G.; Barber, W.M. : "Steam trapping and air venting", Hutchinson of London, (1968).

88. Ostrom, C.W. : "Time series analysis - Regression Techniques ; Series - quantitative application in the Social Science", Sage Publication, (1978).

89. Payne, G.A. : "The energy Managers' Handbook", IPC Business, (1980).

90. Pentherbridge, P.: "Limiting the temperatures in naturally ventilated buildings in warm climates", Buil. res. est. current paper, Feb.1974.

91. Pentherbridge, P. : "Sunpath diagrams and overlays for solar heat gain calculations", Building research station current paper 39/65, 1965.

92. Penwarden, A.D.; wise, A.F.E. : "Wind environment around Building", HMSO, London (1974).

93. Perry, R.H.; Chilton, C.H. : "Chemical Engineers' Handbook", 5th edn., Chapter 2, McGraw Hill.

94. Pogers, J. : "Handbook of ventilating and Air conditioning, Butterworth Group, (1976).

95. Potter, I.N. : "Effect of fluctuating wind pressures on natural ventilation rates", ASHRAE, Vol.85, Part 2, 1979.

96. Ratcliffe, R.G. : "The development of an improved degree days to be used in the prediction of building heating fuel consumption for a variety of weather conditions", M.Sc. project, 1981.

97. Reay, D.A. : "Industrial energy conservation a handbook for engineers", Pergamon press, (1977).

98. Reeves, G.A. "Degree day Correction factors : Basic value", ASHRAE TRANS., Vol.87, Part 1, 1981, P.507.

99. Roberts, M.C. : "Energy Management, the human aspects' of energy Conservation", Energy in the '80s, IchemE. Symposium, n 48 P.24.1.

100. Roux, A.J. : "The effect of weather conditions on heat transfer through building elements", Buil, research congress, 1955.

101. Sama, D.A. : "Cost saving through cost work analysis at heat exchanger", Energy Progress, V 3, n 4, P.212.

102. Schimidt, P.S. : "Microcomputer in energy Management", Part 1 Chem. Eng., Jun. 14, 1982, P.169. Part 2 Chem. Eng., July 12, 1982, P.123.

103. Shaw, C.Y. : "A method of Predicting air infiltration rates for tall building surrounded by lower structures of uniform height" ASHRAE TRANS., Vol.85, Part 1, 1979, P.72.

104. Scheman, M.H.; Grmsrud, D.T.; Diamond, R.C.D. : "Infiltration pressurization correlations surfaces pressures and terrain effects" ASHRAE TRANS., Vol.85, Part 2, 1973, P.458.

105. Shiskey, F.G. : "Control Systems can serve energy" Chem.Eng. Progress, May 1978, P.43.

106. Silverman, J. : "Energy Storage", Pergamon Press Oxford, (1980).

107. Singh, R.P. : "Energy Consumption in food sterilization", Food tech., Vol.31, n 3, P.977, P.59.

108. Singh, R.P. : "Energy accounting in food process operation" Food tech., April 1970, P.40.

109. Spiegel, W.P. "Challenge - Engineering and the conservation ethics", ASHRAE J. May 1975, P.25.

110. Streich, H.J. ; Fanaritis, J.P. : " Heat recovery in Process Plants", Chem.Eng. May 28, 1978, P.78.

111. Sutton, G.P. : "Save energy with pumps", Hydrocarbon Processing, June 1978, P.103.

112. SzoKalay, S.V. : "Solar energy and building", The archtectural press Ltd., (1976).

113. Tamura, G.T. : "The calculation of house infiltration rates", ASHRAE TRANS. 1979, P.58.

114. Thomopolos, N.T. : "Applied forecasting methods", Prentice and Hall inc., (1980).

115 Valentine, A.C. ; Wildman, S.V. : "Energy Conservation from steam turbines", the Chem. Eng., Sept. 1975, P.516.

116. Verschave, J.A. : "Designs and operations of industrial boilers for maximized fuel economy", Symposium on energy use of fuels in process and manufacturing industries, April 1974, P.165.

117. Weston, H.B. : "Energy a way of thinking", Maintenance Eng. April 1975, P.54.

118. Wilkins, C.L.; Kioplenstein, C.E. ; Isenhon, T.C.; Jurns, P.C. : "Introduction to computer Programming for Chemists, BASIC version" Allyin and Bacon Inc. (1974).

119. Wilson, A.C. : "Industrial thermal insulation materials application methods, specifications" McGraw Hill book company Inc. (1959).

120. Woodward, R.H. ; Goldsmith. ; "Mathematical and statistical techniques for industrial cumulative sum Techniques" ICI, (1964).

