DESIGN OF SANITARY AND MECHANICAL SERVICES SYSTEMS IN A DAIRY PLANT

BY

JOSEPH IFEANYI NJOKU

A THESIS SUBMITTED TO THE POSTGRADUATE SCHOOL, AHMADU BELLO UNIVERSITY, ZARIA IN PARTIAL FULFILLMENT FOR THE AWARD OF THE DEGREE OF:

M.Sc (BUILDING SERVICES)

OCTOBER, 1984.

DECLARATION

I declare that this thesis has been written by me and that it is a record of my own research work. It has not been presented in any previous application for a higher degree. All quotations are indicated and the sources are specifically acknowledged by means of references.

J.I. NJOKU
October, 1984

AHMADU BELLO UNIVERSITY, ZARIA
DEPARTMENT OF BUILDING

We certify that we have read this Thesis and in our opinion, it conforms to acceptable standards of scholarly presentation and is fully adequate in scope and quality as a thesis for the degree of Master of Science in Building Services.

Sign:_____
Mr. J. Ciborski
(Internal Examiner)

Sign:_____
Mr. T. Sroka
(Internal Examiner)

Sign:_____
Name:_____
Date:_____
(External Examiner)

This Thesis was submitted to the Faculty of Environmental Design, the Post Graduate School and the Senate of Ahmadu Bello University, Zaria

Sign:_____
Tim. Mosaku (Dr.)
Head of Department of Building

Sign:_____
G. Osuide (Professor)
Dean of PostGraduate School

Date of Approval/Examination_____

ACKNOWLEDGEMENTS

The author is grateful to the Almighty God for inspiration, power and good health granted him throughout the period of the work.

This work has been made possible also due to the excellent cooperation that he had from so many people, to whom he is highly indebted. Prominent among these are:

1. Mr. J. Ciborski - A specialist in Engineering services in building. He was a very good adviser. He took tgreat troubles to guide and correct me all through the work.

. Mr. Yinka Obadofin - My noble colleague and friend who always cheered me up when the going gets too tough. He was kind enough to proof read my work.

. Production Managers of Nigeria Dairies Limited, Kaduna; Madara Dairies Limited, Jos; Minna Dairy Plant, Minna; Northern Dairies Limited, Funtua - They were highly cooperative. They kindly answered all my questions and took me right round their entire works with comprehensive explaination.

ABSTRACT

This thesis is on the design of the building services requirement of a dairy plant. Investigation of the engineering services problems of four existing medium scale dairy plants at Kaduna, Funtua, Jos and Minna showed that the greatest problem of the dairy industry is sanitation, and the building services requirement being: Water (cold and hot) services, sewage disposal, Refrigeration and Airconditioning. The work therefore tried to optimize the design of these services based on production capacity of 18,000 litres of milk products per shift of 8 hours. Factors considered in the design included the number of people that work in the plant, the processes that the milk products undergo in the industry which determine temperatures of the various rooms, the hot and cold water demand the nature of wastes and waste treatment, and the refrigeration capacity. The work investigated pipeline and insulation problems and specified materials and sizes. The production capacity and processes determined the machine and equipment specifications. This in turn determined the plant layout and manpower requirement as shown in the organisation chart. The building services were then designed to suite the plant layout. Detailed designs of the plant layout and these building services are shown in Drawings 1 to 6

(folded and attached at the back cover). The various component designs are illustrated in figures 1 – 43 The design of the storm drainage system is based on the severest rainfall intensity in Kaduna as shown in tables 4-13 extracted from the Agro-Meteorological Bulletin of Nigeria (1980-1983).

THE AUTHOR

The author, Joseph Ifeanyi Njoku was born on June 23, 1953.

He attended St. Clement's Primary School, Onicha Mbaise from 1959 to 1965. And from 1966 to 1971 (civil war intervening), he attended St. Paul's Secondary School Ikenanzizi. He passed the West African School Certificate Examination in Division One. He worked in WAATECO (Nig) Ltd., Kano for a brief period before attending the University of Nigeria Nsukka, where he studied Mechanical Engineering from 1973 - 1978; and passed in Second Class Honours.

Since graduation, the author has gained varied working experiences as follows:

1978-1979 - NYSC Lecturer, Federal Polytechnic Akure

1979-1980 - Industrial Engineer, Nigerial Industrial Development Bank Ltd., Lagos

1981-1983 Project Engineer, CADIS (Nig) Ltd., Kaduna

He is at present engaged with Post Graduate work leading to M.Sc (Building Services).

DEDICATION

This work is dedicated to the memory of my ever loving Mother who slept in the Lord on August 16, 1982.

She was a wonderful mother whose light continues to guide all her children.

J. I. Njoku

TABLE OF CONTENTS

		PAGE
TITLE PAGE		i
DECLARATION		ii
ACKNOWLEDGEMENTS		iv
ABSTRACT		v
THE AUTHOR		vii
DEDICATION		viii
LIST OF TABLES		xiii
LIST OF FIGURES		xv
ABBREVIATIONS AND SYMBOLS ...		xviii
CHAPTER ONE		
INTRODUCTION		1
1.1	Background	1
1.2	Need for the Work	1
1.3	Objectives of the Design ...	2
1.4	Scope and Methodology ...	3
1.5	Technology of Milk Production ...	3
1.6	Aseptic Production of the Dairy Products	6
CHAPTER TWO		
PLANT DESIGN		12
2.1	Design Requirements	12
2.2	Aseptic Design	12
2.3	Cleaning-in-Place	13
2.4	Design of the Cleaning-in-Place (CIP) System	13
2.5	Automation	16
2.6	Plant	16

TABLE OF CONTENTS (Contd.)

		PAGE
CHAPTER THREE		
WATER SUPPLY		18
3.1	Source of Water Supply ...	18
3.2	Water Demand Estimations ...	27
3.2.1	Deep Well Delivery	29
3.2.2	Reservoir and Pumping Capacities Estimations	29
3.2.3	Well Design	30
3.3	Water Treatment	33
3.3.1	Aeration	34
3.3.2	Filteration	34
3.3.3	Disinfection	34
3.3.4	Softening	34
3.4	Pipeline Design and Problems ...	36
3.4.1	Types of Sterlization Process ...	36
3.4.2	Direct Sterilization ...	37
3.4.3	Indirect Sterilization ...	43
3.4.4	Pipeline Design Specifications ...	47
3.5	Pipeline Problems ...	48
3.5.1	Selection of Pipe Materials ...	48
3.5.2	Head Losses in Pipelines ...	50
CHAPTER FOUR		
SEWAGE DISPOSAL		55
4.0	Background	55
4.1	Storm Water Drainage System ...	55
4.2	Details of the Design of the Storm Drainage Gutter ...	76
4.3	Sewers System	77
4.4	Sewage Treatment	78
4.5	Analysis of Sewage	78
4.6	Design of the Sewage Treatment Plant	81

TABLE OF CONTENTS (Contd.)

		PAGE
4.6.1	Screening	81
4.6.2	Design of the Screen ...	83
4.6.3	Design of the Settling Tanks ...	84
4.6.4	Horizontal Flow Tanks ...	84
4.6.5	Vertical Flow Tanks ...	88
4.6.6	Radial Flow Tanks	90
4.7	Calculation of the Capacity and size of the Primary Settlement Tank	91
4.8	Calculation of the Capacity of the Aeration and the Final Settlement Tanks	93
4.9	Specification of the Sewage Recycling Pump	93
CHAPTER FIVE		
DESIGN OF THE REFRIGERATION SYSTEM ...		94
5.1	Background	94
5.2	Estimation of the Refrigeration Capacity	96
5.3	Refrigeration Cycle ...	99
5.4	Component Specifications ...	106
5.5	Summary of Specification of Components	115
CHAPTER SIX		
HEATING, VENTILATION AND AIR CONDITIONING		116
6.1	Desired Aircondition of each of the Spaces	116
6.2	Cooling Load Estimations ...	117
6.3	Determination of the Mass Flow Rate of Chilled Water (Kg/hour) Required to Maintain the Air at the Desired Conditions	119
6.4	Determination of the Chilled Water Tank Capacity and the Chilled Water Pump Capacity	121

TABLE OF CONTENTS (Contd.)

		PAGE
6.4.1	Tank Capacity	121
6.4.2	Capacity of the Chilled Water Pump Pc	122
6.5.	Design of Stirrer for the Chilled Water Tank ...	122
6.6	Component Specifications of the Airconditioning System ...	124
6.7	Summary of Specifications ...	127
6.8	Distribution of Air ...	127
6.9	Designing the Duct Network (Using Velocity Method) ...	129
6.9.1	Velocity Method	130
6.9.2	Area of Sections	132
6.10	Equivalent Rectangular Ducts ...	134
6.11	Summary of Equivalent Ducts	137
6.12	Fan Size and Duct Layout ...	137
6.13	Distributing Cooling and Heating Media	138
	CHAPTER SEVEN	
	THERMAL INSULATIONS	142
7.1	Need for Thermal Insulations	142
7.2	Calculation of Optimal Thickness of Insulation of the Cold Room	144
7.3	Calculation of Optimal Thickness of Insulation of the Ice Cream Freezing Room .. ;..	145
7.4	Pipe Insulation	146
	CHAPTER EIGHT	
8.1	Control of the Airconditioning System	150
8.2	Control of the Water Supply System	159
8.3	Control of the Disinfection System	160
	CHAPTER NINE	
	CONCLUSION AND RECOMMENDATIONS ...	161

LIST OF TABLES

1. Table 1 - Aschaffenburg Test
2. Table 2 - Run Off Factor for Various Land Forms
3. Table 3 - 1984 Rainfall Register for Kaduna
4. Table 4 - Values of Monthly and Annual Rainfall Normals (1951-1980)
5. Table 5 - Rainfall Intensities for Heavy Precipitation (May 1981)
6. Table 6 - Rainfall Intensities for Heavy Precipitation (June 1981)
7. Table 7 - Rainfall Intensities for Heavy Precipitation (July 1981
8. Table 8 - Rainfall Intensities for Heavy Precipitation (May 1982)
9. Table 9 - Rainfall Intensities for Heavy Precipitation (June 1982)
10. Table 10 - Rainfall Intensities for Heavy Precipitation (July 1982)
11. Table 11 - Rainfall Intensities for Heavy Precipitation (May 1983
12. Table 12 - Rainfall Intensities for Heavy Precipitation (June 1983)
13. Table 13 - Rainfall Intensities for Heavy Precipitation (July 1983)
14. Table 14 - Qaulity Characteristic of Waste Water
15. Table 15 - Quality Characteristic of Treated Waste Water
16. Table 16 - Temperature Ranges of Different Rooms in the Dairy Plant
17. Table 17 - Specific Heat of Some Substances

18. Table 18 - Properties of Refrigerant 12 (Freon 12)

19. Table 19 - Halocarbon Refrigerants

20. Table 20 - In-organic Refrigerants

21. Table 21 - Sources of Heat gains

22. Table 22 - Estimation of Cooling Load

23. Table 23 - Recommended Duct Velocities of the Air conditioning System taken from 1965 ASHRAE Guide and Data book

LIST OF FIGURES

1. Fig. 1 - (a) Temperature Range of UHT Sterilization
   (b) Aschaffenburg Test
2. Fig. 2 - Indirect Working UHT PLant Temperature/Time Diagram
3. Fig. 3 - Cleaning in place (CIP) System
4. Fig. 4 - CIP for tanks
5. Fig. 5 - CIP for the Heat Exchangers
6. Fig. 6 - Scheme for Water Supply
7. Fig. 7 - Scheme of Preliminary Treatment
8. Fig. 8 - Scheme of Chlorination
9. Fig. 9 - Water Distribution System
10. Fig. 11 - Water Supply Plan
11. Fig. 12 - Gravel Packed Well with Submersible Pump
12. Fig. 13 - Steam and Dulled Water Supply Plan
13. Fig. 14 - Organization Chart
14. Fig. 15 - Methods of UHT Sterilization

15. Fig. 16 - Methods of Direct Heating

16. Fig. 17 - Method of Steam Supply

17. Fig. 18 - Pipe Arrangement for an Injection type Sterilizer

18. Fig. 19 - Proposed Modification of Fig. 18

19. Fig. 20 - Pipe arrangement in a Plate Heat Exchanger (PHE) Sterilization System

20. Fig. 21 - Pipe arrangement for Plant Using heat Exchanger

21. Fig. 22 - Energy Losses in pipes

22. Fig. 23 - Storm Water Drainage

23. Fig. 24a - Section of the Storm Drainage Gutter

24. Fig. 24b - Fall of the Gutter in degrees

25. Fig. 26a - Scheme of Waste Water Treatment Plant

26. Fig. 26b - Design of Horizontal Flow Setting Tank

27. Fig. 26c - Vertical Flow Tank Design

28. Fig. 27 - Heat Engine Cycle

29. Fig. 28 - Refrigeration cycle

30. Fig. 29 - Equipment Set-up for Refrigeration

31. Fig. 30 - Temperature-Entropy Diagram

32. Fig. 31 - Pressure-Enthalpy (P-H) Diagram

33. Fig. 32 - Chilled Water Tank Stirring System

34. Fig. 33 - Distribution Pattern of air

35. Fig. 34 - Four pipe Distribution with two Coils and space temperature ranges

36. Fig. 35 - Steam Pipe Insulation

37. Fig. 36 - Ice Water Pipe Insulation

38. Fig. 37 - Refrigerant Pipe Insulation

39. Fig. 38 - Face and Bypass Dampers Control System

40. Fig. 39 - Psychrometric Chart Entering and Leaving the Conditioner

41. Fig. 40 - Meeting the requirements Latent-Heat removal by lowering the refrigerant temperature

42. Fig. 41 - Using Reheat to Satisfy a Heavy Latent Load

43. Fig. 42a - Face and Bypass Dampers

44. Fig. 42b - Psychrometric chart of air conditions when using face and Bypass Dampers

45. Fig. 43 - Control of the Disinfection System

LIST OF ABBREVATIONS AND SYMBOLS

Abbreviations and Symbols	Meaning
ASHRAE	American Society of Heating and Refrigeration and Airconditioning Engineers
CIP	Cleaning in Place
CM	centimetre
Contd.	continued
C.O.P.	Coefficient of performance
C.V.V.	Constant vacuum valve
Dept.	Department
D.P.	Differential pressure
Drg	Drawing
Eqn	Equation
Fig	Figure
i.e.	that is
kg	kilogram
m	metre
min	minute
PH*	a number used to express degrees of acidity or alkalinity in solutions

Abbrevations and Symbols	Meaning
P - h	Pressure - enthalpy
PHE	Plate heat exchanger
RE	Refrigeration effect
s, sec	Second
S.V.	Solenoid valve
Tab.	table
UHT	Ultra heat treatment
A	Surface area
a	length of rectangular section
b	Width of a rectangular section
D, d	Diameter
f	friction factor
$\emptyset$	Diameter or angle of inclination
G, g	acceleration due to gravity = 10 ms^{-2}
h	gradient, slope or fall tan $\emptyset$ = h; specific enthalpy in KJ/kg ^{o}C
k	heat conductance of a material
L	Length of pipe
P	Pressure
τ	Stress
μ	Viscosity

Abbrevations and Symbols	Meaning
ρ	density in kg/m^3
Q, q	Quantity of water Quantity of Sewage Quantity of Seat Rate of Flow of cooling load.
V	Velocity or volume
Wc	Work of compression
W_R	Mass of refrigerant
^oC	degrees centigrade
$\therefore$	therefore
$>$	greater than
$<$	less than
$=$	equal to
$\geq$	greater than or equal to
$\leq$	less than or equal to

CHAPTER ONE
1. INTRODUCTION

1.1 Background

1.2 Need for the Work

1.3 Objectives of the Design

1.4 Methodology

1.5 Technology of Milk Production

1.1 Background

The building services requirement in the dairy industry is a function of the nature of the dairy products, the raw materials, the processes and the sterility of the environment required to produce them. But the scope of the design depends on the production capacity for any particular product.

1.2 Need for the Work

Milk serve as a very nutritious food. In the past milk was consumed fresh and after fermentation as 'yoghurt'. But it was not possible to consume all the milk in these forms with the result that so much milk resources was either not fully exploited or wasted. To guard against this, it has become necessary to find other ways of consuming milk, storing milk such as cream cheese, butter, curd and whey or converting into some industrial products such as

casein products, paints, plastics, bristles and cosmetics. So much work has been done in this direction that both the Federal and State Governments of Nigeria and enterpreanueres alike have developed tremendous interest to invest in the dairy industry. The extent to which this purpose (the purpose to fully exploit milk resources) can be achieved and the quality of the products depend on the efficiency of the building services. Hence the n eed to d esign an effective and efficient building services system for the industry.

1.3 Objective

The objective of this work is to investigate the problems of sanitary and mechanical services system in a Dairy Plant with a view to proposing an optimal design that will minimize these problems.

1.4 Scope and Methodology

This work is limited to designing the building services requirement of a medium scale dairy plant in the northern states of Nigeria. Investigation of the services problem was therefore limited to four existing dairy plants at Kaduna, Funtua, Jos and Minna. The method of this investigation being by questionnaire, visits to the four plants and interviews with the production and services managers of these plants. Through these investigations, the adequacy of the building services system as they existed was assesed; and an optimal design is then proposed.

1.5 Technology of the Milk Production

The first requirements in the production of milk are healthy, clean cows. The processes through which milk should undergo can be presented in the form of flow chart as follows:

Chilling (4-10°C). The milk collected must be cooled immediately and the termperature maintained until processed.

Clarification (filteration). In this process all the foreign solid bodies beyond the size of 0.3 micron are removed.

Standardization. The fat content is increased to 3.5% for pasteurized milk through evaporation or to 10.5% for Yoghurt through evaporation and/or addition dry milk powder.

Heat treatment (including homogenization - breaking down fat globules from 3mm to 0.02 - 0.3 micron, (4-75°C)

Cooling to storage temperature or for futther heat treatment (ultra heat treatment) ranging up to 150°C.

In order to produce Yoghurt, 4% starter culture is added to the pasteurized milk and allowed to incubate at the temperature of 40-45°C for 2-3 hours. After incubation the product is cooled at low temperature below 10°C.

Yoghurt needs the higher solids content. Hence skim milk is necessary. In fresh milk solids content is about 2.5% but for Yoghurt making solids content of about 10.5% is needed. It is important for the solids content to be high for the following reasons:

1. It improves the consistency of the product.
2. It provides more of the sugar (Lactose) for the fermentation process.
3. Stability will be enhanced both during storage and transportation.
4. It prevents or minimizes whey separation in the final product.

Hence 1-2% skim milk should be added to the full cream milk for the production of yoghurt. Milk produced under rigid sanitation practices is labelled 'certified milk'. The 'Long Life' milk usually undergoes an additional process of ultra heat treatment to kill all bacteria present.

Cream is the sweet fatty liquid obtained from the centrifugal separation of milk. "Curd" is a thick substance which separates from milk when mixed with acid and the watery part of the curdled milk is termed "whey" whereas "cheese" is a nutritious milk product that is usually made from curds that have been concentrated and ripened. Its high protein content

makes it an excellent low-cost substitute for meat. "Butter" conotes the fatty product that is made from cream by churning.

"Buttermilk", byproduct of butter making, contains about 8.5% or more of the solid non-fat (SNFs) and a small part of the fat lost during the churning process of butter making.

Skim milk refers to the non-fat milk that contains all the nutrients present in milk except fat. Depending on flavouring strawberry milk drink or chocolate milk drink could be made. "Ice cream", a produce that may contain a variety of ingredients, is essentially a frozen product made from cream and sugar, with or without a natural flavouring and stabiliser.

1.6 Aseptice Production of the Dairy Products

As already stated in section 1.3 of this work, high level hygiene is important in the production of dairy products. These products are low acid foods and so vulnerable to rapid growth of bacteria, yeast and moulds. Hence they should be aseptically handled, pasteurized or sterlized.

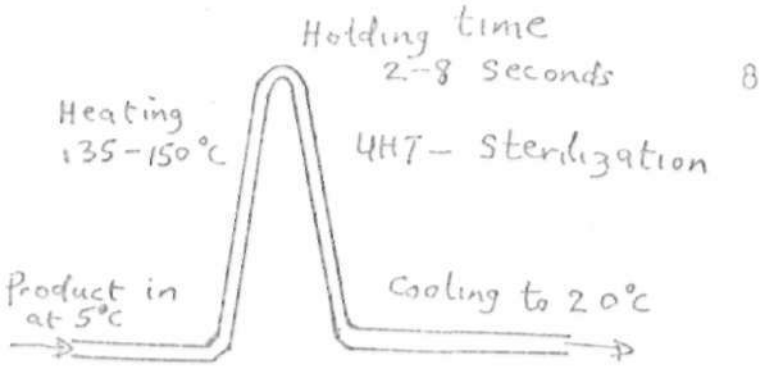
In sterilization processes, the product should enter the sterilizing equipment at a temperature of

4 to 7°C and is subsequently heated to sterilization temperature. In Nigeria, the Food and Drug Administration Division of the Federal Ministry of Health specifies the sterlization temperature to be in the range of between 135°C and 150°C. See fig. 1a. Subsequently the product is rapidly cooled, usually down to room temperature.

Definition of pasteurized milk is related to a time/temperature treatment. To comply with the legal requirements in the country, pasteurized milk has to show a negative phosphatase or a negative peroxidase test. Phosphatase and peroxidase are enzyme systems present in milk, which are inactivated by a time/temperature combination slightly above the one necessary for killing mycobacterium tuberculosis. Mycobacterium tuberculosis is the most heat resistant vegetative disease-causing bacterium. These tests are an indication of sufficient pasteurization from the public health point of view.

Furthermore, pasteurized milk should show a positive outcome of the Aschaffenburg test (fig. 1b).

The milk proteins may be seperated into whey proteins and casein. In raw milk, whey proteins are native. Those proteins may be denatured by heat treatment.



UHT = Ultra Heat treatment

Fig. 1.a: Temperature Renge of UHT Sterilization

	Whey Protein Denatured by heating	Aschaffenburg
Pasteurized Milk	Whey Protein Unchanged	+ ... +
Long Life Milk	Whey Protein denatured Partly	+
Sterilized Milk	Whey Protein denatured completely	-

Fig. 1.b: Aschaffenburg Test

The process of denaturation of whey proteins implies a physical change only: the soluble whey proteins are turned into an insoluble state. From nutritional point of view, this process does not result in any losses, on the contrary, certain indications are available indicating a better digestability of the denatured whey proteins making them easier available to infants, elderly persons, and persons suffering from stomach problems.

Pasteurization treatment should be gentle enough not to denature the whey proteins. Consequently, the Aschaffenburg test should be strongly positive, indicating the presence of native whey proteins.

One of the differences between "in-batch sterilization" and "in-flow sterilization"[6] is that all whey protein is denatured after in-batch sterlization", while "in-flow sterilization" results in a partial denaturation only, i.e. a slightly positive Aschffenburg test.

Ultra teat treatment (UHT) milk has been subjected to in-flow sterilization and has a shelf life of at least four weeks at room temperature. It should show a positive outcome of the Aschaffenburg test. "In-batch sterilized" milk is defined as a

product which is free from living bacteria and which has a negative outcome of the Aschaffenburg test.