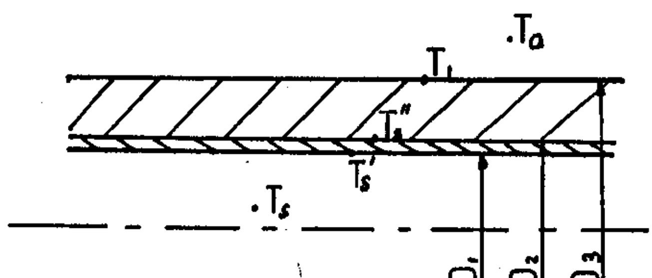
9. APPENDICES

APPENDIX A

A. 1. Heat Loss From the Pipeline to the Clinical Supplies Unit

The clinical supplies unit was built under the Company expansion programme, and came on line in 1979. Thus it is located some few metres from the main site, and more than a hundred meters from the boiler house. The steam for this unit is supplied at 100 psi by a 160 metre long, 4" diameter pipeline, which was installed some time in 1978, at a rate of 150 lb/hr. Although this pipe is lagged with a 1" fiber glass insulation and passing through a tunnel, which is not perfectly sealed it was anticipated that there is a substantial amount of heat loss along it. However, because of the tunnel, it is expected that the heat loss per metre length will be approximately equal to the heat loss per metre length from the pipe placed in still air.

Thus, it was decided that the economics of supplying steam from the central boiler system should be assessed against an alternative way of installing a small package boiler in the clinical supplies unit. This section is devoted to this aspect.



The resistances to heat flow from a pipe carrying steam at temperature Ts, taken in order are;

(a) The resistance of steam to condense upon and give up heat to the inner pipe surface, this resistance is very small, so that the inner surface temperature of the pipe T_s' is equal to the temperature of the steam.

(b) The resistance of the pipe metal which is very small except for thick walled conduit, so that the temperature of outer surface of the pipe T_s'' is approximately equal to that of steam.

(c) The resistance of the insulation.

(d) The resistance of the surrounding air to remove the heat from the outer surface.

To simplify the calculations the outer surface temperature T_1, was measured by a contact thermometer. Ten readings, measure roughly 15 metres apart, along the pipe line are shown in table A - 1.

Table A - 1: Temperatures measured along the pipe line.

Reading	1	2	3	4	5	6	7	8	9	10	Average
Temperature (°C)	29.5	30.8	28.9	33.3	35.2	29.8	32.6	31.6	33.5	34.5	32

Heat Loss from a pipe per meter length is given by;

$$q = \frac{2\pi K_{ins}\,(T_s - T_1)}{\ln \frac{D_3}{D_2}} \qquad \ldots\ldots\ldots \qquad A.1$$

Data:

Pipe length = 160m (measured)

Pipe diameter; Od = 114.3mm

id = 102.26mm

Insulation thickness = 25.4mm

Type of insulation : Fiber glass, thus $K_{ins} = 0.039\ \frac{W}{mK}$

Steam pressure 100 psi ($T_s = 169.9°C$)

substituting the values in equation A.1

$$q = \frac{2\pi \times 0.039\ (169.9 - 32)}{\ln \frac{165.1}{114.3}} = 91.9\ W/m$$

The heat loss per metre length = 91.9 W/m.
Comparing this value with 96 W/m obtained form reference 27 for a steel pipe with similar dimensions and insulation situated in still air, the assumption made earlier is correct.

$$\text{Total heat loss} = 91.9\,\frac{W}{m} \times 160\ m \times 3600\,\frac{s}{hr} \times 24\,\frac{hr}{day} \times 7\,\frac{day}{week}$$

$$= 8893\ MJ/Week\ (84.5\,\frac{therms}{week}).$$

Weekly average fuel consumption

$$= 2\ 3332\,\frac{litres}{week} \quad (9076\,\frac{therms}{week}).$$

Thus, 0.93% of the total heat supplied per week is lost along the pipe line. This amount is quite insignificant in practice, but the clinical supplies unit is supplied with steam at a rate of 150 lb/hr (300 therms/week). Thus, 28% of the total heat supplied to the clinical supplied unit is lost along the pipeline. In monetary terms, if we assume the price of £4.5 per GJ and a boiler efficiency of 80% this loss costs the company around;

$$\frac{8.893 \times 4.5}{0.8} = £50 \text{ per week}$$

or approximately £2500 per year.

If the gas fired package boiler, which costs around £2850 (from the price list given by Frema combustions Limited) is purchased to supply steam within the clinical supplies unit, assuming a pessimistic figure of £1000 for installation costs and some minor modifications the company will need a capital investiment of around £4000 with the pay back period of 1.5 yrs. which is quite reasonable.

Conclusively, the company can economically use up to £5000 with a pay back period of around 2 years, which is normally accepted by industries to be reasonable, to supply the steam within the clinical supplies unit rather than from the central boiler system.

A.2 Heat Loss from Oil tanks

The calculations were made in order to assess the effectiveness of the insulation. Besides, this amount of heat loss features in the base load of the company and thus, has to be known. Contact thermometers were used to measure both the skin and surface temperatures of the storage tanks. The measurements were taken randomly along the circumference and the height of the tanks.

Tank number 1

height = 8.22m

circumference = 11.58m

diameter = 3.69m

insulation thickness = 57.15mm

					Average
Surface Temperature (°C)	14	14	14	13	13.8
Skin Temperature (°C)	27	25	29	25	26

Heat loss : $$Q = \frac{A \Delta T}{\frac{X}{k}} \quad \ldots\ldots\ldots\ldots\ldots \quad A.2$$

Where: X = insulation thickness

$$A = \pi dh + \frac{2\pi d^2}{4}$$

Thus, $$A = \pi \times 8.22 \times 3.69 + 2\pi \times \frac{3.69^2}{4} = 117 \text{ m}^2$$

and $$Q = \frac{117 \times 0.039\ (26 - 13.8)}{0.05715} = 589 \text{ MJ/week}$$

(14 litres/week)

Tank number 2

height = 9.28 m

circumference = 10.13 m

diameter = 3.22m

insulation thickness = 50.8mm

Surface Temperature (^oC)	14	13	12	15	12	13	Average 12.5
Skin Temperature (^oC)	29	30	35	34	27	26	30.2

$$A = 110.29 \text{ m}^2$$

$$Q = \frac{110.29 \times 0.039\ (30.2 - 12.5)}{0.0508} = 906 \text{ MJ/week}$$

(21.5 l/week)

Tank number 3

Similar dimensions and insulation to tank number 2.

Surface Temperature (^oC)	17	16	17	13	13	13	Average 14.8
Skin Temperature (^oC)	30	32	35	30	32	35	32.3

$$Q = \frac{110.29 \times 0.039\ (32.3 - 14.8)}{0.0508} = 896 \text{ MJ/week}$$

(21.3 l/week)

Tank number 4

height = 8.22m

circumference = 13.24m

diameter = 4.2m

insulation thickness 25.4mm.

Surface Temperature (^oC)	13	11	12	12	14	14	Average 12.7
Skin Temperature (^oC)	25	22	26	25	26	26	25.3

$$A = 137 \text{ m}^2$$

$$Q = \frac{137 \times 0.039\ (25.3 - 12.7)}{0.0254} = 1603 \text{ MJ/week}$$

(39 l/week)

Total heat loss = 3994 MJ/week

(95.8 litres/week)

This amount is within reasonable limits as it constitute around 0.4% of total fuel consumption per week.

A.3 Confidence Interval

A $1-\alpha$ confidence interval for the true standard error is such that, using a two sided significant test all values of σ within the confidence interval do not produce a significant discrepancy with the data at the chosen level of probability α, whereas all the values of σ outside the interval do show such dicrepancy. The quantity $1-\alpha$ is sometimes called the confidence coefficient.

The interval is given by :-

$$\text{Pro.}\left[\frac{S_e}{1+\frac{t_{\alpha/2}}{\sqrt{2n}}} < \sigma < \frac{S_e}{1-\frac{t_{\alpha/2}}{\sqrt{2n}}}\right] = 1-\alpha \quad \cdots\cdots A.3$$

Where : S_e = estimated standard deviation (2176 litres/week)

σ = standard error of estimate

n = number of data lines

Thus, if we want to assert with a probability of 95% the range of σ, then:

$$1-\alpha = 0.95 \implies \alpha = 0.05$$

From statistical table (38), $t_{\alpha/2} = t_{0.025} = 1.96$

substituting the values in equation A.3 we have

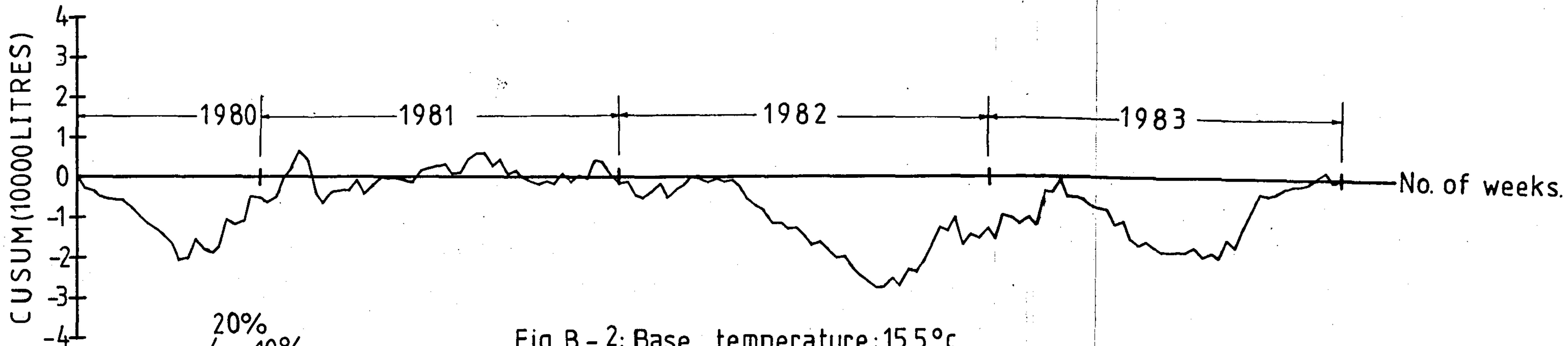
$$\frac{2176}{1+\frac{1.96}{\sqrt{2\text{x}159}}} < \sigma < \frac{2176}{1-\frac{1.96}{\sqrt{2\text{x}159}}}$$

i.e. $1961 < \sigma < 2445$

$\hat{=} \quad \sigma \pm 10\%$

i.e. we assert with a probability of 0.95 that the interval from 1961 to 2445 litres per week contain σ.