All "in-flow sterilization" equipment must have a pre-heating zone where the temperature of the product is raised to about 75 to 80°c. (Fig. 2).

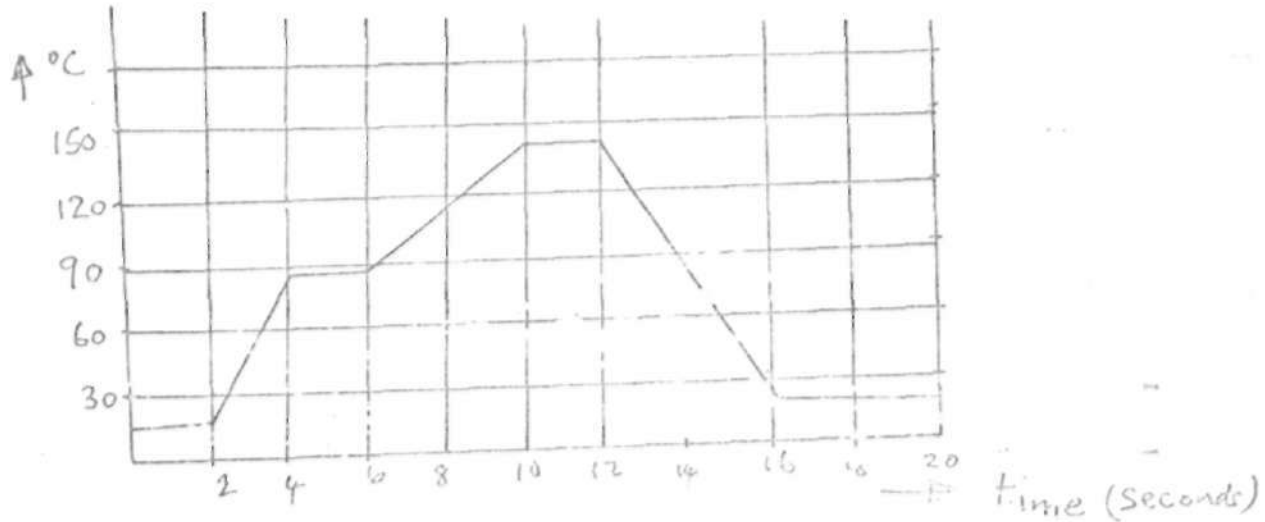


Fig 2: Indirect Working UHT Plant: Temperature/Time Diagram

In plate heat exchangers the product is homogenized at this stage, and then further heated to sterization temperature. It passes a holding cell and is cooled down to filling temperature.

In Nigeria today, the Food and Drug Administration a department of the Federal Ministry of Agriculture makes a legal distinction between three milk products: pasteurized milk, long life milk and sterilized milk.

Pasteurized milk is defined as milk which has been subjected to one of the following heat treatment processes all of which gives a negative peroxidase and/or phosphatase and a positive Aschaffenburg test.

1. Long time pasteurization: 63°c, 20-30 minutes
2. Short-time pasteurization: 72°c, 15 minutes
3. Flash pasteurization: 85° - 95°c, a few seconds to a few minutes (_ 2 minutes)

Sterilized milk is defined as a product which has been heat treated in a hermetically closed container, which is free from living bacteria, and which has been subjected to a heat treatment sufficient to render the product negative as far as Aschaffenburg test is concerned.

Long life milk is defined as a product which has to be processed in the ultra heat treatment (UHT) equipment. The heat treatment is to be in the range between 135 and 150°c.

The product is to be packaged in a pre-sterlized containers which should be light-proof. The packaged product should have a bacteriological shelf-life of at least six weeks at room temperature.

CHAPTER TWO

PLANT DESIGN

2.1 Design Requirement

The design requirements for an aseptic process based on conventional milk products line can be summarized as follows:

1. Tanks, pumps and valves of aseptic design
2. Cleaning-in-place (CIP) systems permanently connected to the plant.
3. Automation of production as well as cleaning/ sterilizing.

2.2 Aseptic Design

All tanks should be of aseptic design with double-sealed agitator cleaning-in-place (CIP) nozzle and hermetically sealed cover. Throughout the process, the tanks should be slightly pressurized with sterile air supplied by the compressed air system provided with filters for the removal of bacteria and bacteriophages. As an alternative to the sterile air (pressurized system), the banks can be fitted with sterile filters (non pressurized system) for respiration to the atmosphere. The bulk starter tanks are equipped for the aseptic transfer of starter to the incubation tanks. The equipment is designed to withstand 150^{o}c and all shafts are fitted with double seats. To avoid

infection through the valve stems, the stems are sealed by a membrane of reinforced rubber.

2.3 Cleaning-in-Place

To safeguard further the high quality of the product, it is important that the entire plant can be cleaned and sterilized from time to time. A closed circuit cleaning-in-place system makes it possible to achieve consistently good results. The cleaning cycle is controlled by a programming unit incorporated in the control panel. A separate sterilization phase, controlled by a time-temperature.programmer, guarantees effective sterilization.

2.4 Design of the Cleaning-in-Place (C.I.P.) System

The cleaning in place system ensures that all pipes utensils and heat exchangers are always aseptically clean This is achieved by circulating a detergent through the units and finally rinsed with water. The most suitable detergents being sodium hydroxide (NAOH) for pipes and utensils and Nitric acid for the heat exchangers. It is economical to recycle the detergents many times before thrown off. Hence it is necessary to rinse the system first with hot water be washing with the detergents and finally rinsing with water. This is best achieved by incorporating with the design control valves and pump as shown in fig. 3. The tanks and

pipes should be made of stainless steel. Details of why stainless steel is preferred will be discussed in the section discussing pipe line problems.

Fig. 3: Cleaning in Place System

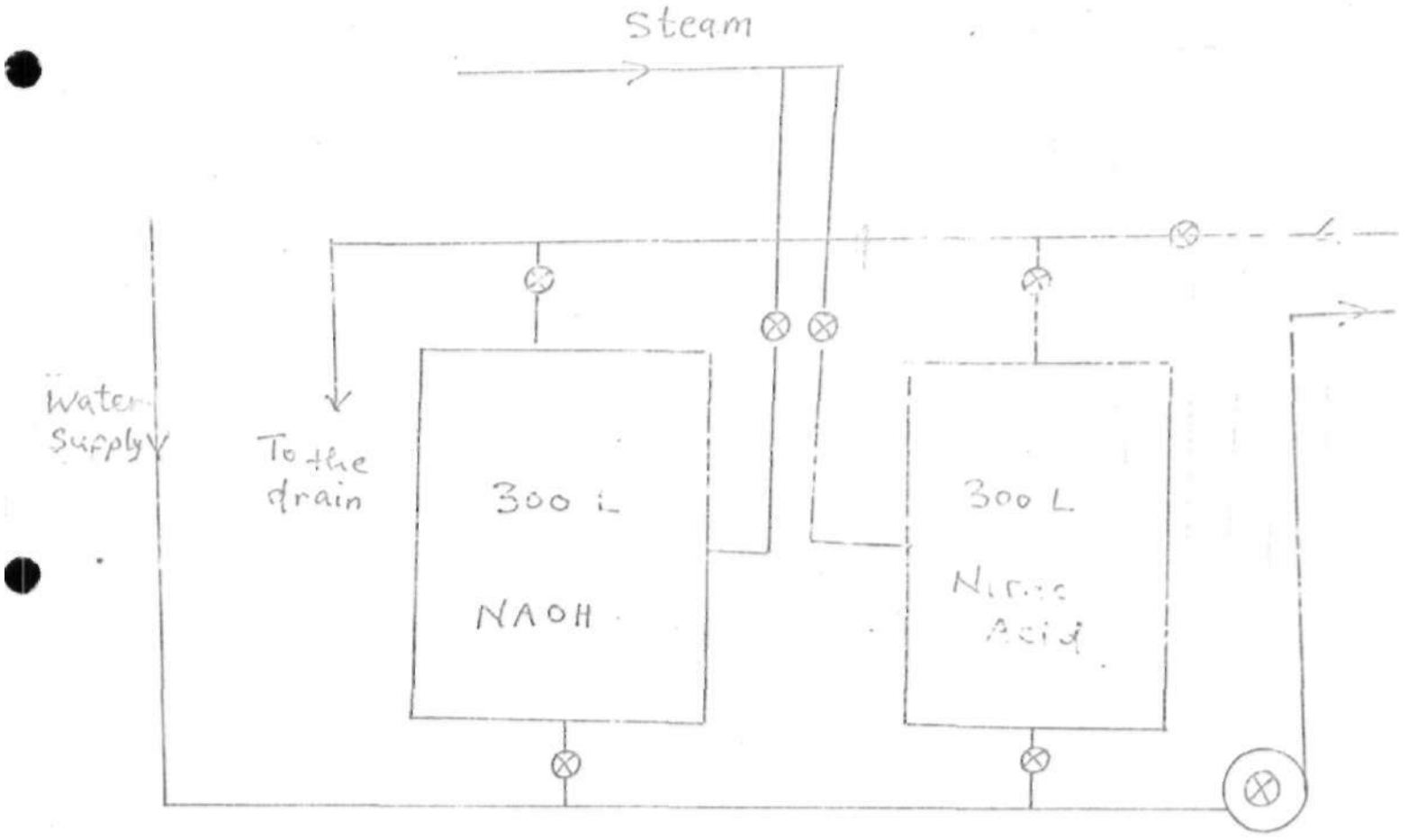


Fig 4: CIP for the Tanks

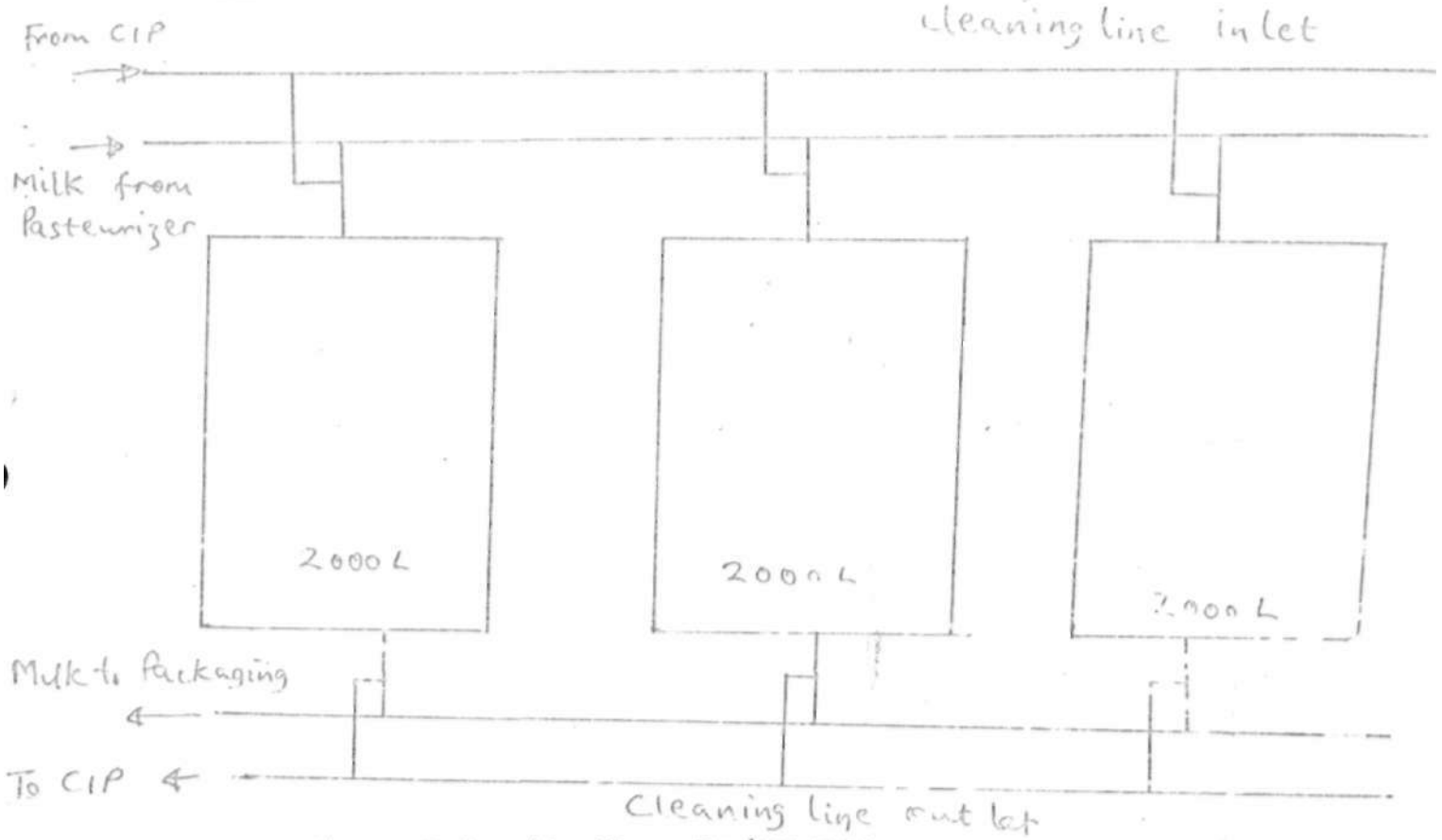
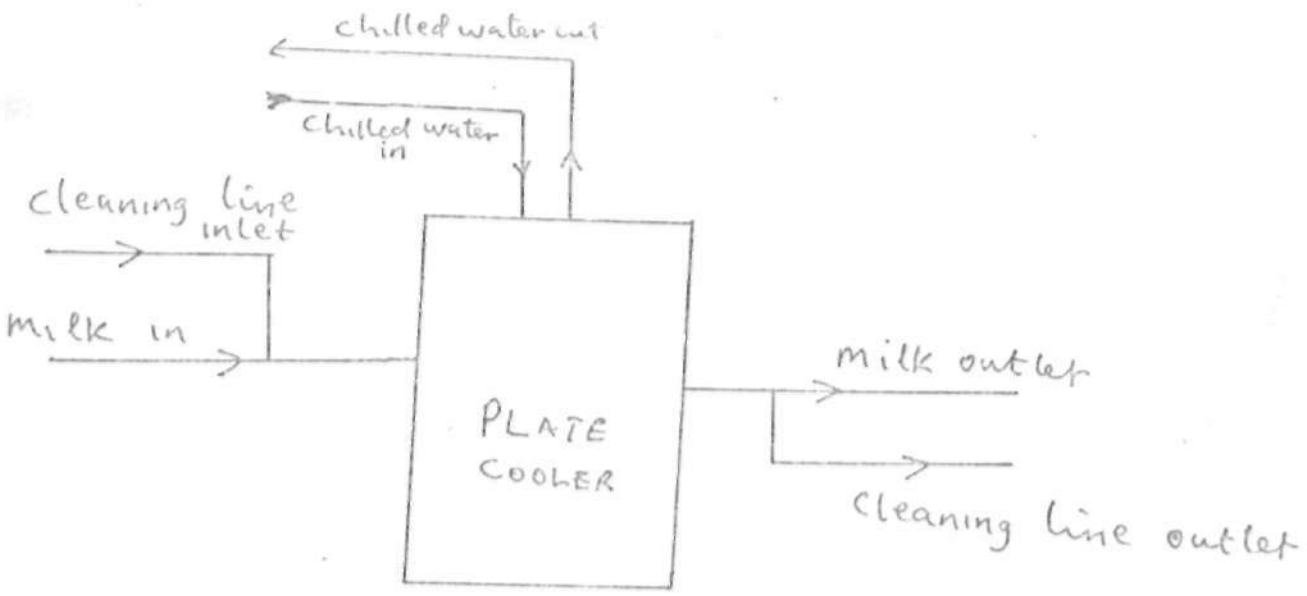


Fig 5: CIP for the Heat Exchangers



2.5 Automation

The aseptic plant requires a certain degree of automation in order to ensure aseptic conditions and effective cleaning. The whole process must be sequence-controlled and operated from a main panel.[7] Yoghurt produced under aseptic conditions still contains a living Yoghurt culture. Undesirable microrganisms have either been killed off or excluded from the product. If the Yoghurt is then packaged aseptically, a shelf life of 4-6 weeks (stored at below $+8^{o}c$) will be assured.

2.6 Plant Layout

The plant layout is as shown in Drawing 1. Space alocation is made up as follows:

Cold store	- 162 m^2
Store	- 18 m^2
Boiler room	- 72 m^2
Paper store	- 36 m^2
Spare parts store	- 18 m^2
Ante	- 72 m^2
Workshop	- 72 m^2
Ice Cream Freezing Score	- 108 m^2
Fuling room	- 36 m^2
Toilets	- 432 m^2
Milk processing room	162 m^2

Ice Cream processing room	- 144 m^2
Powder store	- 108 m^2
Recombining room	- 72 m^2
Reception	-91.8 m^2
Passages	- 108 m^2
Offices	- 72 m^2
Laboratory	- 18 m^2
Main Switch	- 72 m^2
Cooling Equipment and Chilled water tank room	- 72 m^2
Total Area =	1945 m^2

CHAPTER THREE

WATER SUPPLY

3.0 Background

The foregoing discussion on the nature of the Dairy products and the technology of production has shown that water services (hot and cold water system) constitute the most important engineering service in the dairy industry to achieve both the required high level hygiene and the aseptic production itself.[1]

In this work, water supply shall be treated under the following sub-headings:

(a) Source of Water Supply
(b) Water Demand estimations
(c) Well Design
(d) Design of the Water Treatment Plant
(e) Pipe Line Design and Problems
(f) Distribution of Water (Cold)
(g) Distribution of Water (Hot)
(h) Steam Demand

3.1 Source of Water Supply

The plant is to be supplied by both the local water mains, or by its own two bore holes when the local mains fails to supply. The local mains supplies

direct to a second elevated reservoir just above the roof top, while the primary elevated tank is supplied by the two bore holes. Each of the bore holes is 170m deep. A suitable supply arrangement is as shown in fig 6.

Fig. 6: Scheme of Water Supply

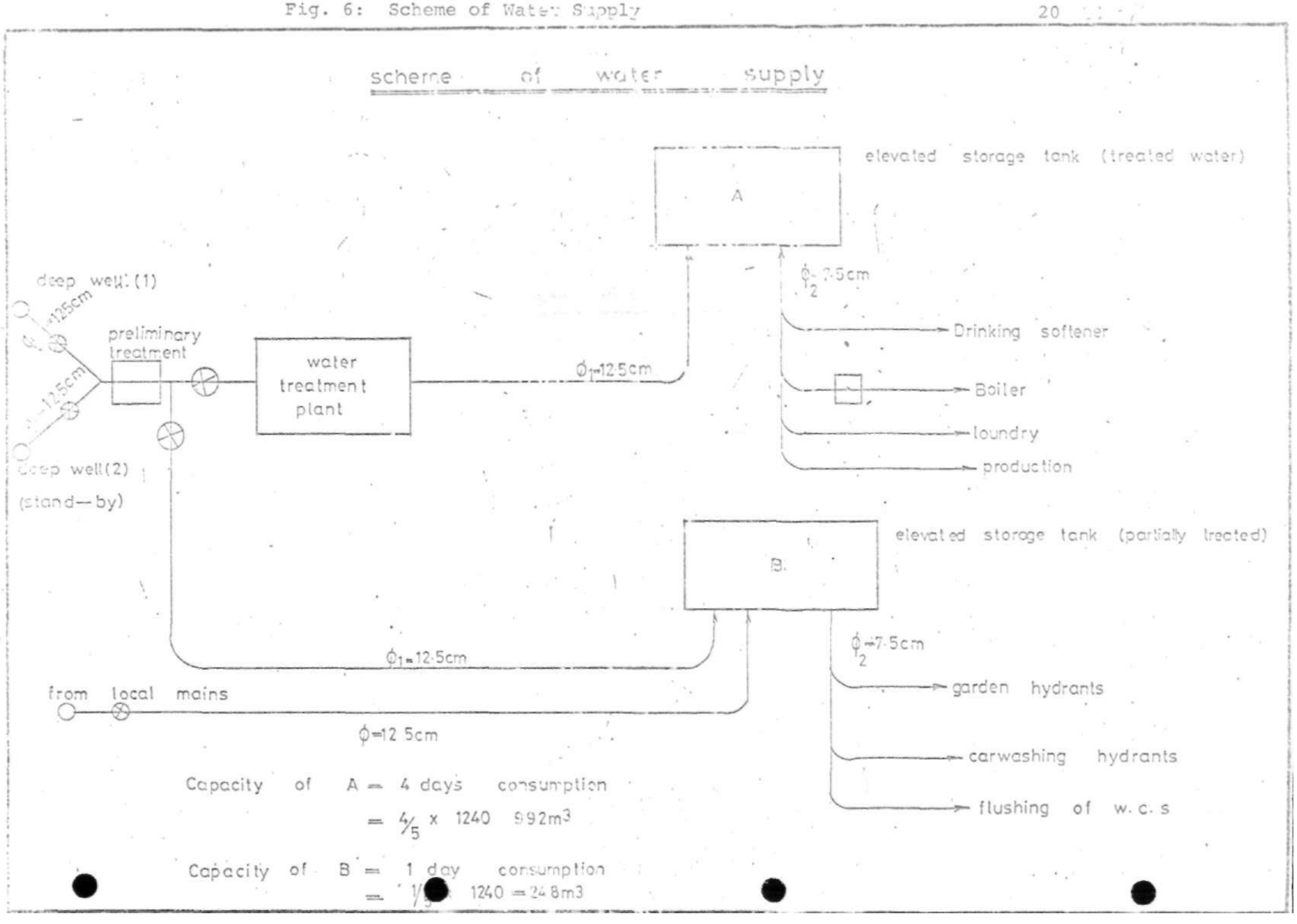
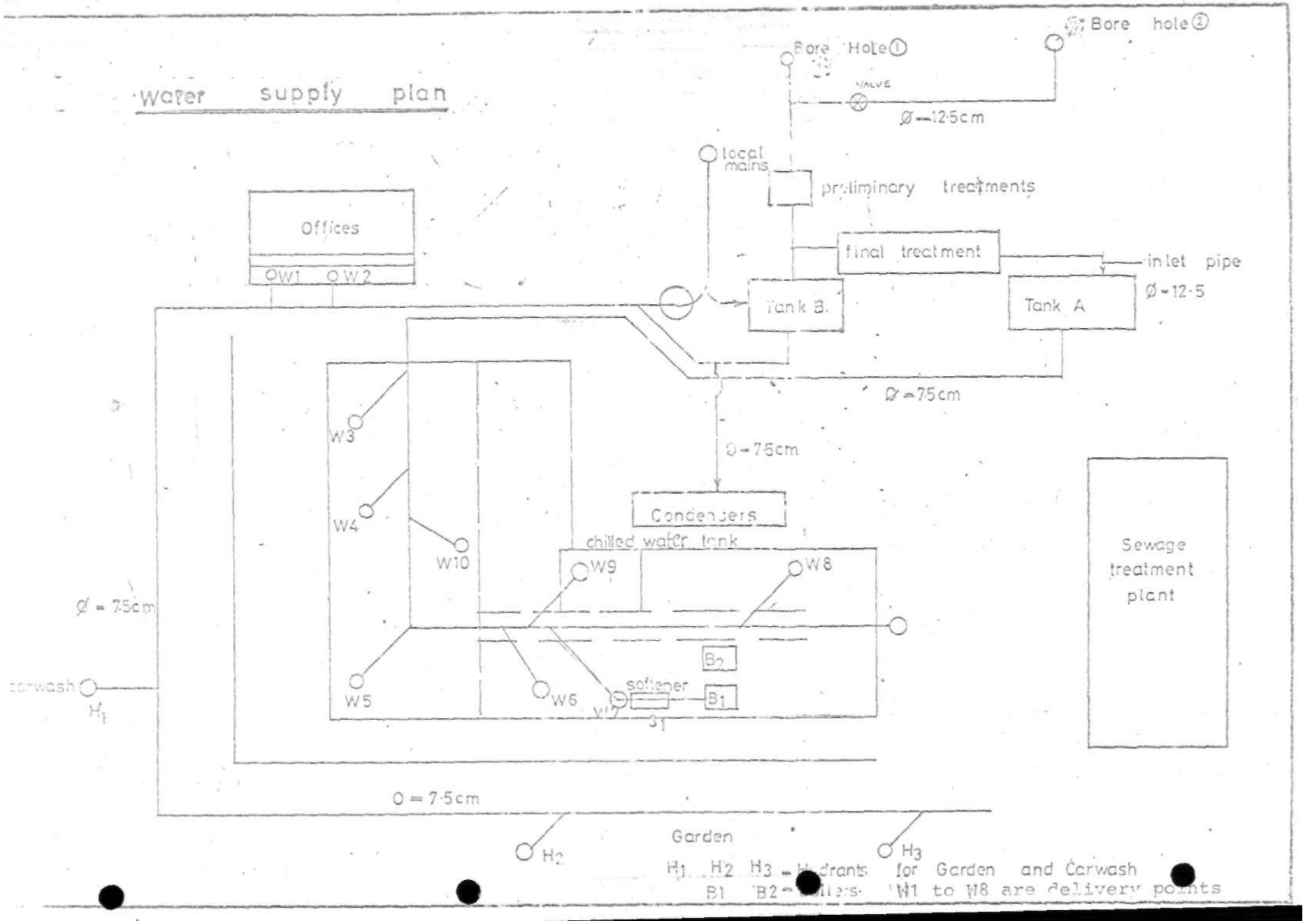


Fig. 11: Water Supply Plan



NOTES ON FIGURE 11

XX -Indicate Boarder of the premises

ø -Diameter of pipes

W_1, W_2, W_3, W_9 -Points of water supply for drinking, production or for toilet use

H_1, H_2, H_3 -Points of water supply for watering the gardens and car washing.


-Indicates non return valve


-two way valve

B_1 and B_2 -Boilers

W_9 -Point of water supply to the water drilling plant. Details of the chilling plant will be described in refrigeration section of this work

FIG. 35: Water Distribution System

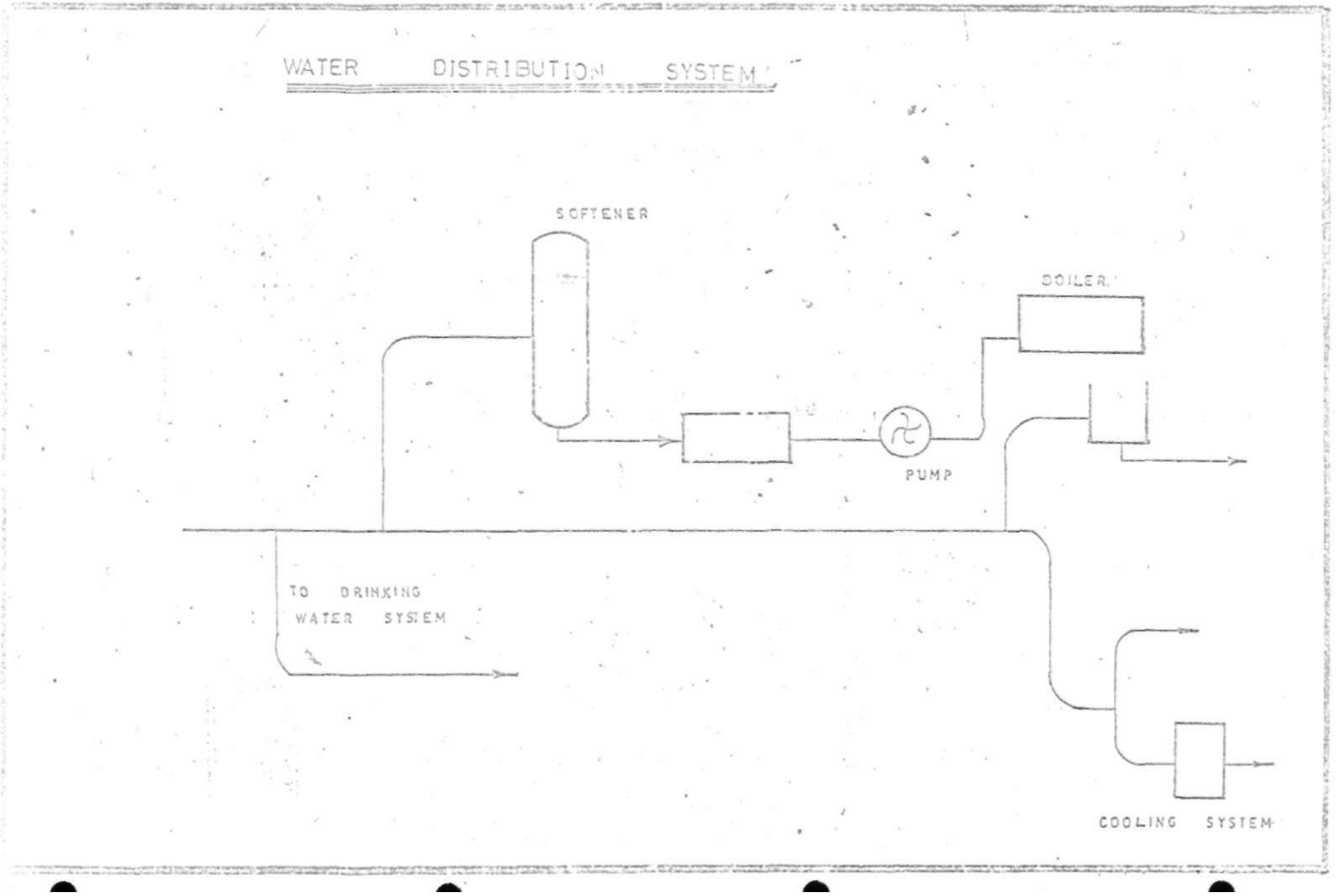
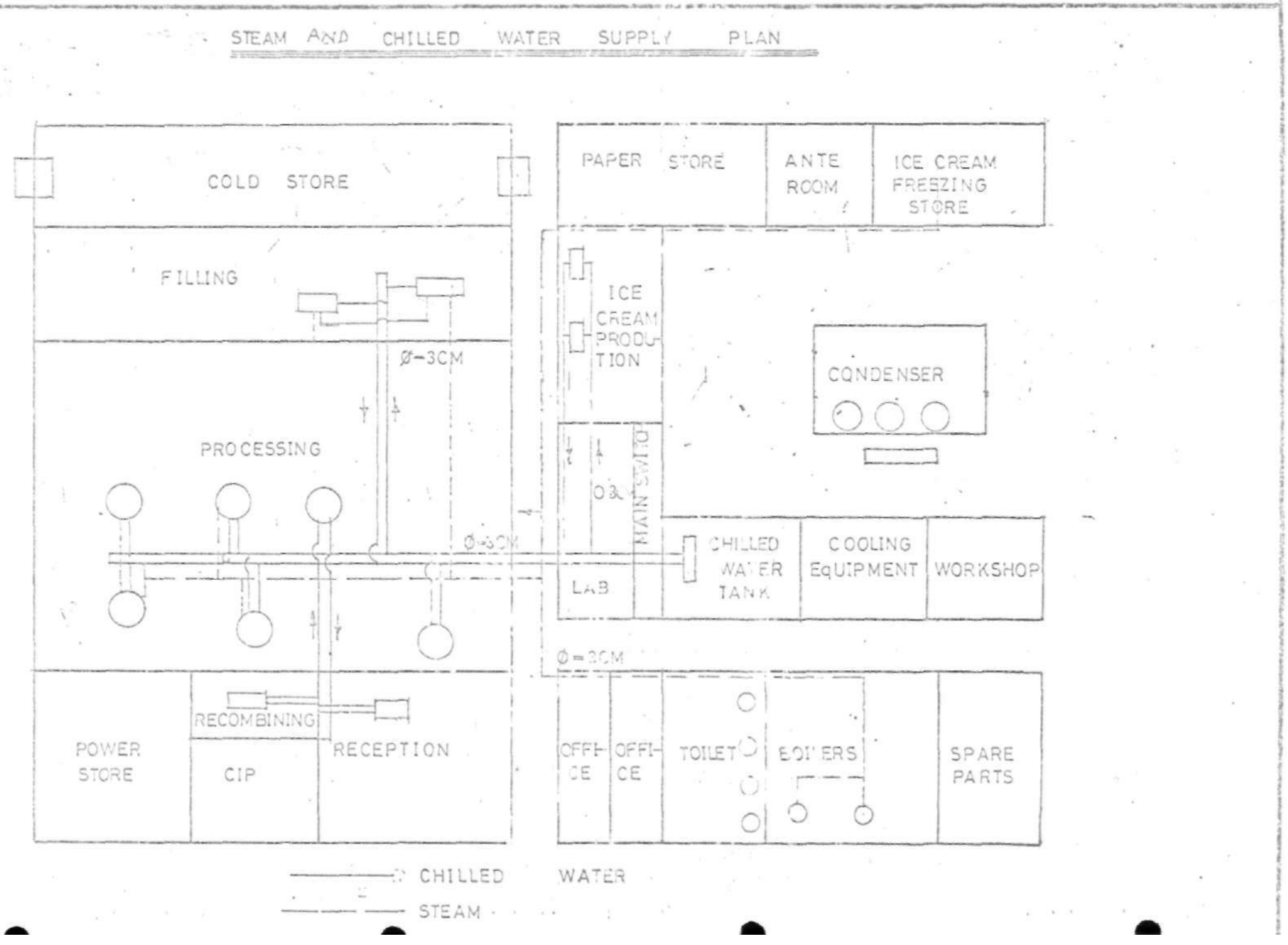


Fig. 13: STEAM AND CHILLED WATER SUPPLY PLAN



Notes on Figure 6:

A: Preliminary treatment includes:

i) Aeration

2. Filteration

Fig. 7 illustrates preliminary treatment.

B: Final treatment includes injection of calculated dosage of chlorine gas. Fig 7 illustrates the chlorine plant.

C: The water going to the boiler has to be softened to avoid scaling. Softening is achieved by injecting calculated dosage of $Na_3 PO_3$ and filtering.

Q = 12.5 cm = Diameter of inlet and outlet pipe of the submersible pump.

The calculation is from:

$$Q = VA = 2.58\ m^3/mm$$

take V = 4m/sec

$$A = \frac{Q}{V} = \frac{2.58}{4 \times 60} = 0.01075 mm^2$$

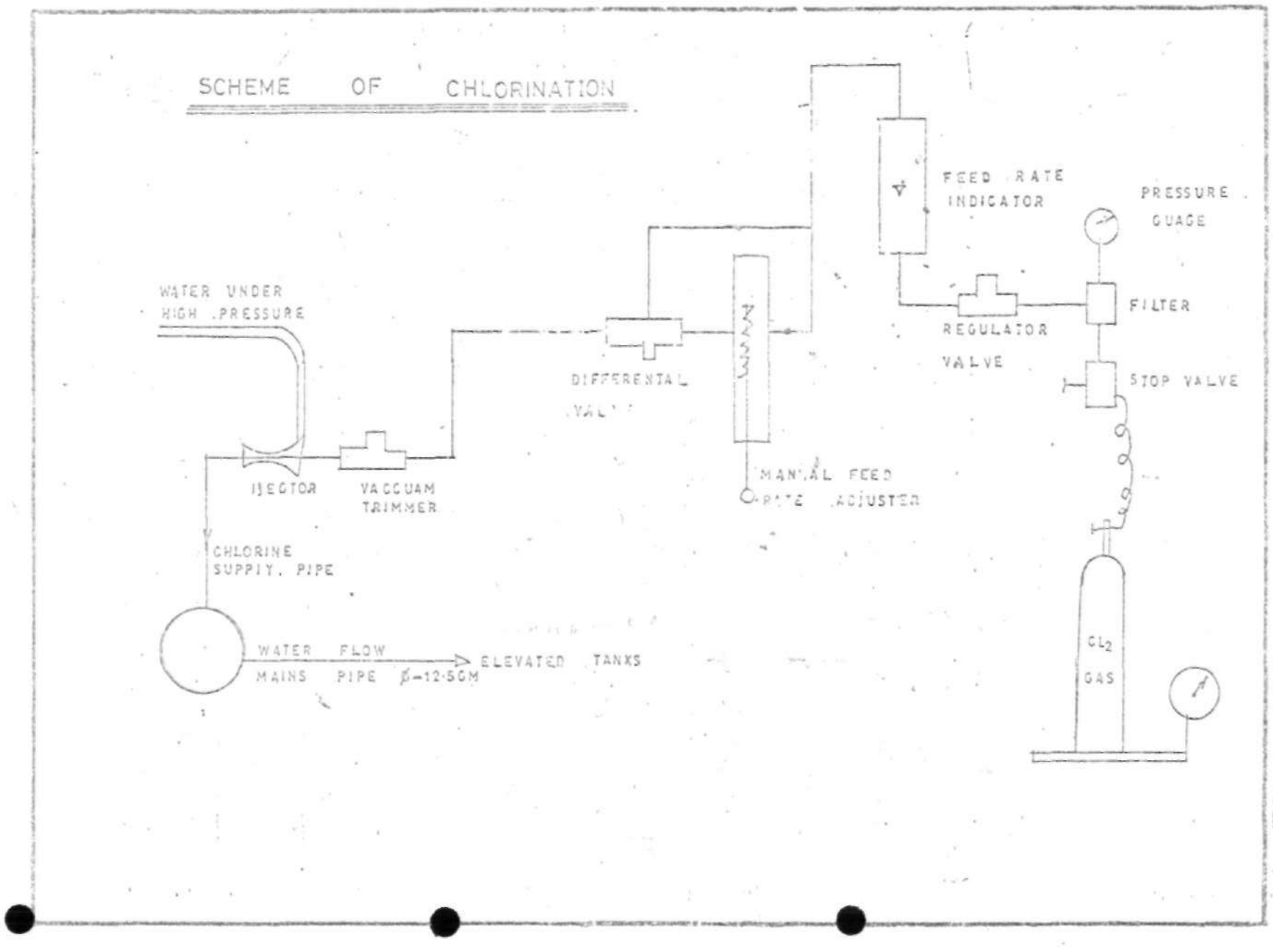
i.e. $\frac{\pi D^2}{4} = 107.5\ cm^2$

or $D = \sqrt{\frac{107.5 \times 4}{\pi}} = 11.7$ cm

The nearest galvanized steel pipe to this is $\emptyset$ = 12.5 cm

$$\therefore \quad V = \frac{2.58 \times 4 \times 60}{\pi \times 10.0125)^2} = 3.5 \text{ m/sec}$$

FIG. 8: SCHEME OF CHLORINATION



3.2 Water Demand Estimations

Specifications of the pumps, water tanks, boilers and pipe network depend on total demand of water. This work takes as basis, a production capacity of 18,000 litres of milk products per day. This level of production requires (see appendix):

(a) 10 (ten) administrative persons
@ 30L/day = 300L;

(b) 55 (fifty five) factory workers
@ 90L/day = 4950L

(c) Laundry capacity of 150kg/day
@ 30L/kg = 4500L

(d) Garden Watering (2 hydrants of $\frac{1}{2}$L/sec)
assuming 4 hours of operation we have
2 x (4 x 3600 sec) x $\frac{1}{2}$ L/sec = 14400L

(e) Car washing
10 (ten) small cars per day
@ 300 L/car = 3000L
3 (three) big vans per day @ 900L/van = 2700L

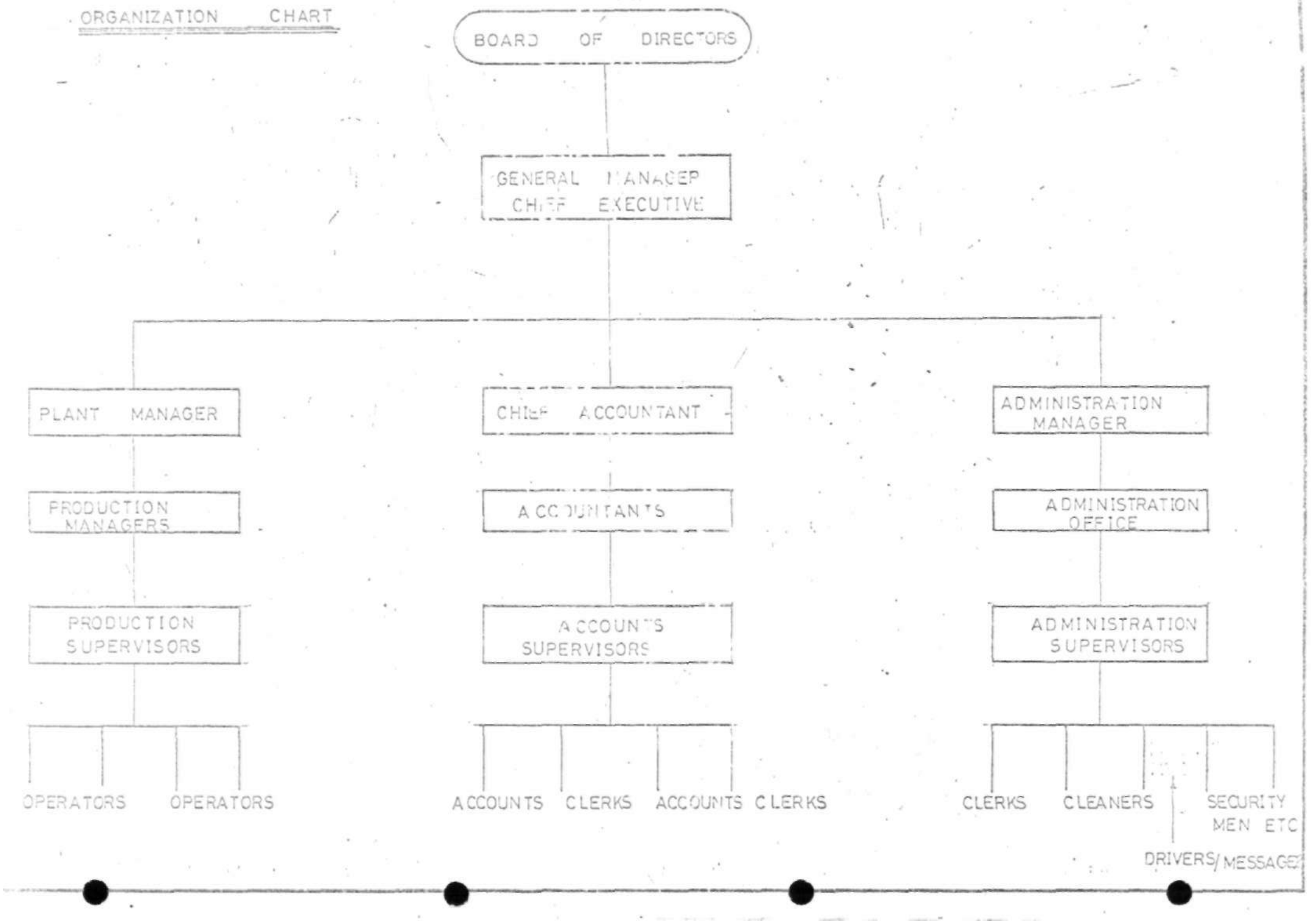
(f) Floor and external part of machines washing
(2566 m^2) @ 32 litres per square metre
32 x 2566 = 82,112L

(g) Production (technological) water (18,000 litres of product) @ 7.5 litre of water/litre of product

Total DEMAND PER DAY = 246962L = 246.96 m^3
135,000L

FIG. 14: ORGANIZATION CHART

ORGANIZATION CHART



Assuming 1 shift = 8 hours

Average output of water per hour

$$= \frac{246962L}{8}$$

$$= 30870.25L$$

$$= 30.87\ m^3$$

Peak demand = $3 \times 30.87 = 92.61\ m^3$/hour

(Peak factor = 3, assumed)

3.2.1 Deep Well Delivery (Pumping Rate, q)

It is desirable that the bore hole pump is capable of delivering in one hour the peak hour water demand of the plant. This means that the total pumping rate of all the bore hole pumps (q) must not be less than $92.61\ m^3$/hour.

i.e. $q \geq 92.61\ m^3/L$

or $q \geq 1.544\ m^3/min.$

3.2.2 Reservoir and Pumping Capacities Estimations

It is desirable that the water reservoir should contain about 5 days water demand. This means that the reservoirs should have a total capacity not less than $1234.8\ m^3$.