APPENDIX B



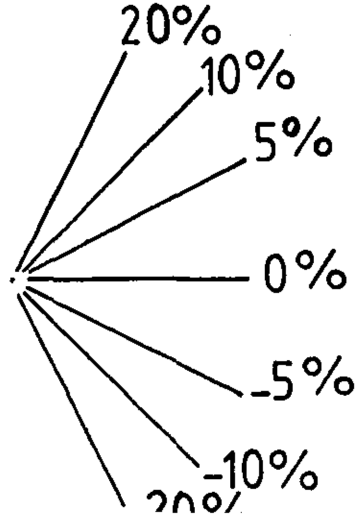


Fig. B - 2: Base temperature: 15.5 °c
y = f (DD)
Regression equation is
y = 13273 + 247DD
Correlation coefficient = 0.9554
Std error = 9.98% of mean.

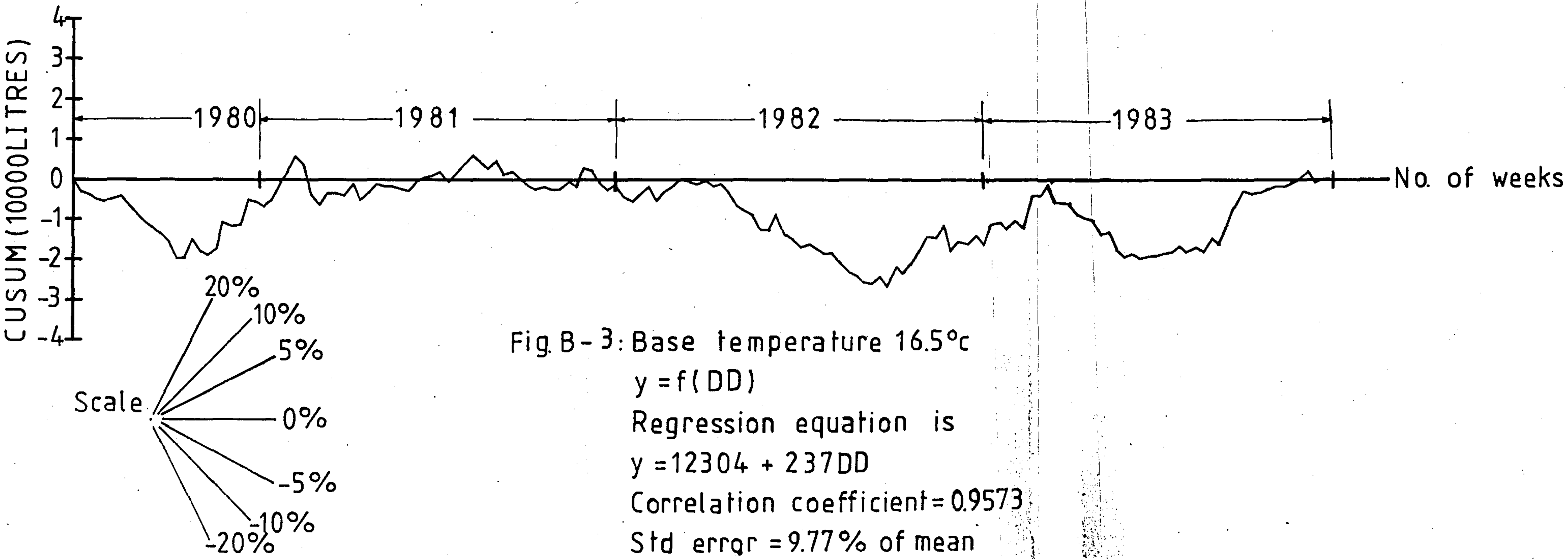


Fig. B-3: Base temperature 16.5°c

$y = f(DD)$

Regression equation is

$y = 12304 + 237DD$

Correlation coefficient = 0.9573

Std error = 9.77% of mean

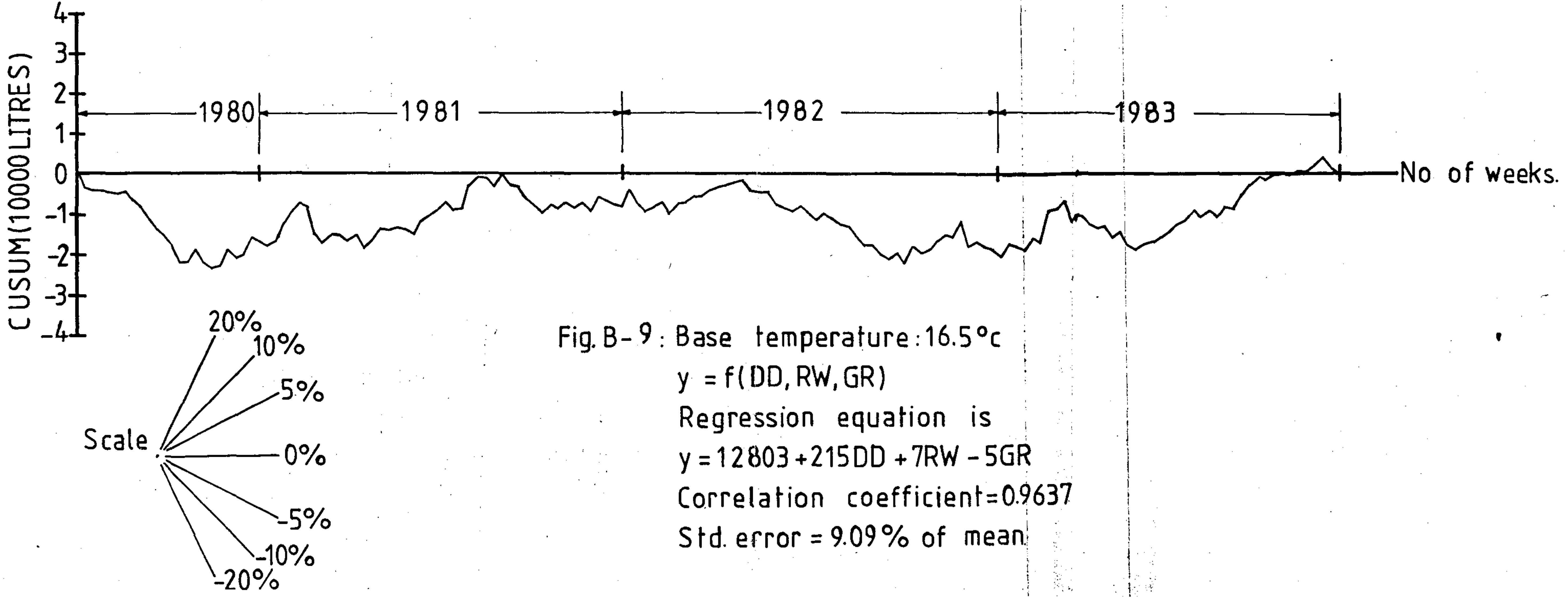


Fig. B-9 : Base temperature : 16.5°c

$y = f(DD, RW, GR)$

Regression equation is

$y = 12803 + 215DD + 7RW - 5GR$

Correlation coefficient = 0.9637

Std. error = 9.09% of mean

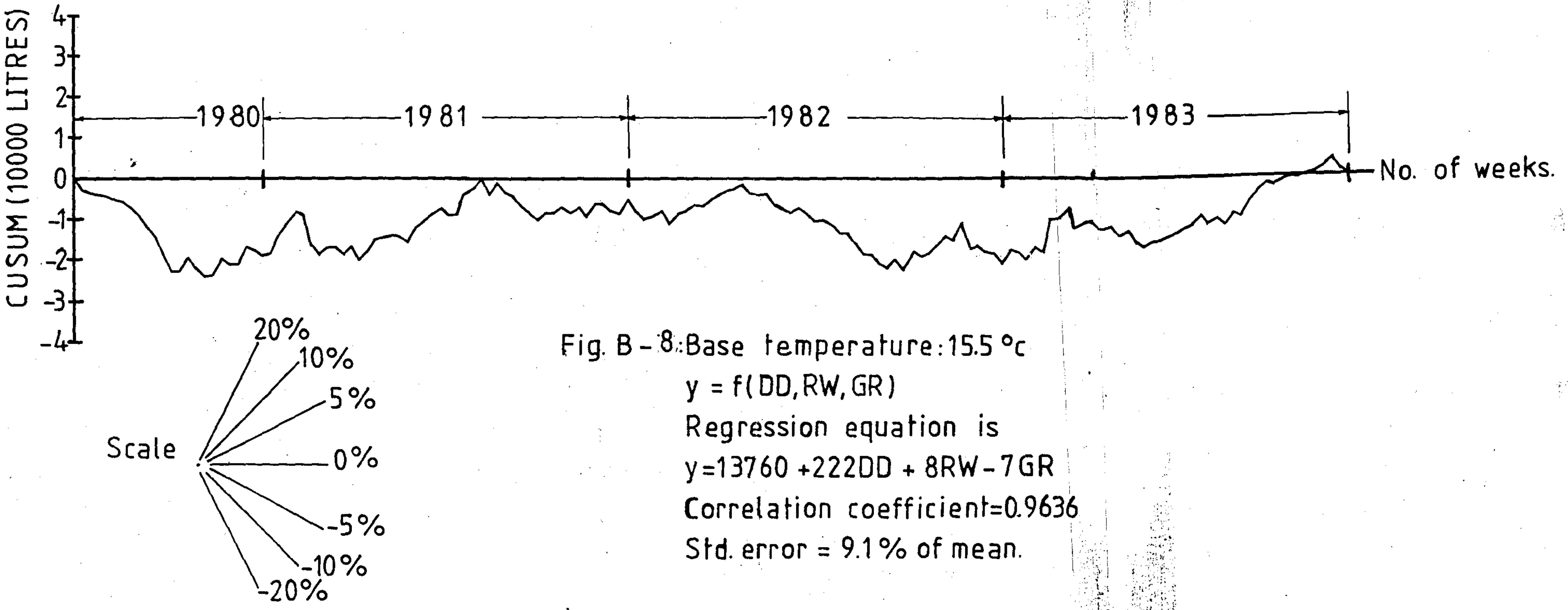


Fig. B - 8: Base temperature: 15.5 °c

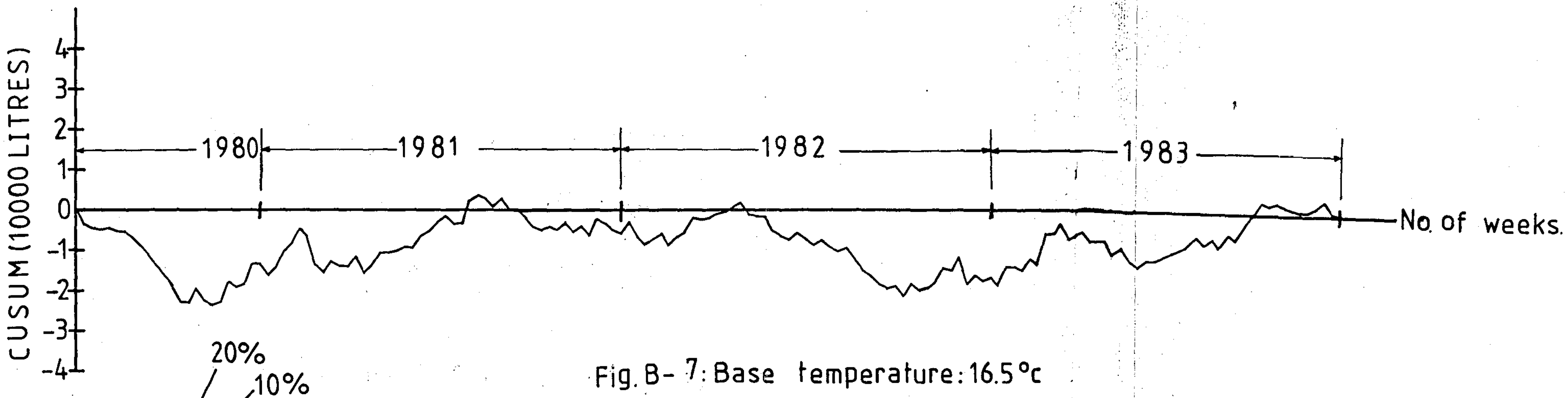
y = f(DD, RW, GR)