A capacity of $1240\ m^3$ is therefore considered good. It is also required that each of the bore hole

pumps is capable of pumping full the reservoir in one day working for a total of eight hours.

$$\therefore \quad \text{Pumping capacity } (Q) = \frac{1240}{8} \ m^3/\text{hour}$$

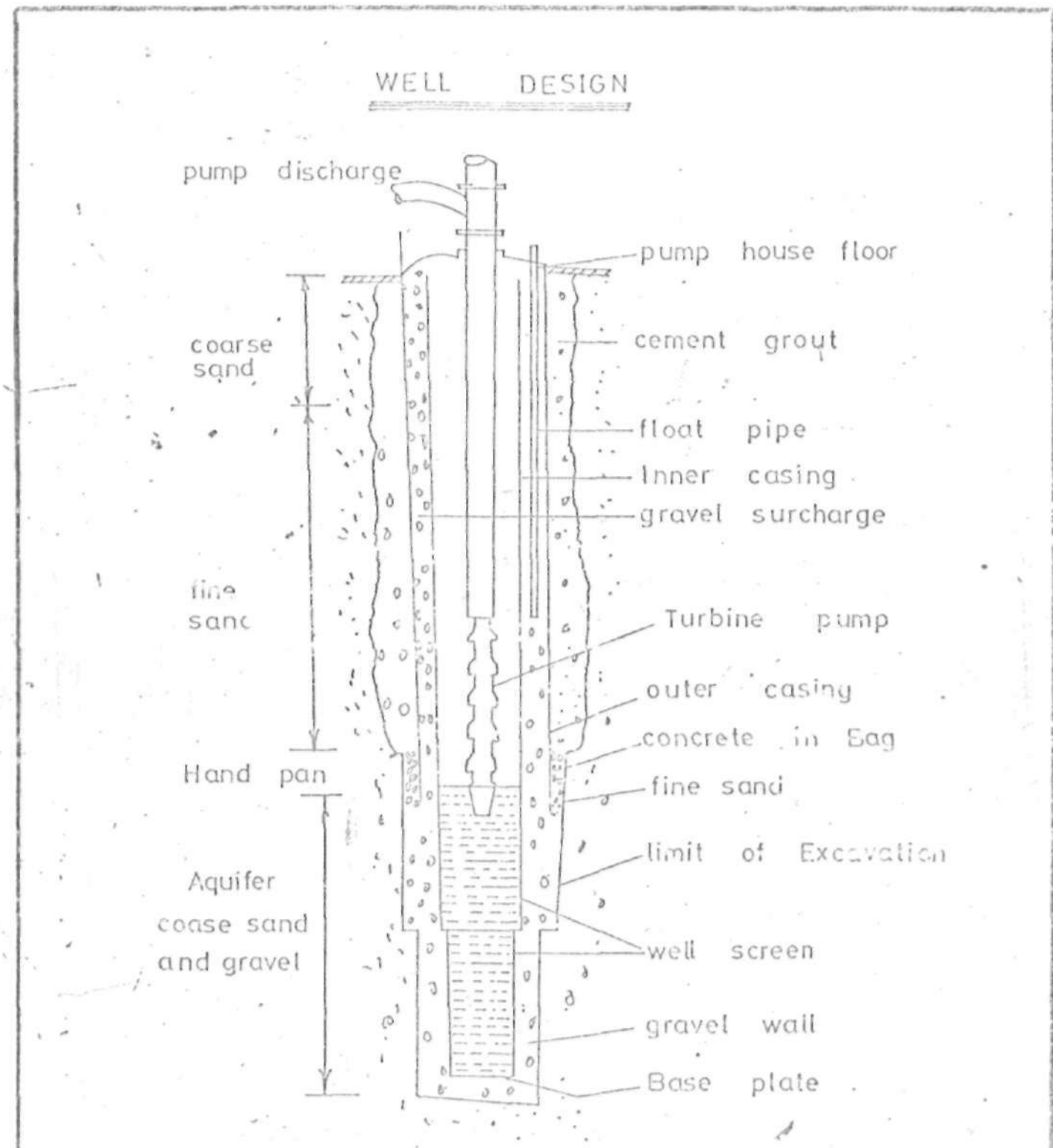
$$= \frac{1240}{8x60} \ m^3/\text{min}$$

$$Q = 2.5833 \ m^3/\text{min}$$

3.2.3 Well Design

A well can be seen as consisting of two parts: (1) the conduit portion of the well, which houses the pumping equipment and provides the passage for the upward flow of water to the pumping intake, and (2) the intake portion, where the water from the acquifer enters the well. In consolidated water bearing materials, the conduit portion is usually cased from the surface to the top of the acquifer, and the intake portion is an uncased open hole. In unconsolidated acquifers, a perforated casing or a screen is required to hold back the water bearing material and to allow water to flow into the well.

The depth of a well depends on the anticipated drawdown for the design yeild, the vertical position of the more permeable strata, and the length of the intake portion of the well.



Gravel packed well with submersible pump

The Well size affects the cost of construction substantially. The diameter of the Well is ogoverned by (a) the proposed yield of the well, (b) entrance velocity and loss and (c) the method of construction. The controling factor is usually the size of the pump that will be required to deliver the design yield. The diameter of the casing should be two norminal sizes larger than the size of the pump bowls, to prevent the pump shaft from binding and to reduce well losses. Table 1 gives the casing sizes recommended for various pumping rates.

Anticipated Well Yield in Litre/Min.	Norminal Size of Pump Bowls in Cm.	Optimum Size of Well Casing in Cm.	Smallest Size of Well Casing in Cm.
Less than 788	10.00	15.00	12.50
788 to 1800	12.50	20.00	15.00
1800 to 2925	15.00	25.00	20.00
2925 to 4000	20.00	30.00	25.00
4000 to 5850	25.00	35.00	30.00
5850 to 8100	30.00	45.00	35.00
8100 to 13500	35.00	55.00	40.00

Table 1: Recommended Well Diameters

The wells are generally lined or cased with mild steel pipe, which should be grouted in place in order to prevent caving and contamination by vertical circulation and to prevent undue deterioration of the well by corrosion.

Screens are used in wells in unconsolidated acquifers. The width of the screen openings (slot size) depends on the critical particle size of the water-bearing material to be retained and on the grain-size distribution, and is chosen from the sieve analysis of the acquifer material. With a relatively coarse and graded material, slot sizes are selected that will permit the fire and medium-sized particles to wash into the well during development and to retain a specified portion of the acquifer material around the screen. A graded filter is thereby generated around the well, which has higher permeability than the undisturbed acquifer material.

Perforated casings are generally used in uncemented wells when relatively large openings are permissible. If the casing is slotted in place after installation, the smallest practical opening is 3.0mm. Machine perforated casings are also available. Fabricated well screens are available in a wide variety of sizes, designs and materials. The choice of material is governed by water quality and cost.

3.3 Water Treatment

The water from the bore holes is treated as follows:

3.3.1 Aeration:

Air under high pressure from a compressor is injected into the intake pipe line so that the oxygen of the air removes any objectionable odour and forms globles with any available mineral in solution so that it can be filtered out.

3.3.2 Filteration:

The water is then passed through the high pressure filter where most of the globles and any solids are removed from the water.

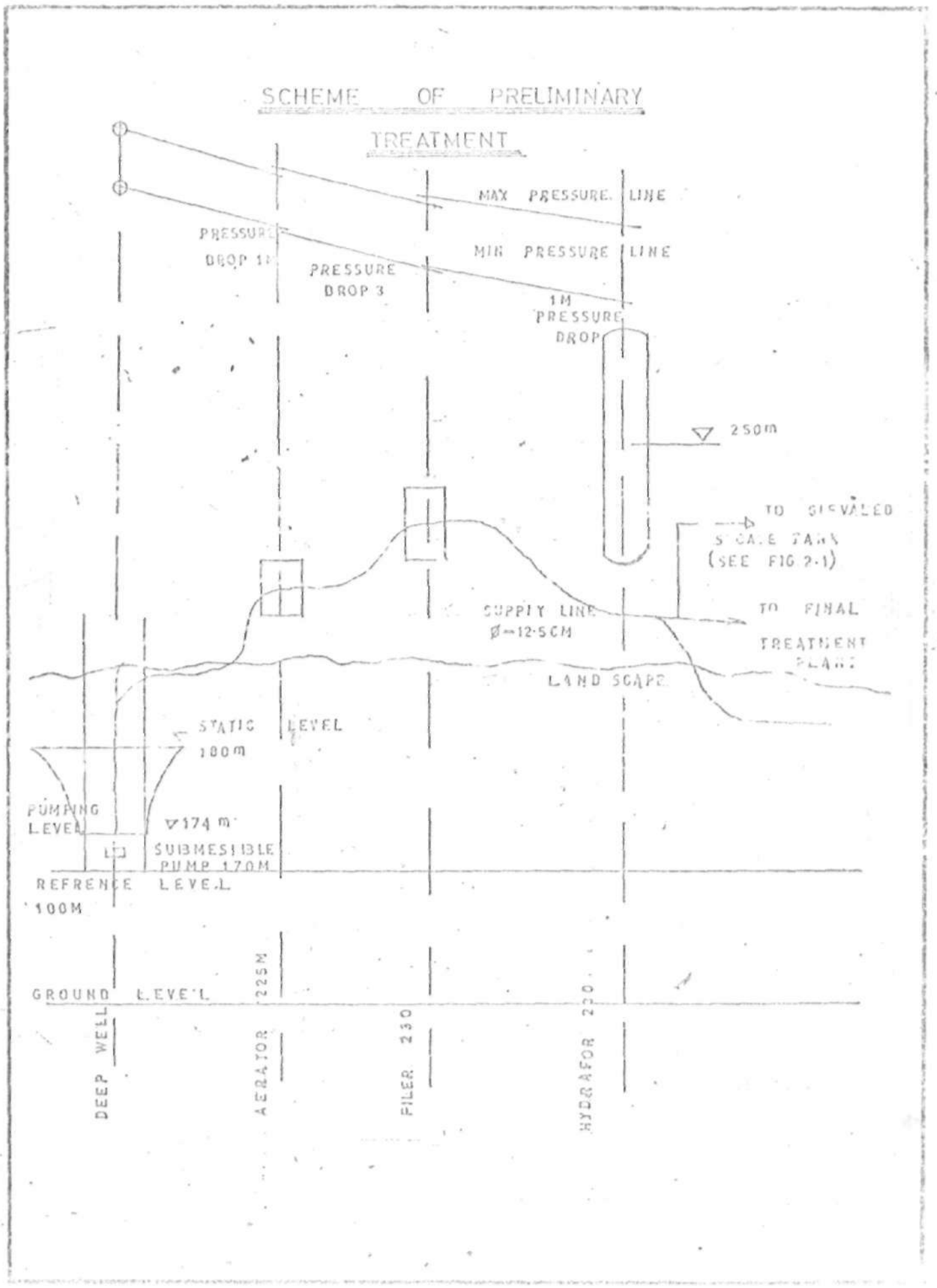
3.3.3 Disinfection:

The water is disinfected by chlorination. Details of the chlorination scheme is shown in figure 8. The feed rate is adjusted so that the dosage is in the order of the part chlorine to one million part water.

3.3.4 Softening:

Only the quantity of water required for the boilers is softened by addition of 4 to 5 mg of tri-sodium phosphate to each litre of water. Such that if any calcium carbonate (which causes hardness of water) is present it will react with the tri-sodium phosphate to form sodium carbonate and calcium phosphate, both of which are precipitated and filtered off, while the softened water feeds the boilers.

FIG. 7: SCHEME OF PRELIMINARY TREATMENT



The reaction can be represented as follows:

$$NO_3\ PO_3 + 3Ca\ CO_3 \qquad 3NaCO_3 + Ca_3PO_3$$

3.4 Pipe Line Design and Problems

The design of the pipe line system is closely associated with the production processes especially the sterilization and cleaning processes.

3.4.1 Types of Sterilization Process

UHT sterilizers are equipment capable of heating the product in continuous flow to a temperature between 135°c and 150°c. In principle there are two different types of this equipment (fig 15), the direct and the indirect sterlizers.

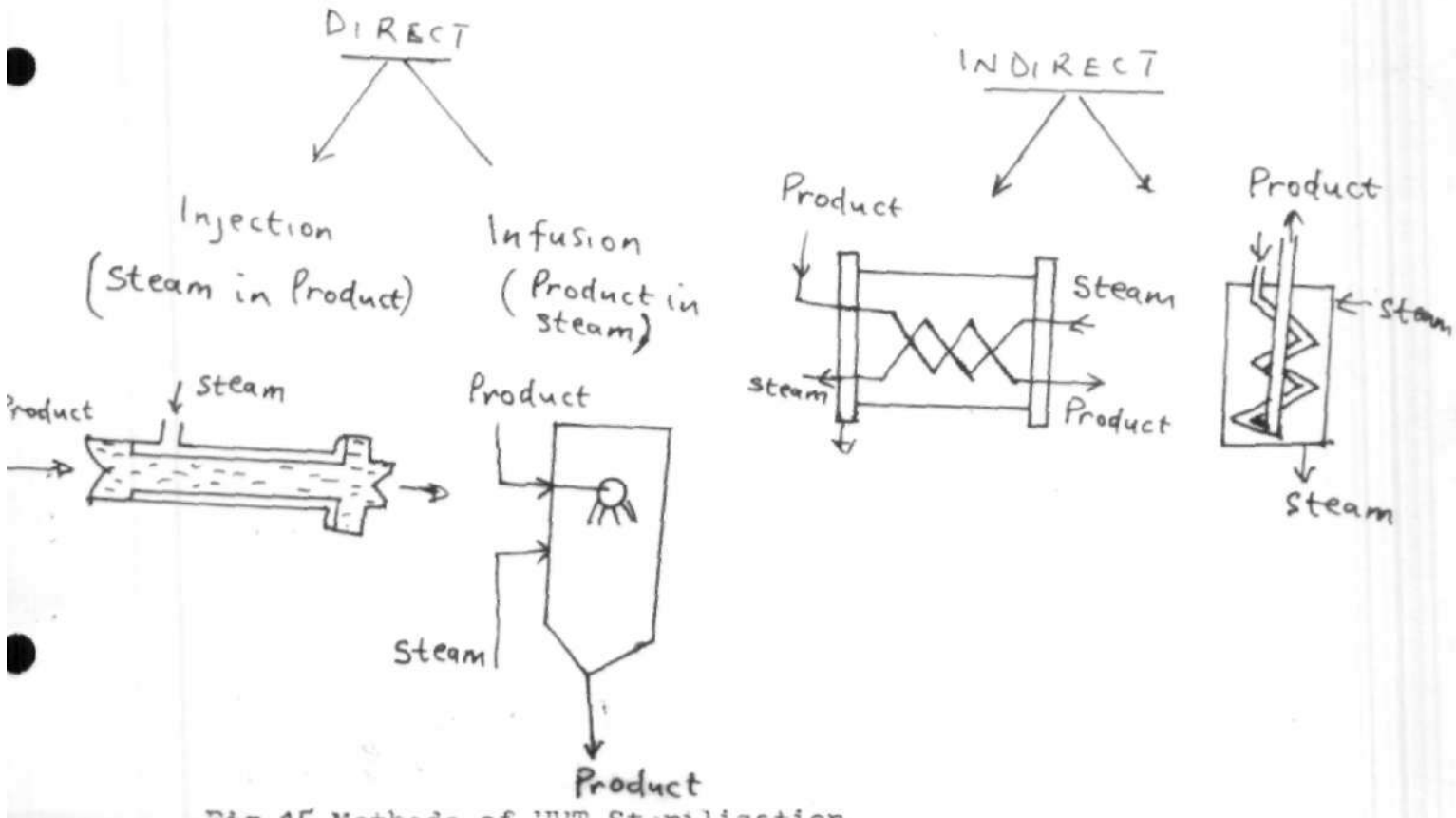


Fig 15 Methods of UHT Sterilization

3.4.2 Direct Sterilization

Direct heating features a direct contact between the heating medium, i.e. steam, and the product to be heated, while in indirect heating, a heat exchange surface is sperating the heating medium i.e. steam or superheated water from the product.

Direct heating methods are also sub-divided into two sub-groups fig (16).

1. Injection heating
2. Infusion heating

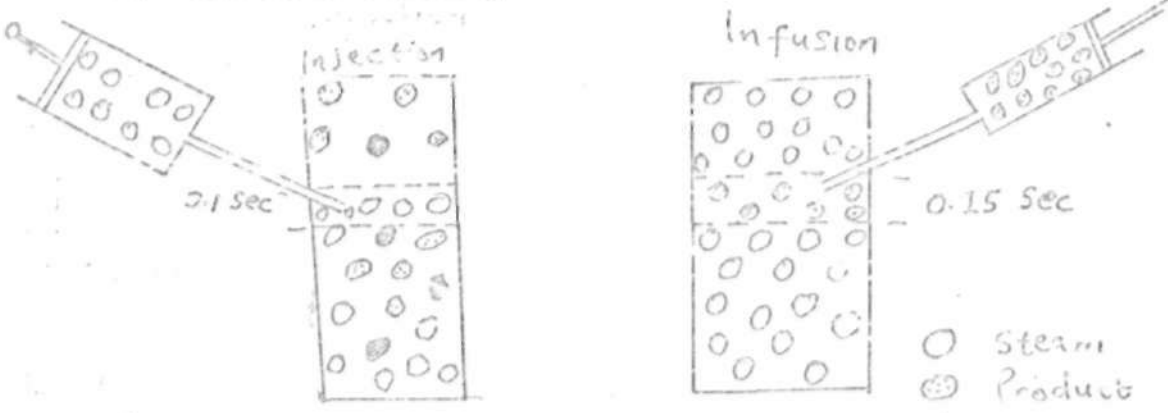


Fig 16: Methods of Direct Heating

Since there is direct contact between product and super heated steam, special demands have to be made (and are legally enforced in this country) on the quality of the steam. The Food and Drugs Administration Division of the Federal Ministry of Health insists that the steam has to be produced from potable water sufficiently treated to ensure that it is of minimum certified quality free from objectionable bodies

(whether in solution or in suspension) or tests and odour. Other required qualities for production convenience are softness and acidity level of PH value ranging from 3.5 to 4.5. The piping between the steam generator i.e. boiler and the heating equipment must be of stainless steel, and a cyclone and an active carbon filter must be included to purify the steam (fig 17). The pipes are lagged up to a sufficient extent depending on the size of the pipes. Only special cleaning agents are permitted to be used for cleaning the boiler (see clean in place design).

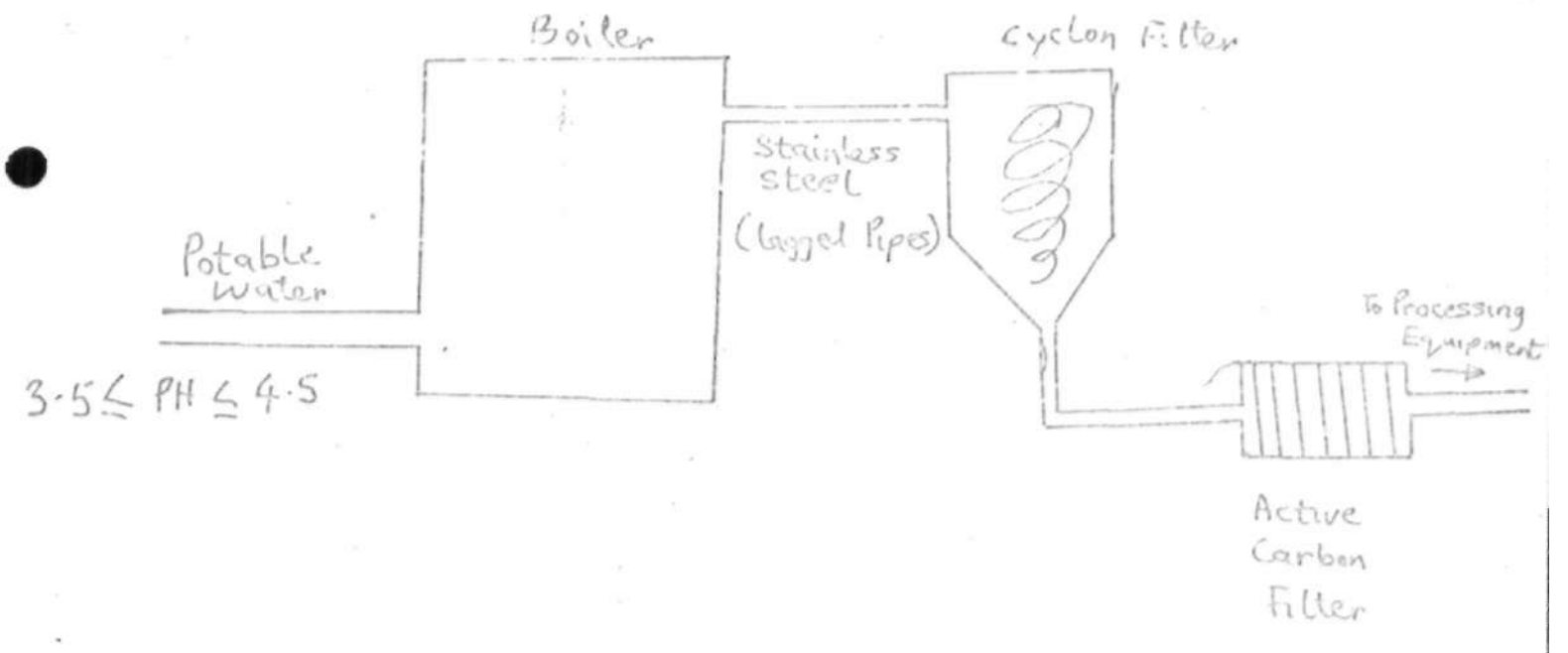


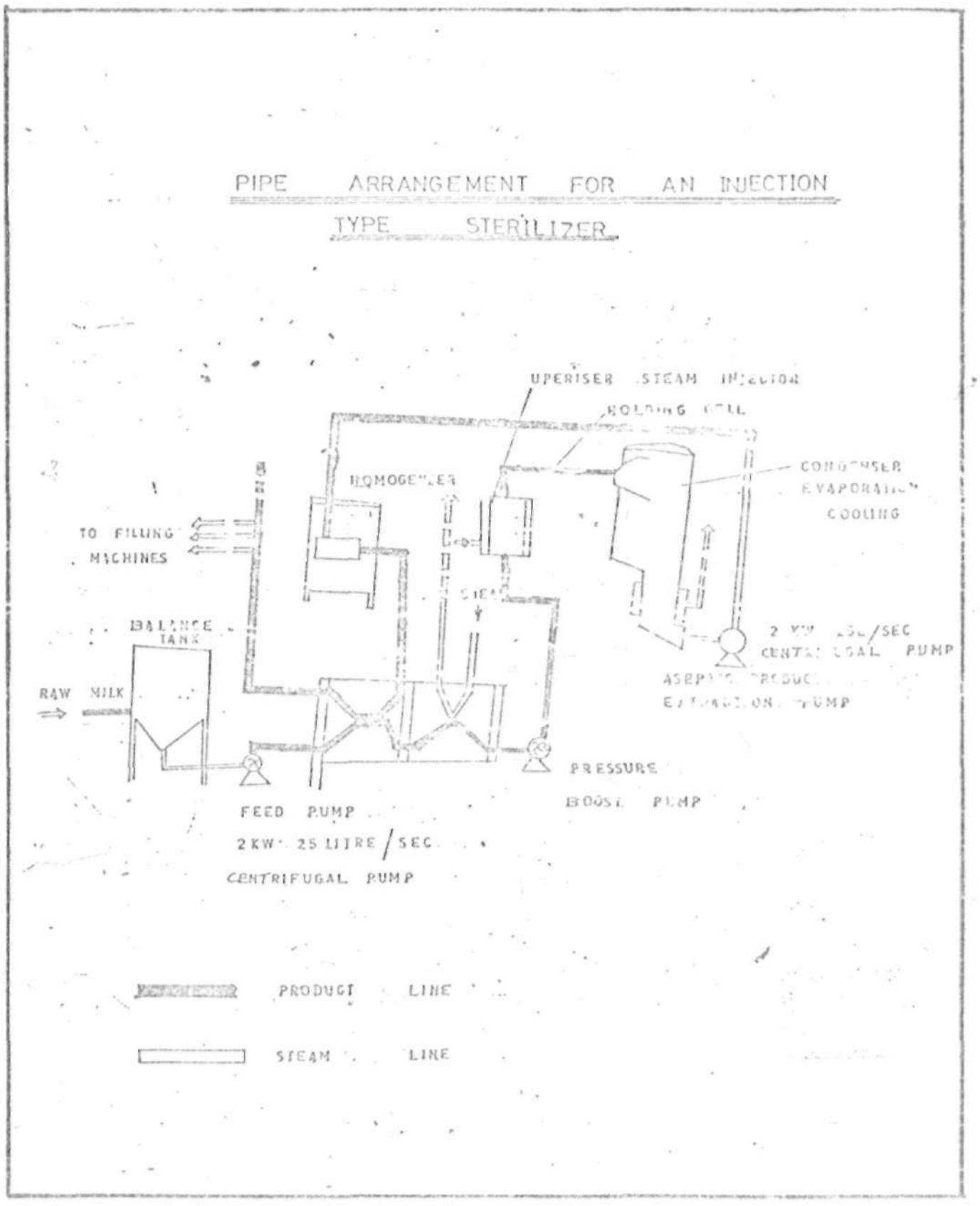
Fig. 17: Method of Steam Supply

In principle, a time/temperature diagram of the direct heating process shows a preheating of the milk entering the equipment at a temperature of 4-7°c and heated by conventional plate or tubular heat exchangers (indirect heating) to a temperature of 75-80°c. At this stage steam is injected into the product or the product is infused into a steam chamber raising the temperature very rapidly to 140-150°c. Heat transfer is mainly due to condensation of steam. In terms of dilution by water, approximately 10% of product dilution takes place. This water has to be removed in order to give a product having the same dry matter content as the incoming product. Thus evaporation cooling becomes a necessity.