Regression equation is

y = 13760 + 222DD + 8RW - 7GR

Correlation coefficient = 0.9636

Std. error = 9.1% of mean.



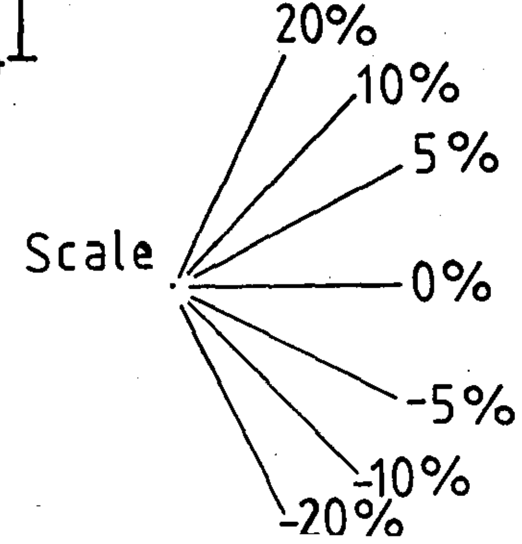
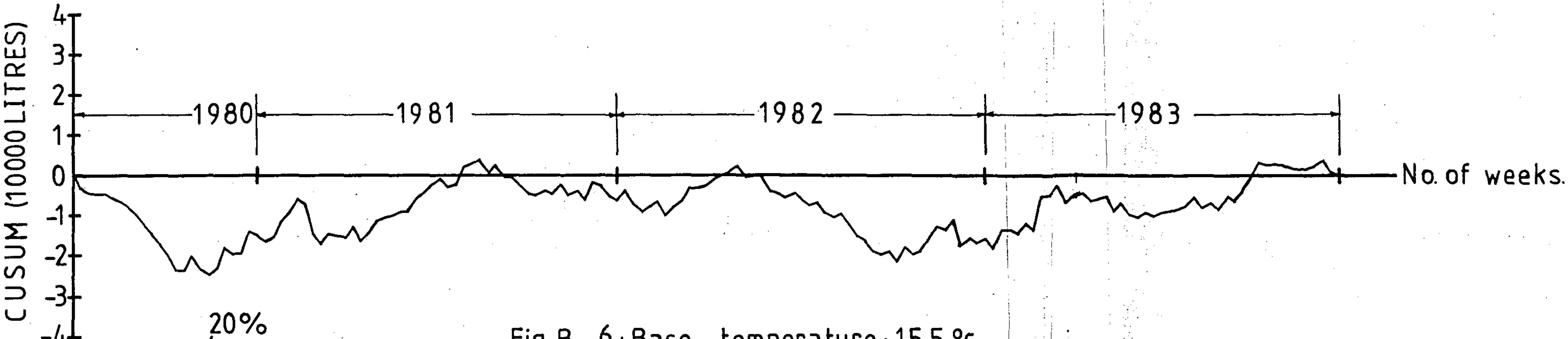


Fig. B- 7: Base temperature: 16.5 °c
y = f(DD, GR)
Regression equation is
$y = 15111 + 213DD \quad 7GR$
Correlation coefficient = 0.9614
Std error = 9 33% of mean.



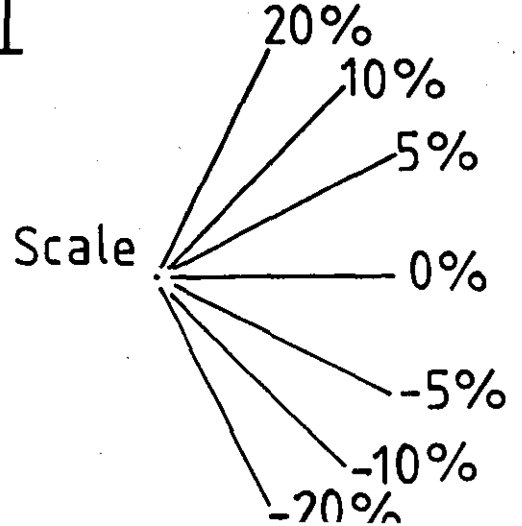


Fig. B- 6 : Base temperature : 15.5 °c

$y = f(DD, GR)$

Regression equation is

$y = 16327 + 219 DD - 7 GR$

Correlation coefficient = 0.9608

Std error = 9.40 % of mean.

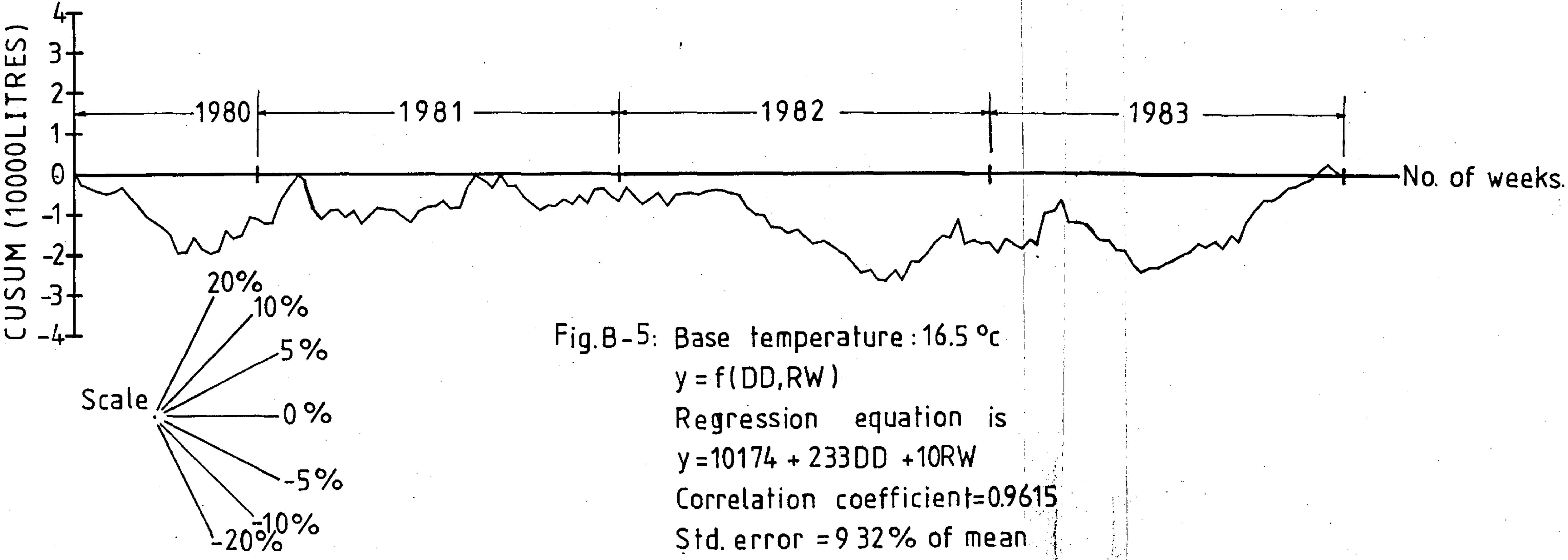
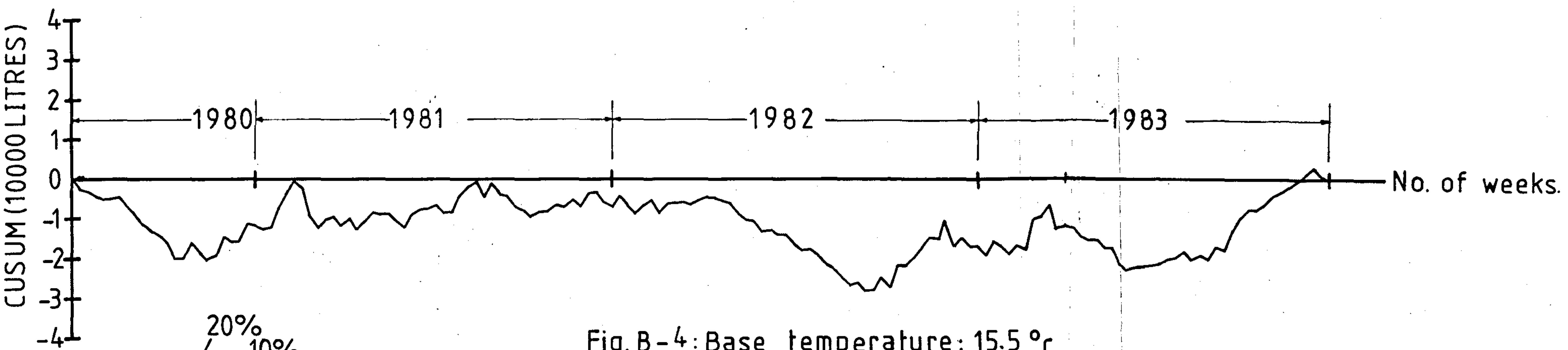