After final heating and after the holding cell, the product is exposed to evaporation cooling. In general terms it enters a vacuum chamber. In this chamber the condesed water is removed. Adjustment of the vacuum which is product-dependent ensures the neccessary removal of water and simultaneously controls the dry matter content of the final product.

As far as milk and milk-based products are concerned homogenization should be down-stream, i.e. on the sterile side which means that an aseptic homogenizer is necessary in the system. Fig 18 shows pipe arrangement for an injection type sterilizer.

FIG. 18: PIPE ARRANGEMENT FOR AN INJECTION TYPE STERILIZER

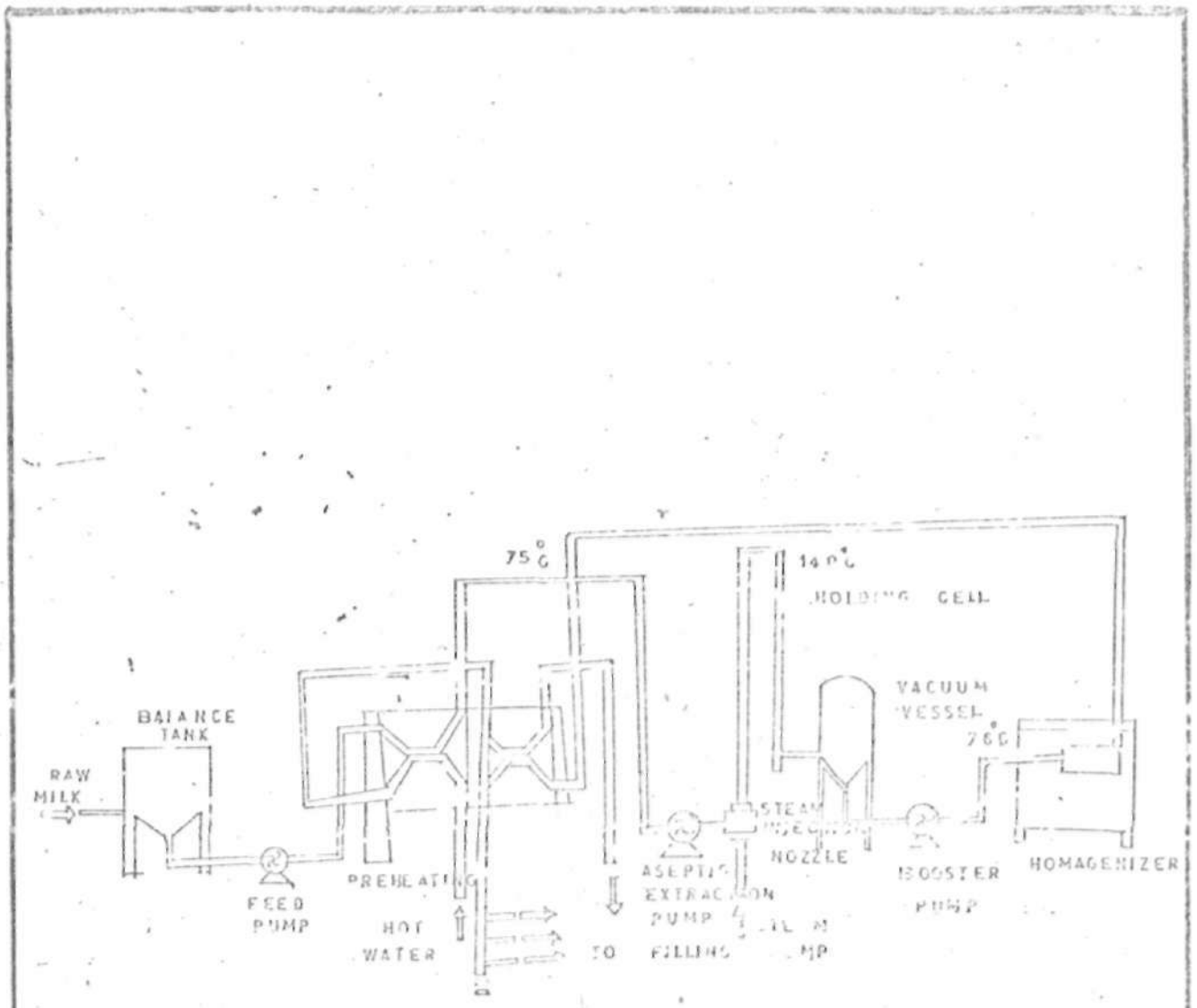


The problem with the system shown in fig 3.4 is that steam is used both in the preheating circuit and for actual sterization process such that the dryness factor of the sterilizing steam varies with the temperature of the entering raw milk. Hence the dry matter content of the product after evaporation in the vacuum chamber cannot be the same as that of the raw milk entering the balance tank. To improve on the result, it is my submission that preheating section be served with hot water system while steam serves only the sterilization section. My proposed design is as shown in fig 19.

The advantages of direct heating are:

1. Very rapid heating and cooling and consequently a low total temperature on the product.
2. Less scaling, i.e. deposit formation, in the equipment (as compared to indirect heating) and less burnt flavour changes in the product (only applicable when indirect heating equipment is run for excessive periods of time.
3. Low oxygen content of the final product. Expansion cooling does not only remove water but also dissolved gases (O_2) resulting in an oxygen content of the final product of 0.1-0.5 ppm.

FIG. 19: PROPOSED MODIFICATION IN DESIGN OF FIG 18



PROPOSED MODIFICATION IN DESIGN OF FIG 18.

4. Less scaling also permits longer production runs, and, consequently, such equipment may require less frequent cleaning and pre-sterilization cycles.
5. Direct UHT sterilizers are comparatively insensitive to pressure drops and, consequently, relatively visions products like Yoghurt can be processed.

The disadvantages of this type of equipment are the higher investment costs, the higher level of technical complexity, and connected to that is more maintenance. A further disadvantage is that in expansion cooling some flavour compounds may be removed together with the steam from some products.

3.4.3 Indirect Sterilization

In the indirect type of equipment, the sterlization temperature is somewhat lower, ranging between 135 and 140°c, occassionally up to 145°c. This is due to the fact that some sterilizing action is already achieved during the heating up of the product.

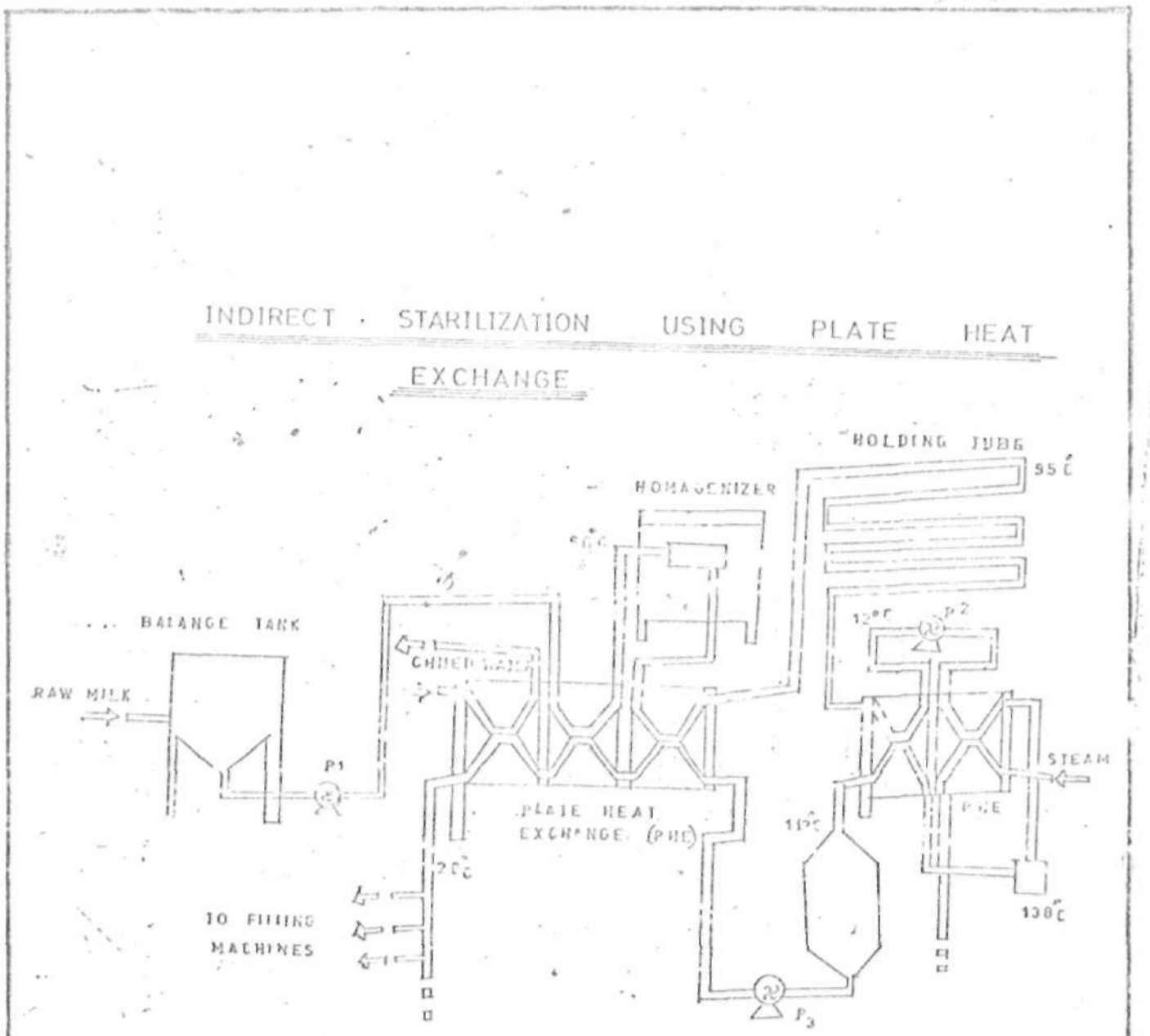
Three different types of indirect UHT - processing equipment can be distinguished: the plate heat exchanger the tubular heat exchanger, and the scraped, or swept-surface sterilizer. Scraped surface sterilizers feature rotating knives placed in tubular structures

constantly removing deposited material from such surfaces. Such equipment is expensive in investment cost, demands a relative high level of maintenance, and has relative low regenerative energy recovery: it is expensive in operating costs. The main advantage of scraped surface sterilizers is their ability to tolerate high pressure drops, i.e. processing of very high viscous products and to handle products containing sizeable particles. Such products however, cannot be packed aseptically in Tetra pak aseptic packaging system. In Nigeria today only plate and tubular indirect UHT sterilizers operate with Tetra Pak aseptice packaging systems.[12]

Fig 20 shows Indirect Sterlization using plate heat exchanger. In this system, the milk enters the sterilizer and is preheated to about 60°c, homogenized, heated to 95°c passed through a holding cell, heated to sterilization temperature (138°c) and sent to the flow diversion valve. It is returned to the first cooling section and then admitted to a deaerator. Next the product is extracted by a pump. Cooled and discharged for packaging.

For a plant that uses the tubular heat exchanger the pipe arrangement is as shown in fig 21.

FIG. 20: INDIRECT STERILIZATION USING PLATE HEAT EXCHANGE



pipe arrangement in a plate heat Exchange (PHE) starilization system

One of the disadvantages of this type of sterilizer is that it is liable to get heavier burning-on and heavier deposits. Another disadvantage of the tubular heat exchangers is that they are difficult to clean. In order to get a satisfactory cleaning effect, turbulance has to be created on the surface to be cleaned. It is much easier to create turbulance on plate heat exchanger surfaces than it is on tubular heat exchanger surfaces.

Normal cleaning procedures for plate heat exchangers are as follows: water rinse - alkaline cleaning-water rinse-acid cleaning-water rinse. This cleaning procedure is quite efficient for plate heat exchangers. It takes approximately an hour and a half. The procedure for tubular heat exchangers is: water-acid-water-alkaline-water-acid-water.

In the case of tubular heat exchangers, it is recommended to add a further cleaning step to the normal cleaning procedure: The cleaning time increases to approximately two hours. Of course, this procedure is more expansive as more water, time and energy is needed.

Advantages of Indirect heating equipment are:

1. Low investment cost
2. Good regenerative energy recovery - up to 85-90%.

3. Technologically relatively simple solution - relatively low maintenance cost.
4. No special demands on the quality of the heating medium - either steam or superheated water.

Comparing plate and tubular heat exchangers, the following advantages can be seen in the respective system.

A: Plate Heat Exchangers

1. Less expensive in investment cost
2. Easier to clean
3. Equipment can be opened for inspection
4. Somewhat better regenerative energy recovery.

B: Tubular Heat Exchangers:

1. Fewer gaskets - microbiologically safer, less maintenance.
2. Tolerates higher pressure drops, longer production runs, viscous products can be processed.

Hence this project recommends the plate heat exchanger.

3.4.4 Pipeline Design Specifications

The sterilizer capacity of 8000 litres per hour is selected. Hence the design specifications based on fig 20 are as follows:

Balance Tank = 500 Litre, stainless steel

Pumps:

P_1, P_2, P_3 = 8000 Litre/hour each

Linking Pipes:

Diameters, ϕ = 7.5cm all round

Length, the varying lengths are indicated on the working drawings.

3.5 Pipe Line Problems

3.5.1 Material Selection

Milk poisoning due to presence of metal ions in the product is a problem that can only be solved by using the right material for the pipe line works. An optimal material selection is one that minimizes corrosion, adsorption and deposits, precipitation and colour changes, as well as withstand all the mechanical and thermal stresses involved in the production processes. In the dairy industry, the temperation variations in the pipe line range from $-4^{o}c$ for the chilled water line to $160^{o}c$ for the steam line and the chemicals that of necessity must flow in the pipe network from time to time are Lactic acid ($CH_3CH(OH)COOH$), trisodium phosphate (Na_3PO_3), sodium hydroxide ($NaOH$) and Nitric acid (HNO_3), chlorine (cl_2) and Bacteria. Different materials will now be considered in turns with a view to selecting the best for this purpose.

Glass: Glass is resistant to all alkalis and acids that are used in the milk production, or in the detergent for washing the pipe line, but it is easily damaged mechanically.

Stainless Steel: Stainless steel is resistant to alkalis and to certain acids. It is corroded by hydrochloric and sulphuric acids and also by lactic acid, which causes spot corrosion. It is further corroded by chlorine, particularly in its gaseons state or at low PH values.

Copper, Brass, Tin and Tinned Materials: These are sensitive to both strong acids and strong alkalis.

Aluminium: Aluminium is corroded by bacteria, by lye and strong acid except metalicate and nitric acid. Chlorine and iodine oxidize aluminium, but the oxide formed remains on the surface to protect the metal against further attacks.

Alloys: The trouble with using alloys is electrolytical corrosion. When two different metals obtain contact with one another through an electrolyte, for example, a detergent, the electrolysis the proceeds result in electrolytic corrosion. Stainless may be of different alloy compositions, for example, for reasons of machining, the steel used for unions is not the same as that used for piping. According to Dr. ⁻.V. Bockelmann

in a lecture delivered to Tetra Pak International AB, Lund Sweden in 1982, Weight losses due to electrolytic corrosion have been measured at 0.02 mg/cm^2/month in piping of nickel-chromium steel as compared with 0.7 mg/cm^2/month in the pipes of cupro-nickel steel. The weight loos of tin in the same system amounted to 0.015 mg/cm^2/month.

Plastics and Rubber: Water is absorbed by some plastics. Nylon adsorbs approximately 50 times as much water as polypropylene. PVC and acrylic plastics also absorb water. Teflon adsorbs protein which comes off in flakes. Parts made of rubber, like those made of plastics, can easily be damaged mechanically. Natural rubber adsorbs fat. Temperatures of over 100^{o}c are harmful to all rubbers.

From the foregoing discussions, nickel-chromium steel is recommended as the best material for pipe works.

3.5.2 Head Losses in Pipes

Other problems associated with the pipe line network is the fluid flow itself. Here we shall consider:

1. The Gas Lines; and
2. The Liquid Lines.

1. Gas Lines

For the gas lines, the fluid is compressible and therefore pressure variation is an infinitesimal quantity and hence can be neglected for all practical purposes.

2. Liquid Lines

For the liquid lines, the fluid is considered incompressible, and therefore we assume laminar flow all through. That is the fluid particles move along straight, parallel paths in layers or laminae. Because the magnitudes of the velocities of adjacent laminae are not the same, there exists sheer stress (τ) in between laminae. Laminar flow is governed by the law relating shear stress (τ) to rate of angular deformation, that is the product of the viscocity (μ) of the fluid and velocity grandient

$$\frac{dv}{dy}$$

$$\text{or} \quad \tau = \mu \frac{dv}{dy} \qquad N/m^2$$

The energy exchange for a non-compressible fluid in a pipe is presented in figure 22.

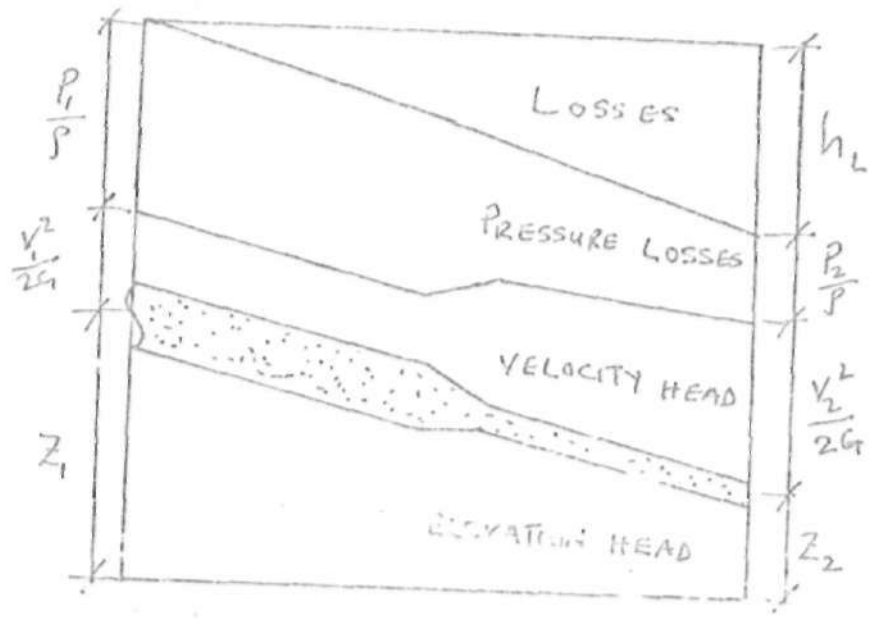


Fig. 22: Energy Losses in Pipes[14]

Bernoulli's equation, expressing the principle of energy conservation stated in the first law of thermodynamics can be written thus:

$$Z_1 + \frac{V_1^2}{2G} + \frac{P_1}{\rho} = Z_2 + \frac{V_2^2}{2G} + \frac{P_2}{\rho} + h_L$$

Where Z = Elevation above reference line

V = Velocity

P = Pressure

ρ = Density

h_L = Loss in head

Where some motive force is applied to the fluid, for example, by installing a pump in the pipeline, the additional head, h_p, would be recorded on the

left hand side of the energy balance. Thus:

$$h_p + Z_1 + \frac{V_1^2}{2G} + \frac{P_1}{\rho} = Z_2 + \frac{V_2^2}{2G} + \frac{P_2}{\rho} + h_L$$

It is important to note that flow continyity is preserved in a full-flowing pipeline, that is, the fluid flow rate constant. Thus:

$$Q_1 = Q_2 = Q$$

$$= A_1V_1 = A_2V_2$$

Where Q = flow rate

A_1 and A_2 are the cross-sectional areas at two points along the length of the pipeline.

The head loss, h_f, is mainly attributable to viscous-friction effects and to turbulent flow characteristics. Emperical expressions have been developed to estimatetthe viscous losses which may arise in a particular system, and the most useful equation is:

$$h_L = \frac{L}{d} f \frac{V^2}{2G}$$

where h_L = head loss

L = Length of the pipe

d = diameter of the pipe

f = friction factor

From the foregoing discussions we can deduce that head losses can be minimized by having:

1. Shorter runs of pipe
2. Pipes of larger diameters
3. Flow of lower velocities
4. Smaller differences between Z_1 and Z_2
5. Minimum number of bends, sudden enlargements and contraction of the pipe diameters.

CHAPTER FOUR

SEWAGE DISPOSAL

Sewage represents the rate of daily water supply with a small amount of inpure material that polutes the whole. This is a clear reason to separate the handling of storm water for dispersal to the ground or direct disposal to the water ways rather than permitting it to swell the volume of sewage requiring treatment.

The sewage system has to remove and dispose of the following:

1. Storm water from access roads, roofs of buildings gardens etc.
2. Domestic sewage i.e. excreta, urine and slop water from baths, wash hand basins, laundries.
3. Industrial wastes from production areas.

4.1 Storm Water Drainage System

Storm water drainage system is made up of all waste water that is considered safe for direct disposal to the surrounding ground or to the waterways. These include:

1. Water for garden watering = 14400 litre/day
2. Water for car washing = 5700 litre/day
3. Rain water, which is a function of the actual quantity of rain collected in the premises

area, and permeability of the ground (run-off factor) and the rate of rainfall per minute.

The most severe storm for which provision has to be made is the basis of design. In the northern part of the country, rainfall average is 100.150 centimetre during the wet seasons. In Kaduna this spans from April through September and the highest rate of rainfall ever recorded in Kaduna being 3.19 cm/hour (1.28 inches/hour) which occured on July 18, 1981 at 1342 hours Nigerian local time.[28] Therefore the basis of design for storm drainage in Kaduna and its environ is a rain intensity of 3.19 cm/hour (1.28 iches/hour).

The run-off factor for various land forms are as follows:

DESCRIPTION	RUN-OFF FACTOR
Water tight roof surfaces	0.90 - 0.98
Gardens, Lawns	0.25 - 0.50 0.05 - 0.25
Asphalt surfaces	0.80 - 0.90
Sandy soil	0.30 - 0.60
Concrete surface	0.85 - 0.95

Source: Agro-Met orological Bulletin of Nigeria, 1982 by the Nigerian Meteorological Services Headquarters, Lagos

TABLE 2

Date	Jan	Feb	Mar	Apr	May	June mm	July mm	Aug mm	Sept mm
1	-	-	-	-	-	8.2	-	0.1	-
2	-	-	-	-	-	-	26.1	-	14.4
3	-	-	-	-	-	-	-	1.5	-
4	-	-	-	-	7.8	-	45.5	6.3	-
5	-	-	-	-	33.5	8.3	11.2	16.6	-
6	-	-	-	-	2.3	12.6	1.3	-	-
7	-	-	-	-	1.5	-	-	-	-
8	-	-	-	-	10.1	0.7	10.8	12.8	-
9	-	-	-	-	-	-	-	-	-
10	-	-	-	-	-	-	2.2	-	-
11	-	-	-	-	-	-	-	-	-
12	-	-	-	-	-	1.9	2.8	1.8	-
13	-	-	-	-	-	20.4	-	-	-
14	-	-	-	-	-	-	0.5	-	-
15	-	-	-	-	7.6	-	-	-	-
16	-	-	-	-	-	7.7	0.9	31.9	-
17	-	-	-	-	22.6	-	-	7.4	-
18	-	-	-	-	-	-	-	-	-
19	-	-	-	-	21.2	1.4	51.8	-	-
20	-	-	-	-	1.4	2.0	6.1	12.6	-
21	-	-	-	-	18.1	2.8	15.6	-	-
22	-	-	-	-	-	27.4	-	-	-
23	-	-	-	-	-	-	-	-	-
24	-	-	-	-	8.7	-	0.9	-	-
25	-	-	-	-	20.2	-	13	-	-
26	-	-	-	-	4.1	26.4	13.8	22.6	-
27	-	-	-	-	-	-	-	-	-
28	-	-	-	-	20.8	-	53.2	-	-
29	-	-	-	18.5	-	29.8	-	15.9	-
30	-	-	-	0.7	-	-	4.4	5.5	-
31	-	-	-	-	-	18.1	16.4	6.6	-

Table 4: Values of Monthly and Annual Rainfall Normals 1951 – 1980

STATION	ELEVATION IN FEET	LONG. O 'E	LAT. O 'E	IDENTIFICATION NUMBER	JAN.	FEB.	MARCH	APRIL	MAY	JUNE	JULY	AUG.	SEPT.	OCT.	NOV.	DEC.	ANNUAL NORMAL
1. OPOBO	10	07°33'	04°34'	0407.33	1.96	3.64	6.53	8.61	15.11	23.12	26.87	20.80	21.08	15.36	7.62	2.29	153.05
2. FORCADOS	20	05°26'	05°21'	0505.22	1.34	2.44	5.61	8.62	15.62	20.71	27.07	9.71	22.87	15.25	6.33	2.55	136.71
3. CALABAR	203	08°21'	04°58'	0403.52	1.6	3.27	6.57	9.09	11.33	16.77	16.07	13.73	16.27	12.77	7.52	2.09	117.58
4. PORTHARCOURT	64	07°01'	04°51'	0407.50	1.23	2.82	5.67	7.79	9.62	11.82	13.57	10.93	15.76	11.19	4.80	1.69	96.89
5. OWERRI	230	07°01'	05°29'	0507.20	0.94	1.85	5.18	6.93	10.33	12.20	13.89	11.60	17.31	10.85	3.18	0.73	95.02
6. DEGEMA	10	06°46'	04°45'	0406.44	1.63	3.07	5.11	7.58	9.14	11.47	11.87	9.71	14.65	10.28	4.94	1.93	91.36
7. SAPELE	20	05°42'	05°54'	0505.54	0.94	1.71	4.67	6.95	9.53	13.10	15.71	8.67	15.10	10.65	3.50	0.73	91.12
8. IKOTEKPENE	250	07°43'	05°11'	0507.14	1.58	2.45	4.87	7.26	9.31	10.55	12.49	10.91	13.10	9.96	4.11	1.54	88.07
9. BENIN	255	05°36'	06°19'	0605.13	0.65	1.38	4.17	6.54	8.17	11.83	12.86	8.08	12.42	9.70	3.02	0.53	79.32
10. ONITSHA	300	06°47'	06°10'	0606.14	0.55	1.28	2.81	6.34	7.67	10.13	11.75	8.98	12.99	9.98	2.09	0.50	75.07
11. NKWELE	350	06°50'	06°14'	0606.15	0.53	1.10	2.73	5.36	8.41	9.66	10.97	8.04	12.29	10.23	2.29	0.44	72.38
12. LAGOS	8	03°24'	06°27'	0603.32	1.31	1.78	4.16	5.46	11.27	17.5	9.86	2.59	5.90	8.09	2.99	1.04	71.98
13. OGOJA	750	08°43'	06°39'	0608.34	0.53	0.80	2.05	4.70	8.77	11.03	7.85	10.02	11.46	11.45	2.20	0.51	71.39
14. ENUGU	464	07°33'	06°28'	0607.23	0.84	1.10	2.69	5.33	10.05	9.80	7.76	6.48	12.15	10.26	1.99	0.65	70.11
15. WAMBA	940	04°50'	07°06'	0704.05	1.98	1.25	4.32	5.89	7.02	8.51	9.33	5.62	10.10	8.22	3.03	0.99	66.26
16. WAMBA	1050	08°37	08°55'	0808.53	0.01	0.25	1.11	3.27	5.96	7.55	13.12	14.93	12.95	5.14	0.53	0.08	64.91
17. JOS	4215	08°54'	09°52'	0908.56	0.09	0.14	0.93	3.37	8.07	9.03	12.50	10.80	8.94	1.55	0.18	0.07	55.97
18. AGEGE	140	03°18'	06°39'	0603.31	0.86	1.28	3.82	4.24	8.13	10.28	6.38	2.13	6.32	6.96	3.18	1.08	54.67
19. IJU	150	03°21'	06°41'	0603.42	0.70	1.10	3.42	4.20	8.02	10.15	7.19	2.22	6.23	7.09	3.35	0.77	54.43
20. YANDEV	400	09°01'	07°23'	0709.20	0.21	0.55	1.54	4.21	7.15	7.50	5.25	6.17	10.46	8.52	0.68	0.20	52.45
21. MINNA	848	06°32'	09°37'	0906.33	0.03	0.18	0.85	2.27	5.44	6.95	8.25	9.48	10.83	5.52	0.40	0.14	50.35
22. ILORIN	1008	04°35'	08°29'	0804.23	0.47	0.58	2.66	3.66	6.60	7.59	5.19	5.44	9.87	6.72	1.11	0.42	50.30
23. KADUNA	2116	07°27'	10°36'	1007.32	0.04	0.17	0.59	2.44	5.78	6.50	8.05	10.73	11.30	3.58	0.21	0.06	49.39
24. BIDA	472	06°01'	090°6'	0909.00	0.20	0.22	1.12	2.89	5.75	7.72	7.66	8.18	10.06	3.79	0.34	0.09	48.15
25. OFFA	1250	04°43'	08°09'	0804.04	0.46	0.83	2.67	4.57	6.39	6.10	3.95	4.38	8.13	7.88	1.35	0.58	47.47
26. IBADAN AGRIC	650	03°51'	07°23'	0703.25	0.47	0.83	3.56	5.15	5.92	6.83	5.53	3.17	6.69	6.85	1.89	0.32	47.19

Table 4: Values of Monthly and Annual Rainfall Normals 1951-1980 (Contd.)

STATION	ELEVATION IN FEET	LONG. O 'E	LAT. O 'N	IDENTIFICATION NUMBER	JAN.	FEB.	MARCH	APRIL	MAY	JUNE	JULY	AUG.	SEPT.	OCT.	NOV.	DEC.	ANNUAL NORMAL
27. ABEOKUTA	350	03°21'	07°07'	0703.09	0.56	1.13	3.48	[illegible]	5.81	7.25	6.04	3.35	5.95	6.09	1.90	0.57	47.03
28. IBADAN AGRIC	745	03°54'	07°16'	0703.2[illegible]	0.48	0.94	3.61	[illegible].66	5.99	7.04	5.36	3.27	6.53	6.53	1.88	0.44	46.71
29. KONTAGORA	1300	05°28'	10°21'	1005.22	0.11	0.05	0.36	1.76	4.[illegible]6	5.87	7.90	10.19	10.66	4.22	0.26	0.03	45.56
30. OYO	750	03°43'	07°54'	0703.54	0.21	[illegible].02	3.11	5.18	[illegible].74	5.67	4.25	3.68	7.02	6.76	2.00	0.35	45.00
31. MAIGANA	2100	07°56'	11°02'	1107.05	0.01	0.07	0.44	1.47	5.[illegible]3	6.06	8.46	12.98	8.50	1.36	0.02	0.00	44.40
32. BAUCHI	2200	09°50'	10°18'	1009.15	0.00	0.03	0.19	1.52	[illegible].93	5.74	9.67	14.07	7.26	1.38	0.05	0.01	43.84
33. LOKOJA	135	06°44'	07°48'	0706.54	0.12	0.47	1.94	3.05	5.75	5.99	6.47	5.74	7.76	4.90	0.41	0.12	42.76
34. ZURU	1400	05°13'	11°26'	1105.21	0.01	0.01	0.13	[illegible].0[illegible]	3.22	4.48	6.72	11.74	9.99	2.97	0.13	0.01	40.45
35. YOLA	610	12°±4'	09°28'	0912.12	0.00	0.00	0.32	1.5[illegible]	5.19	6.08	6.06	7.04	7.99	2.50	0.15	0.02	37.19
36. KANO	1650	08°33'	11°59'	1108.53	0.00	0.01	0.06	0.3[illegible]	[illegible].80	4.69	8.24	12.24	5.41	0.54	0.01	0.00	34.33
37. AZARE	1350	10°19'	11°40'	1110.[illegible]1	0.00	0.00	0.01	0.35	2.[illegible]4	3.17	7.58	11.94	5.10	0.70	0.02	0.00	31.20
38. HADEJIA	1150	10°04	12°27'	1210.20	0.00	0.00	0.01	0.07	1.1[illegible]	2.77	7.21	10.54	3.75	0.29	0.00	0.00	26.10
39. MAIDUGURI	1160	13°05'	11°51'	1113.50	0.00	0.00	0.02	0.[illegible]5	1.34	3.05	7.09	8.94	4.40	0.90	0.01	0.00	25.92

SOURCE: Agro-Meteorogical Bulletin of Nigeria

TABLE 5: RAINFALL INTENSITIES FOR HEAVY PRECIPITATION May 1981

Station	Day	15 min.		30 min.		60 min.		2 hrs		3 hrs.		6 hrs.		12 hrs		18 hrs.		24 hrs.	
		T	A	T	A	T	A	T	A	T	A	T	A	T	A	T	A	T	A
KADUNA	20							2016	2.02										
ZARIA	9	1554	0.86	1554	0.93														
"	26							0654	1.20										
YELWA	23			1943	0.95					1945	2.35								
MAIDUGURI	4			1443	0.62			1443	2.9										
NGURU	8	1648	0.62			1643	0.06	1640	1.84										
	29			1448	0.70	1448	1.3												

N.B. T - Time of Commencement of Intensity Measurement

A - Amount of Rainfall in specified Period in Ins.

TABLE 6: RAINFALL INTENSITIES FOR HEAVY PRECIPITATION JUNE 1981

Station	Day	15 min		30 min		60 min		2 hrs.		3 hrs.		6 hrs.		12 hrs.		18 hrs.		24 hrs.	
		T	A	T	A	T	A	T	A	T	A	T	A	T	A	T	A	T	A
KADUNA	12	1840	0.13					1420	2.10										
ZARIA	16					0.93	1725	0.91											
	30			0354	0.53														
YELWA	15			1943	0.45														
MAIDUGURI	6			1443	0.62	1443	1.40												
NGURU	17	1648	0.62	1648	0.52	1643	1.06												
	30	1448	0.66	1448	0.70														

N.B. T - Time of Commencement of Intensity Measurement

A - Amount of rainfall in specified period in Ins.

TABLE 7: RAINFALL INTENSITIES FOR HEAVY PRECIPITATION

Station	Day	15 min		30 min		60 min		2 hrs.		3 hrs.		6 hrs.		12 hrs		18 hrs		24 hrs	
		T	A	T	A	T	A	T	A	T	A	T	A	T	A	T	A	T	A
KADUNA	18			1054	0.66	1342	1.28												
ZARIA	10	1554	0.86	1554	0.55														
	22			0354	0.53														
YELWA	28			1943	0.95	1943	1.15												
MAIDUGURI	5			1443	0.62	1443	1.49												
NGURU	8	1648	0.62	1648	0.52	1643	1.06												
	28	1448	0.66	1448	0.70														

N.B T - Time of Commencement of Intesity Measurement
A - Amount of Rainfal in Specified Period in Ins.

TABLE 8: RAINFALL INTENSITIES FOR HEAVY PRECIPITATION MAY 19

Station	Day	15 min		30 min		60 min		2 hrs.		3 hrs.		6 hrs.		12 hrs		18 hrs		24 h
		T	A	T	A	T	A	T	A	T	A	T	A	T	A	T	A	T
KADUNA	10			0154	0.66			1725	1.5									
ZARIA	14	1554	0.86	1554	0.53													
	23			0354	0.53													
YELWA	9	0705	0.61			1940	1.15	1943	1.13									
MAIDUGURI	4			1443	0.62	1443	1.49											
NGURU	2	1648	0.62	1648	0.52			1648	0.98									
	19	1448	0.66			1448	0.87											

N.B. T - Time of Commencement of Intensity Measurement
A - Amount of Rainfall in Specified Period in Ins.

TABLE 9: RAINFALL INTENSITIES FOR HEAVY PRECIPITATION JUNE 1982

Station	Day	15 min		30 min		60 min		2 hrs.		3 hrs.		6 hrs.		12 hrs		18 hrs		24 hrs	
		T	A	T	A	T	A	T	A	T	A	T	A	T	A	T	A	T	A
KADUNA	6	0520	0.3					1050	1.32										
ZARIA	21					1508	0.9												
	13			0354	0.53														
YELWA	15	1940	0.2			19	1.05												
MAIDUGURI	21			1443	0.62	14	1.49												
NGURU	17			1648	0.52														
	20					1430	1.2												

N.B. T - Time of Commencement of Intensity Measurement
A - Amount of Rainfall in Specified Period in Ins.

TABLE 10: RAINFALL INTENSITIES FOR HEAVY PRECIPITATION JULY 1982

Station	Day	15 min.		30 min		60 min.		2 hrs.		3 hrs.		6 hrs.		12 hrs		18 hrs		24 hr
		T	A	T	A	T	A	T	A	T	A	T	A	T	A	T	A	T
KADUNA	8	0415	0.4					1940	2.05									
ZARIA	12	1554	0.86	1554	0.53													
	27			0354	2.53													
YELWA	23	1820	0.21															
MAIDUGURI	26							1350	2.22									
NGURU	13	0230	0.59	0230	0.70	0230	1.65											

N.B. T - Time of Commencement of Intensity Measurement
A - Amount of Rainfall in Specified Period in Ins.

TABLE 11: RAINFALL INTENSITIES FOR HEAVY PRECIPITATION MAY, 1983

Station	Day	15 min. T	15 min. A	30 min. T	30 min. A	60 min. T	60 min. A	2 hrs. T	2 hrs. A	3 hrs. T	3 hrs. A	6 hrs. T	6 hrs. A	12 hrs. T	12 hrs. A	18 hrs. T	18 hrs. A	24 hrs. T	24 hrs. A
KADUNA	4	0125	0.42					1740	1.70										
ZARIA	[illegible]					1320	1.15												
	30	1940	0.29	0354	0.53														
YELWA	13	1526	0.31					1930	1.40										
MAIDUGURI	[illegible]	1204	0.31																
NGURU	20	1648	0.62	[illegible]48	0.52	1643	1.06												
	29	1520	0.63	1[illegible]	0	0.71													

N.B. T - Time of Commencement of Intensity Measurement
A - Amount of rainfall in specified period in Ins.

TABLE 12: RAINFALL INTENSITIES FOR HEAVY PRECIPITATION JUNE 198

Station	Day	15 min.		30 min.		60 min.		2 hrs.		3 hrs.		6 hrs.		12 hrs.		16 hrs.		24 hrs.	
		T	A	T	A	T	A	T	A	T	A	T	A	T	A	T	A	T	A
KADUNA	16			0430	0.68	1520	1.20												
ZARIA	18					0354	1.23												
	30	1830	0.85																
YELWA	10	1640	0.23																
	19			1800	0.54	1800	1.15												
MAIDUGURI	20	0453	0.15			0453	1.40												
NGURU	7	1648	0.62	1648	0.52	1643	1.06												
	30	1448	0.66	1448	0.70														

N.B. T - Time of Commencement of Intensity Measurement

A - Amount of Rainfall in specified period in ins.

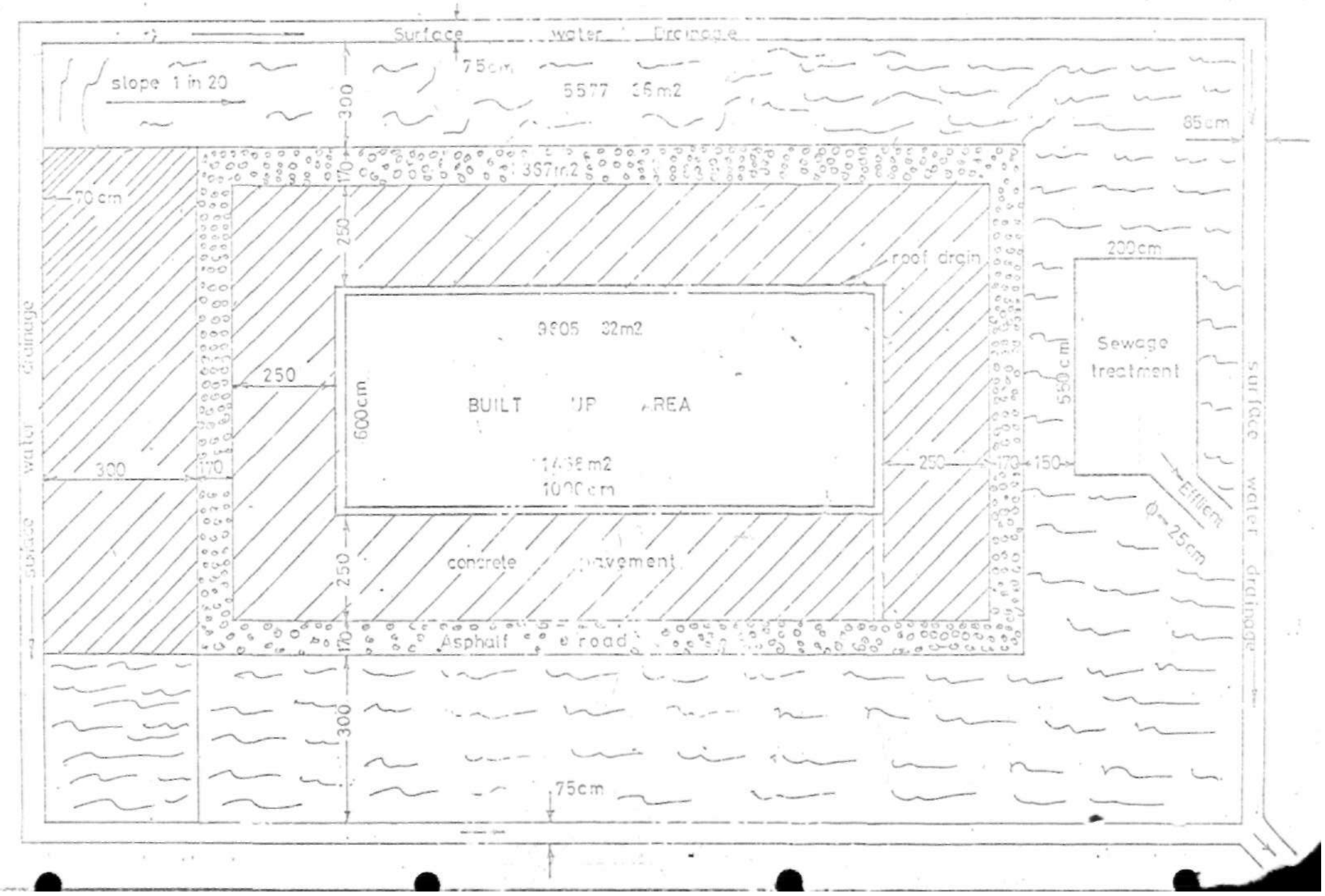
TABLE 13: RAINFALL INTENSITIES FOR HEAVY PRECIPITATION JULY 1983

Station	Day	15 min.		30 min.		60 min.		2 hrs.		3 hrs.		6 hrs.		12 hrs.		18 hrs.		24 hrs.	
		T	A	T	A	T	A	T	A	T	A	T	A	T	A	T	A	T	A
KADUNA	3	1615	0.37	1615	0.67	1615	1.26												
ZARIA	30	0253	0.82	0354	0.53														
YELWA	13	1943	0.21																
MAIDUGURI	6	1130	0.31	130	0.65														
	20					1443	1.49												
NGURU	15	0608	0.60	0608	0.96	0608	1.20												

N.B. T - Time of Commencement of Intensity Measurement
A - Amount of Rainfal in specified period in Ins.

SOURCE: Agro-Meteorological Bulletin of Nigeria

FIG. 23 DESIGN OF STORM WATER DRAINAGE SYSTEM



The land area of the premises is 9605.32 m^2 broken down as follows:

Grass lawns and gardens	- 5577.36 m^2
Concrete surfaces	- 1367 m^2
Built up (roofed) area	- 1458 m^2
Asphalt surfaces	- 1312.32 m^2

From table 2 the combined run-off factor for these surfaces is:

$$\frac{(0.5\times5577.36) + (0.95\times1367) + (0.98\times1458) + (0.9\times1312.32)}{9605.32}$$

$$= 0.6972$$

And from table 5, the severest rate of rainfall to be provided for in the design of storm drainage system is 3.19 cm/hour or 0.0319 m/hour.