Fig.B-5: Base temperature: 16.5 °c
y = f(DD,RW)
Regression equation is
$y = 10174 + 233DD + 10RW$
Correlation coefficient=0.9615
Std. error = 9 32% of mean



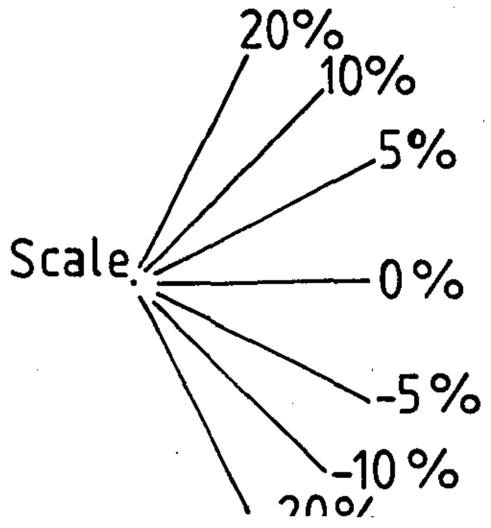


Fig. B-4: Base temperature: 15.5 °c
y = f(DD, RW)
Regression equation is
$y = 10842 + 242DD + 11GR$
Correlation coefficient=0.9608
Std. error 9.41 % of mean

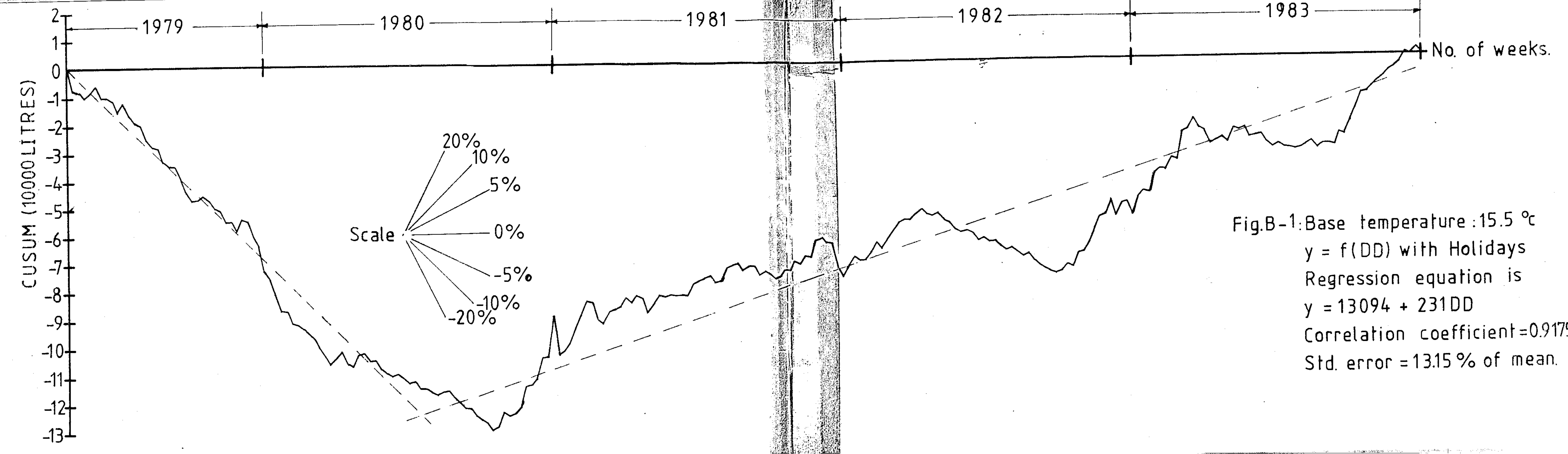


Fig.B-1: Base temperature : 15.5 °c
y = f(DD) with Holidays
Regression equation is
y = 13094 + 231DD
Correlation coefficient = 0.9175
Std. error = 13.15 % of mean.