Hence flood rate Q = 9605.32 x 0.0319 x 0.6972

= 213.6 m^3/hour

Hence to optimize the effectiveness and safety of the storm drainage system the capacity of the gutters (V) should not be less than 213.6 m^3

From fig 23, perimeter of the gutters

= 375.35 m

∴ Minimum cross sectional area of the gutter:

$$= \frac{213.6}{378.35} = 0.566m^2$$

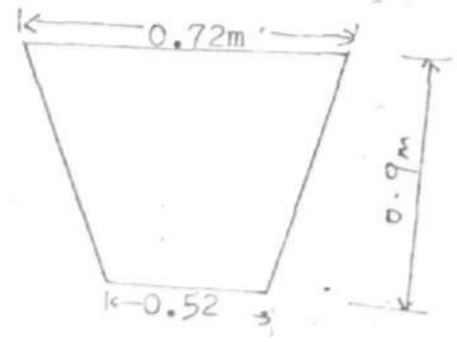
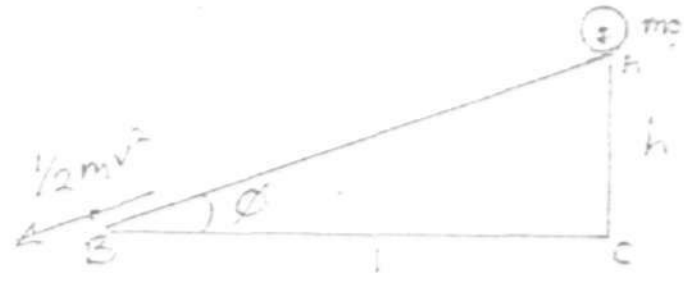


Fig 2ba: Section of the Storm Drainage Gutter

The fall of the gutter is such that will give a velocity of flow of 110-150cm/sec or 72-90m/min. To compute the fall (n) of the gutter consider a quantity of storm water of mass m at a point in the gutter as shown at zero velocity i.e. $V_A = 0$.



Let h be the fall (slope) of the floor of the gutter, such that:

tan Ø = h

let V = velocity of the storm water at point B i.e. $V_B = V$

For energy blance,

Potential energy at point A = kinetic energy at Point B

i.e. $\frac{1}{2} mv^2 = mgh$

or $v^2 = \sqrt{2gh}$

Hence the formula used for calculating the discharge of a channel is

$Q = av = \sqrt{2gh}$

where Q = discharge in m^3/sec

a = area of cross section in m^2

v = mean, velocity in the channel in m/sec.

Cd = coefficient of discharge which is = 1 for open gutters and varies between 0.6 and 0.8 for pipes.

g = acceleration fdue to gravity = $10mS^{-2}$

h = fall of the gutter (or slope) = tangent of the angle subtended by the floor of the gutter and the horizon.

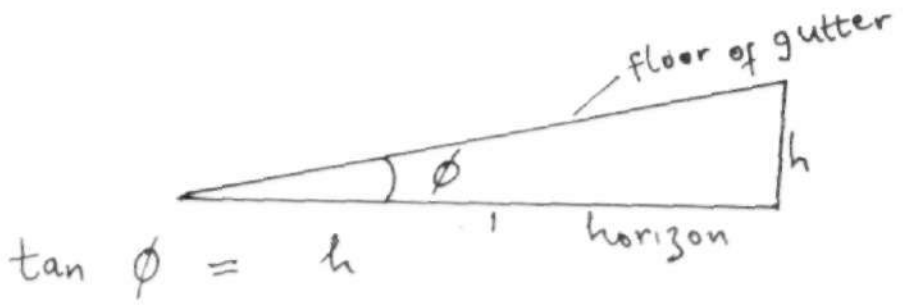


Fig 24b: Fall of the Gutter in Degrees

A desirable velocity of flow of 120-150 cm/sec i.e. 1.2 to 1.5 m/sec has already been specified. Hence substiting in

$$Q = av = Cd\, a\sqrt{2gh}$$

$$\text{or} \quad v = Cd\sqrt{2gh}$$

Cd = 1 for open channel

g = 10 m^{-2}

$v^2 = 20\,h$

$$h = \frac{v^2}{20}$$

$$\phi = \tan^{-1} h = \tan^{-1} \frac{v^2}{20}$$

when $V = 1.2$, $\phi = \tan^{-1} \frac{(1.2)^2}{20}$

$= \tan^{-1} 0.072$

$= 4^{\circ}$

When $V = 1.5$, $\phi = \tan^{-1} \frac{(1.5)^2}{20}$

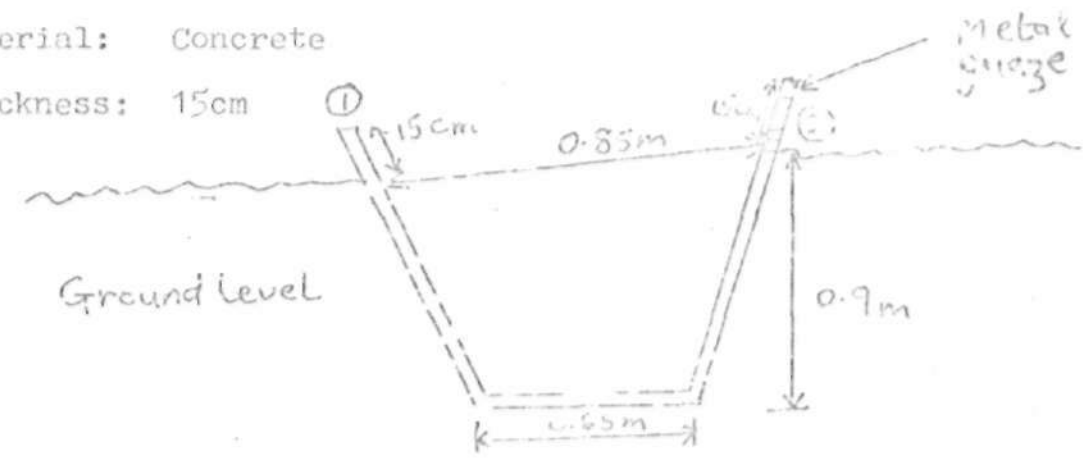
$= \tan^{-1} 0.1125$

$= 6.4^{\circ}$

4.2 Details of the Design of the Storm Drainage Gutter

Material: Concrete

Thickness: 15cm

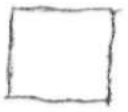


Edge 1: All outer edges of the gutter should project 15cm above ground level to exclude flood from outside the premises of the factory.

Edge 2: At the same level with the ground for unhindered flow of flood into the gutter.

The inner edges should be equiped with about 15cm metal guaze to prevent soil and other solids from entering the gutter to tend to block it. Effort should be made to clear the guaze of debris from time to time.

Notes on Fig 23



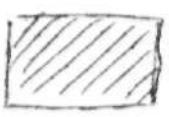
Built up area



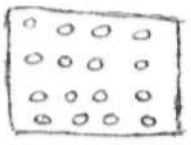
Gutter



Grass Lawns and Gardens



Concrete Pavements



Asphalt Roads

The land area of the premises is 9605.32 m^2 broken down as follows:

Grass lawns and gardens	- 5577.36 m^2
Concrete surfaces	- 1367 m^2
Built up (roofed) area	- 1458 m^2
Asphalt surfaces	- 1312.32 m^2

From table 2 the combined run-off factor for these surfaces is:

$$\frac{0.50 + 0.95 + 0.98 + 0.90}{4}$$

$$= 0.8325$$

And from table 4.4b, the severest rate of rainfall to be provided for in the design of storm drainage system is 3.19 cm/hour or 0.0319 m/hour.

Hence flood rate Q = 9605.32 x 0.0319 x 0.8325

= 255.386 m^3/hour

Hence to optimize the effectiveness and safety of the storm drainage system the capacity of the gutters (V) should not be less than 255.386 m^3

From fig 4.1, perimeter of the gutters

= 378.35 m

For energy blance,

Potential energy at point A = kinetic energy at Point B

i.e. $\frac{1}{2}\ mv^2 = mgh$

or $v^2 = \sqrt{2gh}$

Hence the formula used for calculating the discharge of a channel is

$Q = av = \sqrt{2\ g\ h}$

where Q = discharge in m^3/sec

a = area of cross section in m^2

v = mean, velocity in the channel in m/sec.

Cd = coefficient of discharge which is
= 1 for open gutters and varies between 0.6 and 0.8 for pipes.

g = acceleration fdue to gravity
= $10mS^{-2}$

h = fall of the gutter (or slope)
= tangent of the angle subtended by the floor of the gutter and the horizon.

4.3 Sewers System

The sewers system transport all liquid wastes i.e. excreta, urine, slop water from baths, wash hand basins, laundaeries and industrial wastes from production areas. Details of the design is shown on plate 1.

The whole sewers are underground with manholes located as shown to ease inspection and clearing of deposits if any. The pipe sizing at various stages depend on the quality of the sewage as shown.

The velocity of flow should not be less than 0.8m/S in order to be self cleansing. Hence the required slope of laying the sewer pipes is given by eqn̄ 4.1 or

$$\text{or} \qquad V = Cd\sqrt{2gh}$$

where V = velocity of flow = 0.8 m/s

Cd = coefficient of discharge = (0.6-0.8) (for pipes)

g = acceleration due to gravity

h = fall of gutter = tan ϕ where ϕ = angle of slope

hence substituting in eqn̄ 4.1

$$0.8 = 0.7\sqrt{2gh} = 0.7\sqrt{20h}$$

$$\text{or} \quad \sqrt{20h} = \frac{0.8}{0.7} = 1.143$$

$$20h = (1.143)^2 = 131$$

$$\therefore \quad h = \frac{1.31}{20} = 0.065 = \tan\phi$$

$$= 3.74^\circ$$

4.4 Sewage Treatment

Since the passing of the rivers' pollution act it is no longer permissible to pass crude sewage into rivers unless the minimum amount of water in such rivers is at all times capable of diluting the sewage five hundred times. It has then become necessary to treat the sewage so that a harmless and pure effluent is obtained.

4.5 Analysis of Sewage

The impurities present in sewage occure floating, in suspension, in solution and in colloidal form. Suspended matter consists of material of a greater density than water, which is to the treatment plant, because the velocity or flow in the sewers is too great to allow it to settle. Colloids are finely divided particles that appear from physical tests, such as sedimentation to be in solution, and yet will not pass through a filter paper.[18] They exist in an intermediate state between suspension and solution, and their removal forms one of the most difficult problems with which is encountered in sewage disposal. They are too fine for settlement, but are coarse enough to clog filters of fine material.

It is difficult to make an exact chemical analysis of a sample of sewage. Much of the material is organic and includes living organisms of animal and vegitable nature, together with substances produced by their digestive processes and by their death and decay. Such impurities are necessarily unstable, for they are in a continual state of decomposition. Even if it were possible to arrest the development of these changes the various elements, of which the organic matter is made up, are combined together in such a complicated mannerthat it is very difficult to determine the exact composition of the resultant compounds.

It is, however, possible to make certain analyses which afford the necessary information for determining the kind of treatment to which a particular sample of sewage should be subjected, or whether the affluent from a certain process is sufficiently purified. Thus, the determination of the quantity of nitrogen present, and the manner in which it is combined with other elements indicate in the case of crude sewage whether it is fresh or stale and how much putrescible matter it contains, and in the case of an effluent, how far oxidation has gone. The analyses are shown in tables 14 and 15.

Table 14: Quality Characteristics of Waste Water

Characteristic	Sanitary Waste Water	Factory Waste
PH	7-9	9.5
Temperature	20-30°c	20-30°c
Turbidity (mg/L)	30	20
Total Solids mg/L	500	1000
Biological Oxygen Demand (B.O.D.) mg/L	350	1000

Table 15: Quality Characteristic of Treated Waste Water

Characteristic	Effluent
PH	4-6
Turbidity mg/L	10
Total Slids mg/L	150
B.O.D. mg/L	20

4.6 Design of the Sewage Treatment Plant

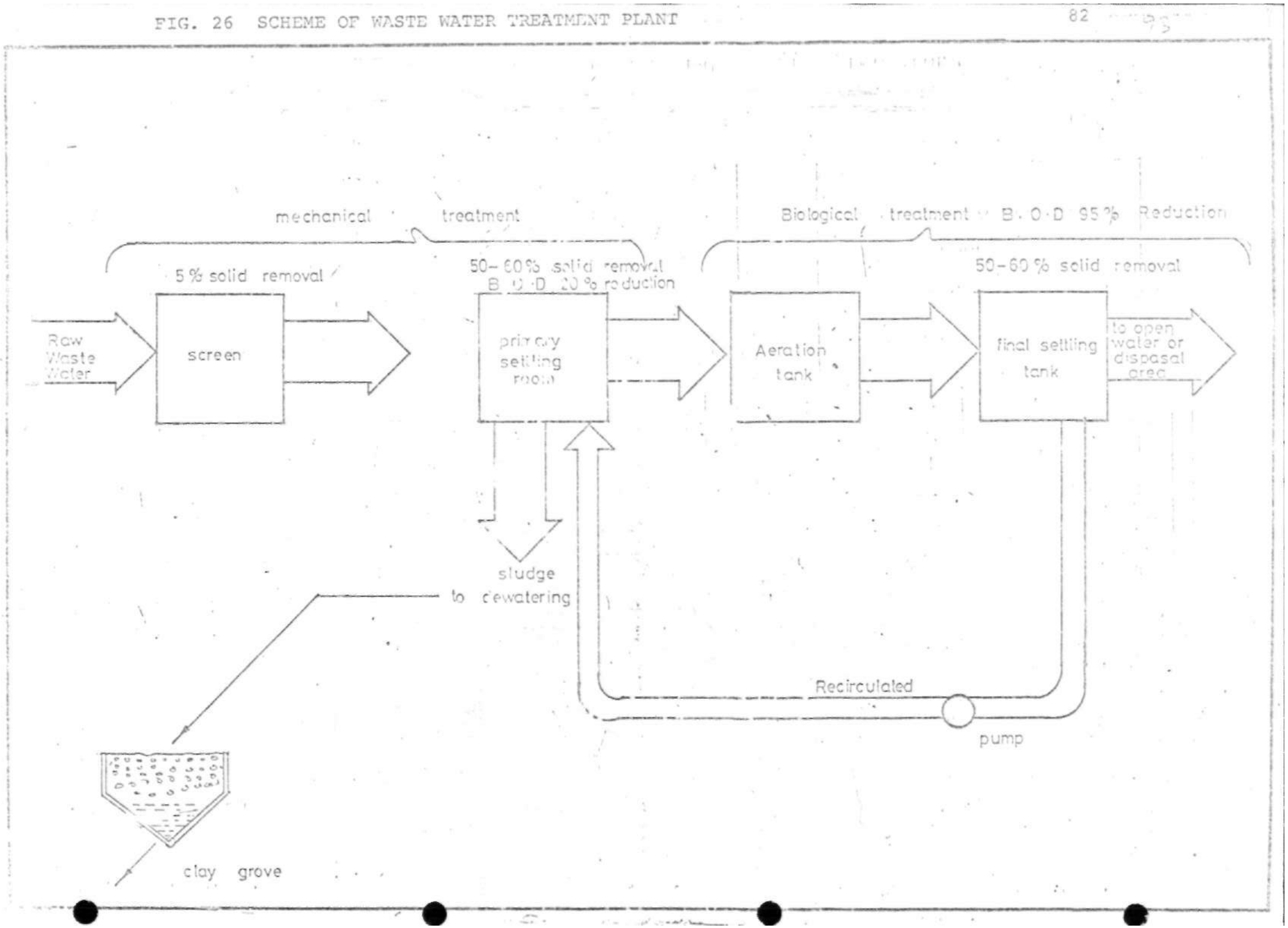
The layout of the waste treatment plant is based upon the worst combination of flow and pollutant - load variation that may be encountered. It is almost inevitable that pipelines will break and tanks will inadvertently be dumped. Emergency operating conditions may increase waste volume even as storm water is not treated. The scheme of the waste water treatment plant is shown in fig 26.

4.6.1 Screening

Screening is a mechanical process that separates particles on the basis of size. There are several types which have static, vibrating or rotating screens. Openings in the screening surfaces range from several centimetresdown to 35 microns (μm). The most commonly used screens in industrial processes are classified as:

1. Bar screens for removing large suspended solids.
2. Rotating drums having screen openings ranging down to 250 μm for removing suspended solids on the outer surface of the drum.
3. Vibrating screens with screen cloth down to 100
4. Inclined stationary screens down to about 250
5. Microstrainers with rotating drums that retain suspended solids on the inside of the drum that

FIG. 26 SCHEME OF WASTE WATER TREATMENT PLANT



screen apertures down to 23 The amount of solids removal depends upon screen size and characteristics of the waste water being filtered. Microstrainers are known to be equivalent to sand filters in their capacity to remove suspended solids, including microscopic solids such as various forms of plankton and general microscopic debris.

4.6.2 Design of the Screen

Bar screen is chosen for this project.

From $Q = VA$

we can determine the area of the bar screen (A)

where Q = quantity of sewage flowing per hour

V = velocity of flow of sewage

A = area of channel = area of screen.

The quantity of seware flowing per hour

= 60% of water demand of factory

i.e. $Q = 0.6 \times 246962$ litres per day

$= \frac{0.6 \times 246962}{24}$ litres per hour

= 6174 litres per hour

= 6.74 m^3 per hour

V = 0.8 m/s = 48m/hour

$\therefore A = \frac{Q}{V} = \frac{6.74}{48} = 0.14\ m^2$

$= 0.5 \times 0.28\ m^2$

i.e. width of screen = 0.28m, depth of sereen = 0.5m

4.6.3 Design of the Settling Tanks

The removal of the suspended matter present in sewage is carried out in the settling tanks. The tanks may be of:

1. the horizontal flow type
2. the vertical flow type
3. the radial flow type

The maximum deposition of material occurs when the velocity of flow is zero. But in practice, the sewage is kept in continual motion during its passage through the treatment works and, if the cross-sectional area of the sedimentation tanks is sufficiently large compared with that of the main sewer and the time of detention in the tank is long enough, the majority of the suspended matter should be removable without stoping flow altogether.

4.6.4 Horizontal Flow Tanks

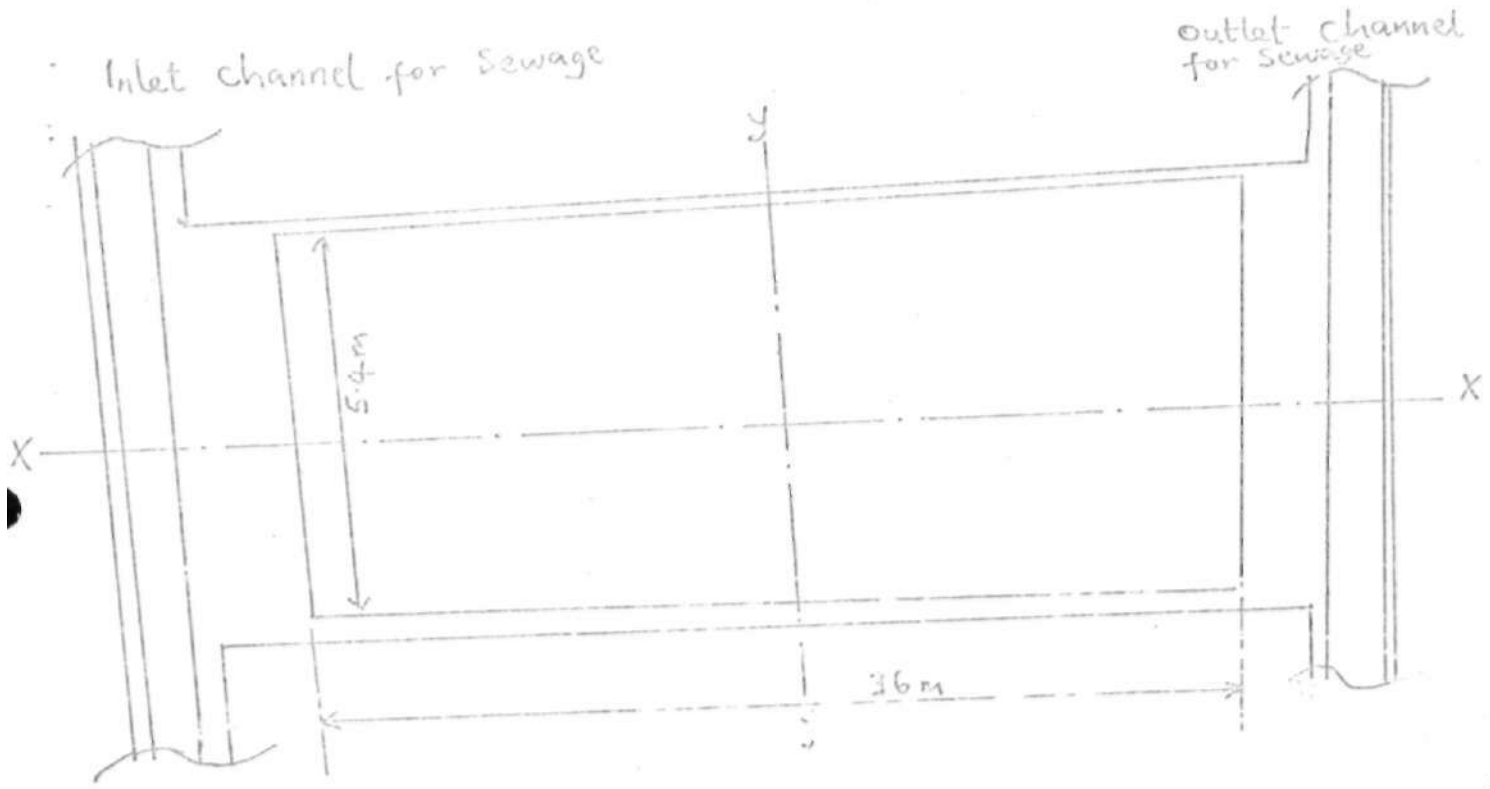
In sedimentation tanks that work on the continuous flow principle, the sewage flows through the tank at a velocity low enough to enable the requisite proportion of suspended solids to settle. The amount of clarification depends upon the velocity of flow and the period of detention. The time of detention is the interval that elapses between the instant that an element of the liquid enters the tank and the instant at which

it leaves it. The longer the period of detention and the slower the velocity of flow, the greater the amount of suspended matter removed. Under normal conditions, with sewages of average strength, about 50% of the suspended solids will settle in two hours, and the rate of deposition falls off considerably after this period. The dry weather flow is usually taken as the basis of design, although the minimum permissible clarification should be obtainable when the tanks are working at the maximum rate, which will usually be when they are dealing with three times that amount. Assuming it is uneconomical to extend the period of detention beyond three to five hours when the maximum quantity is being dealth with, the time of detention based on the dry weather flow will be from 9 to 15 hours.

To prevent it from becoming septic, either the whole of the sludge or that which has remained longest in the tank should be removed at intervals of one week or less. The capacity of the tank must be such that it can accommodate the maximum quantity of sewage retained in it, or flowing through it during the requisite period of detention and the amount of sludge deposited between successive cleanings. The greater part of the sludge is deposited near the inlet because the heavier and coaser solids settle more quickly than

the finer ones. For this reason the sludge outlet should be located near the inlet and the bottom of the tank slopes from the effluent outlet towards it. It is not practicable to construct horizontal flow tank with slopes steep enough to render them self cleansing. Consequently squeegees operated by hand or mechanical means have to be used to push the sludge towards the outlet. The sludge should be pushed into a sump at the outlet end of the tank from which it is pumped away for further treatment or final disposal. Power operated skimming blades should be used when mechanical scaapers are used. These help coagulation of the suspended matter and separate grease and floating matter by pushing it into a trough running transversely across the tanks.

Fig 27: Design of Horizontal Flow Settling Tank

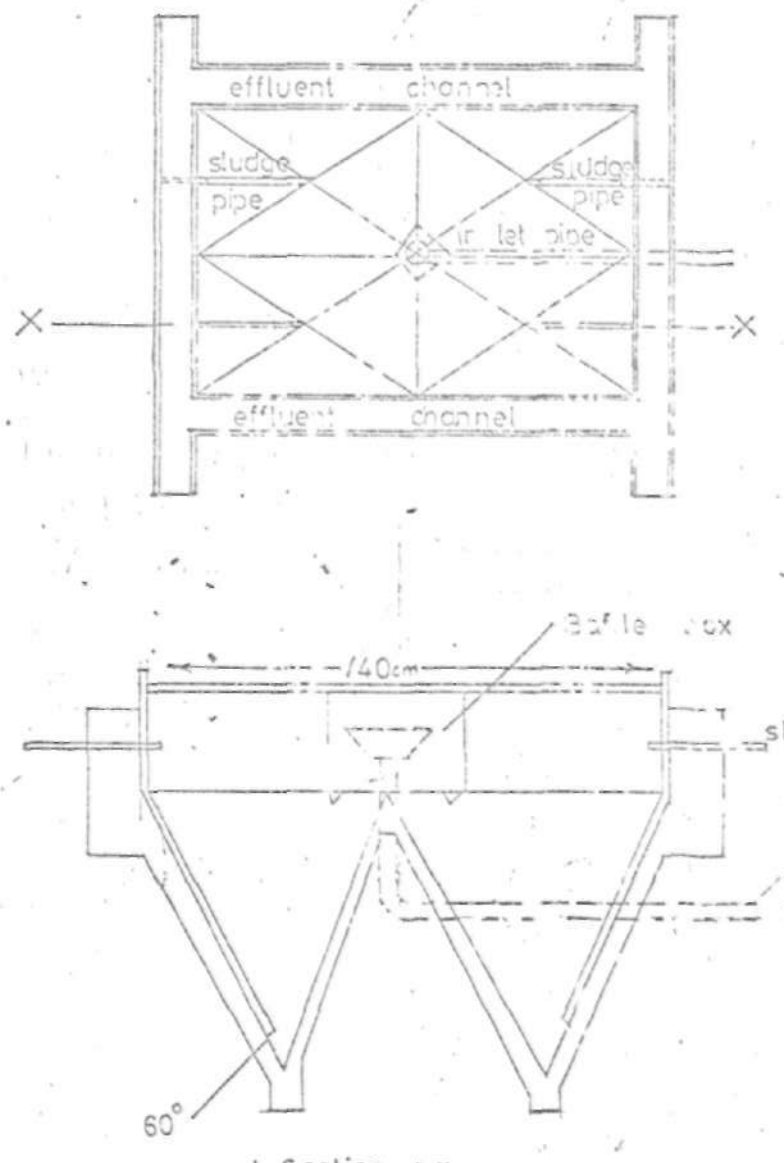


4.6.5 Vertical Flow Tanks

Vertical flow tanks are deep tanks, circular or square in plan, into which the sewage is admitted below the surface, but above the highest sludge level and from which the effluent is discharged into troughs constructed round the periphery of the top of the tank or diametrically across it. The sides are made vertical for the upper part, but at the bottom slope steeply (at 45^{o}c or more) to facilitate gravitational removal of the sludge, the outlet for the latter being provided at the apex of the inverted cone or pyramid formed by the lower part of the tank.

The action that takes place in a vertical flow tank is as follows: Particles of suspended matter, on entering the tank, are borne upwards in the liquid. Their upward velocity diminishes as they rise owing to the downward effect of their own weight. At certain level there is a neutral plane where the motion of the particles is neither upward nor downward and those rising from inlet are met by those falling from the surface. Here the suspended matter collects and coagulates. When the density of the mass becomes too great, portion of it detach themselves and fall to the bottom of the tank in the form of sludge. One of the great

FIG. 28 VERTICAL FLOW TANK



advantages of the vertical flow tank is that owing to the steep slope of the lower part, the hychostatic head of sewage overlying the sludge is sufficient to cause the latter to flow through the outlet without assistance. In case where the disposal site is at a higher level than the tank, the effluent has to be pumped. The disadvantage is that it is inpracticable to construct vertical flow tanks where the ground is unsuitable, depths of 30 feet or more being not unusual.

4.6.6 Radial Flow Tanks

A radial flow tank is a comparatively shallow circular tank in which the diameter is large relative to its depth. The sewage enters at the centre and flows into the collecting channel which completely surrounds the tank. The heavier particles are able to settle out at the relatively high velocities prevailing near the outlet weir enable most of the finer and more resistant particles to settle. The sludge is collected in a conical shaped pocket situated at the centre immediately below the inlet. It is not practicable to give this tank a self-cleansing slope, therefore a power driven scraper is provided which more slowly on the gently-sloping floor and push the sludge into the central pocket.

Surrounding the central inlet chamber is a rack creen with vertical bars which servces to deflect the heavier solids into the sludge pocket, produce a thorough mixture of the inflowing sewage and equalize the flow through the tank, thereby preventing short circuiting. Another rack creen is placed on the outer circumference of the main protion of the tank.

4.7 Calculation of the Capacity and Size of the Primary Settlement Tank

Having considered the various types of continuous flow tanks, the horizontal flow type is considered most suitable for this work both for primary settlement, aeration and final settlement tanks. The primary settlement tank, capacity exceeds the others by volume of the suspended matters which it settles. The capacity of the tank is a function of the volume of sewage, the rate of flow and the detention time required.

The volume of the sewage depends on the total water demand of the factory. It can be safely estimated that sewage is 60% of the water demand in the dairy plant.

= 0.6 x 246962 litre per day

= $\frac{0.6 \times 246962}{24}$ Litres per hour

= 6174 litres per hour

Detention time = 2 hours

∴ Required volume of the primary settle tank = 12348 litres (12.35 m^3)

Average Depth = 3m

∴ Area of tank = 4.116 m^2

Length of the tank is to times the width

i.e. L = 10W

where L = Length of tank

W = Width of tank

Now Area = L x W = 4.116 m^2

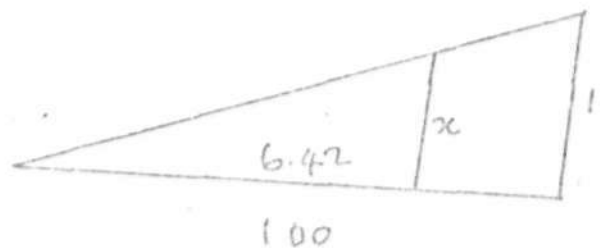
hence $L \times \frac{1}{10}L = 4.116$

or $L = \sqrt{4.116 \times 10}$

= 6.42 m

and W = 0.642 m

A gentle slope of 1 in 100 is applied



Hence Depth at inlet = $3 - \frac{0.0642}{2}$ = 2.968m

and Depth at outlet = $3 + \frac{0.0642}{2}$ = 3.032m

4.8 Calculation of the Capacity of the Aeration and the final Settlement Tanks

Assuming the crude sewage contains 40 parts per 100,000 of suspended matter and that 60% are removed in the primary settlement tank.

Volume of suspended matter removed in the primary tank

= 0.7 x 0.0004 x Volume of primary settling tank

= 0.7 x 0.0004 x 12.348 m^3

= 0.00346 m^3 = 3.46 litres

For all practical purposes this is a small quantity and therefore can be neglected. Hence volume of all the three tanks are constructed in equal volume.

4.9 Specification of the Sewage Recycling Pump

The function of the recycling pump is to pump away any effluent from the final settling tank that still contains more suspended matter than is permissible back to the primary settling tank for further treatment Usually less than 20% of the effluent is recycled. Hence required flow rate that the pump must handle is given by

$$Q = 0.2 \times 6174 \text{ litres/hour}$$

$$= \frac{0.2 \times 6174}{60} = 20.5 \text{ litre/min}$$

CHAPTER FIVE

DESIGN OF THE REFRIGERATION SYSTEM

5.1 Refrigeration Background

Refrigeration is the achievement of a temperation below that of the immediate surroundings. The mainten ma$_1$ntenance of low temperature requires the removal of heat from the product at low temperature and the discharge of this heat to a higher temperature.

The heat which a coolant requires to change phase from solid to liquid, liquid to vapour, or solid to vapour (sublimation) is what is used for refrigeration. Changing phase from solid to liquid is melting and the heat absorbed by the coolant is the latent heat of fusion. The most prominent example of this process is the melting of ice. The disadvantage is the inability of ice to cool a product below $C^{o}\alpha$. One technique used to overcome the temperature limitation is to use a salt solution which freezes and melts at a temperature lower than pure water.

The change of phase from liquid to vapour is vapourization and the heat absorbed by the coolant or by the product directly is the latent heat of varpourization. This process of refrigeration is the most widely used method because it is the

one that occurs in the popular vapour-compression cycle.

The basic equation which expresses the amount of heat transfered in a change of phase is:

$$Q = mL$$

where Q = amount of heat transfered (KJ)

M = mass of coolant (kg) or kg/hour

L = change in enthalpy KJ/kg

Refrigeration is essential in the Dairy industry both for cold treatment and storage of the products and for making the chilled water required in the actual production processes and for air conditioning purposes.

In making the Ice cream for instance, the ingredients (fresh milk) are first pasteurized and thoroughly mixed. Then refrigeration equipment cools the mix to 0 to 4^{o}c whereupon it enters a freezer. The freezer drops the temperature to -7 to -5^{o}c at whach condition the mix stiffens but remains fluid enough to flow into a container. The containers are then moved into the freezing room which is maintained between -24^{o}c and -29^{o}c where the ice cream is hardened. From this point on, until used, Ice cream must be stored below freezing temperatures.

The design of the refrigeration system in the factory, depends upon the production capacity, the processes in the factory, the desired temperatures, the initial temperature of the ingredients as well as the prevailing atmospheric condition.

5.2 Estimation of the Refrigeration Capacity

As already stated in the previous sections of this work, the production capacity is 18,000 litres per day while the temperature ranges are as follows:

Product	Initial Temperature	Process Temperature	Final Temperature	Storage Temperature
Ice Cream	Room temperature	90°c	-28°c	-4 -0°c
Milk	Room temperature	150°c	0-4°c	-4°c
Yoghurt	Room temperature	90°c	0-4'c	0-4'c

Table 16: Temperature Range

Hence the temperature of the products ranges from -28°c to 150°c.

The Refrigeration Capacity, Q = MS $(t_f - t_i)$ in kilo calorie/hour.

where m = mass of the product
s = specific heat or entlalpy of the product
t_f = final temperature
t_p = process temperature

Table 19: Halocarbon Refrigerants

Numerical Designation	Chemical Name	Chemical Formula
Freon 11	Trichloromonofluoromethane	ccl_3 F
Freon 12	Dichlorodifluoromethane	ccl_2 F_2
Freon 13	Monochlorotrifluoromethane	ccl f_3
Freon 22	Monochlorodifluoromethane	CHcl F_2
Freon 30	Methylene chloride	CH_3 CL_2
Freon 40	Methyl chloride	CH_3CL
Freon 113	Trichlorotrifluoroethane	CCl_2 FCCL F_2
Freon 114	Dichlorotetrafluoroethane	$CClF_2CCL$ F_2

Table 20: In-Organic Refrigerants

Numerical Designation	Chemical Name	Chemical Formula
717	Ammonia	NH_3
718	Water	H_2O
729	Air	
744	Carbon dioxide	CO_2
764	Sulfur dioxide	SO_2

Now the obvious means of cooling a product is to bring it in contact with a colder substance or the coolant. Milk is cooled by the rise in temperature of chilled water and ice cream is freezed by being in contact with the evaporator of a refrigeration system in which Ammonia liquid is varpourizing to absorb its latent heat of vapourization from the ice cream.

The quantity of heat Q in kilo joule (KJ) removed by the coolant during a constant-pressure or steady-flow process is given by the equation:

$$Q = MC_p (t_f - t_i)$$

where m = mass flow rate kg/hour

c_p = specific heat at constant pressure KJ/kg/^oC

$t_f - t_i$ = rise in temperature of coolant, ^oC

For optimal design eqn 5.1 = equation 5.2

that is:

$$Ms (t_f - t_p) = 44.3 \text{ KJ/sec} = m'cp (t_f - t_i)$$

m' = mass of refrigerant/sec

cp = specific heat of refrigerant = 0.701

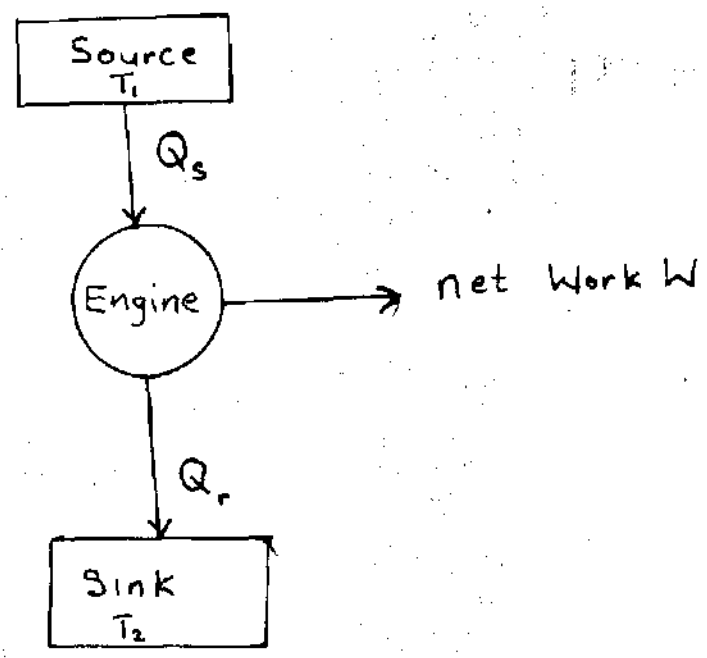
$t_f - t_i$ = change in temperature of refrigerant

$= 30-(-28) = 58^oc$

$$\therefore \quad m' = \frac{44.3}{0.7 \times 58} = 1.09 \text{ kg/sec}$$

5.3 Refrigeration Cycle

The refrigeration cycle can be viewed as a reversed Carnot heat engine cycle. Sadi Carnot proposed a heat engine cycle composed of reversible processes which gave the maximum possible work when working between two fixed temperature limits, T_1 and T_2.



That is $Q = ms(t_f - t_p)$

From table 17 the specific heat of milk is 0.935 and for ice cream it is 0.78. Latent heat of fusion of water at atmospheric temperature is 2107.4 KJ/kg. and mass of the product is:

18000 kg

$t_f = 14^{o}c$

$t_p = 28^{o}c$

∴ Cooling load, Q = heat absorbed from the product to cool it from $150^{o}c$ to $0^{o}c$ plus the latent heat of fusion and the heat absorbed from the product in order to cool it further to $-28^{o}c$

i.e. $Q = 1800 \times 0.935\ (150 - 0) + 18000 \times 2107.4$

$+ 18000 \times 0.78\ (0-(-28)$ KJ

$= 252450 + 3793320 + 39312 = 3822496$ KJ

That is the Refrigeration capacity Q

= 3827496 KJ per day

= 159479 KJ per hour

= 2658.0 KJ per minute

= 44.3 KJ per second

= 44.3 KW

= 14 tons

Table 17: Specific Heats of Some Substances

Material	Temperature °C	Specific Heat KJ/kg(°C)
Air	50	0.31
Ammonia (Anhydrons)	0	1.27
Brine (Calcium Chloride)	0	0.91
Glass	32-70	0.21
Ice	0	0.64
Ice Cream Mix (12% Fat)	0	0.75
Milk	0-80	0.935
Steel	32	0.15
Water	32	1.29
Apples	0	1.03
Eggs	0	1.16
Tomatoes	0	1.16

SOURCE: American Society of Refrigerating Engineers, (1957) "Airconditioning Refrigerating Data Book Applications Volume".

From second law of thermodynamics, heat energy flows from a hot to a cold body

$$T_1 \nearrow T_2$$

such that whenever a net heat $Q_n = Q_s$ Qr flow into the engine, a net work Wn is done by the engine. The first low of thermodynamics requires that Wn = Q_n = $Q_s - Q_R$

where Q_s = heat supplied to the engine

Q_R = heat rejected by the engine

The efficiency of the engine, when measured as its effectiveness in transforming the heat energy supplied into work energy is given by

$$\text{Efficiency} = \frac{\text{Net Work done}}{\text{Heat Energy Supplied}}$$

$$= \frac{Wn}{Q_s}$$

$$= \frac{Q_s - Q_r}{Q_s}$$

$$= 1 - \frac{Q_r}{Q_s}$$

This means that efficiency can only be unity or 100% if zero heat is rejected, that is if Q_r is zero. Since in practice Q_r must have a definite value, then the efficiency must be less than 100%. This means that it is not possible to convert all the heat energy to work energy. A heat engine cycle as illustrated in fig 27 can represent for instance a steam turbine plant in which case the source is the combustion chamber of the boiler and the sink is the cooling water in the condenser. Since the net work Wn is never equal to Qs there is always some heat energy rejected to the sink. It is also known from experience that there will never be any work done unless there is a temperature difference between the source and the sink. In other words "wherever a temperature difference exists, motive power can be produced".

A refrigeration cycle, as a reversed carnot heat engine cycle is shown in fig 28. Here heat Q_1 is extracted from the cold chamber and heat Q_2 is rejected into the warmer surroundings. W is the net work done.

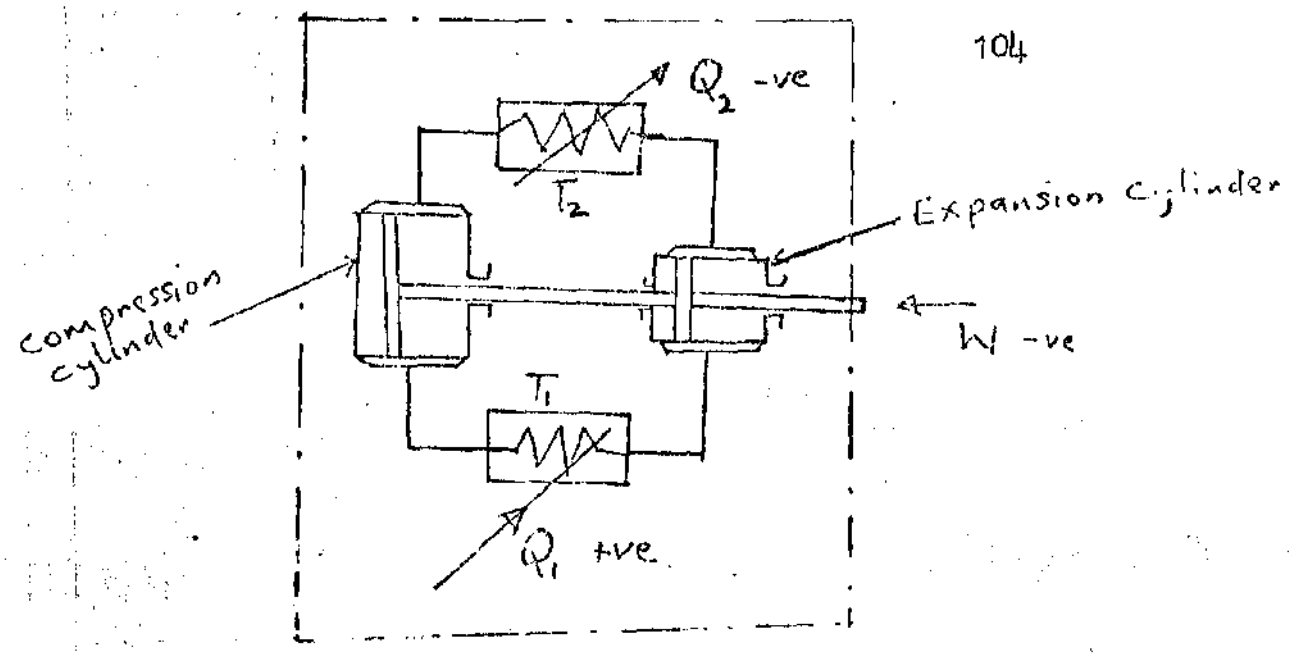


Fig 28: Refregeration Cycle

The effectiveness of the refrigerator is measured by:

Coefficient of Performance (C.O.P.)

$$= \frac{\text{Heat Extracted } Q}{\text{Work Done } W}$$

Now W is a function of $Q_2 - Q_1$ and infact from the earlier discussion on efficiency of the heat engine, $W = Q_2 - Q_1$

$$\text{Hence C.O.P.} = \frac{Q_1}{Q_2 - Q_1} = \frac{T_1}{T_2 - T_1}$$

This cycle is the popular vapour compression cycle. Essentially in the vapour compression cycle a third (the refrigerant) alternately evaporates and condenses, with one of the intervening processes being a compression of the vapour. The diagram of

the equipment set up and the temperature entropy diagram of the refrigeration cycle are shown in figs 29 and 30.

Fig 29: Equipment Set up

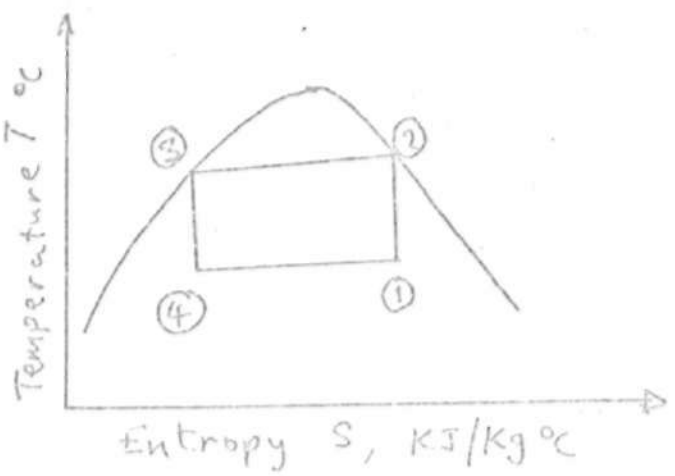


Fig 30: Temperature Entropy diagram.

The processes which comprise the cycle are:

1-2 Adiabatic compression

2-3 Isothermal rejection of heat

3-4 Adiabatic expansion

4-1, Isothermal addition of heat.

All the processes in the carnot cycle are thermodynamic lly reversible. Processes 1-2 and 3-4 are, consequently, isentropic. The withdrawal of heat from the low temperature source in process 4-1 is the refrigeration operation and is the entire purpose of the cycle.

5.4 Component Specifications

The specification of the components of the refrigeration system depend on the cooling load required which as stated before depends on:

1. The production capacity of the factory.
2. The temperature range through which the products must be received, processed and stored.
3. The amount of chilled water required.
4. The size of the cold room and the ice cream freezing room.
5. The thermodynamic properties of the refrigerant used.

With the aid of the pressure-enthalply diagram,

figs 30 and 31 all the significant quantities necessary for sizing the components can be determined. These quantities are:

1. The work of compression.
2. The heat rejection.
3. The refrigerating effect.
4. The coefficient of performance
5. The volume rate of flow per ton and
6. The power per ton.

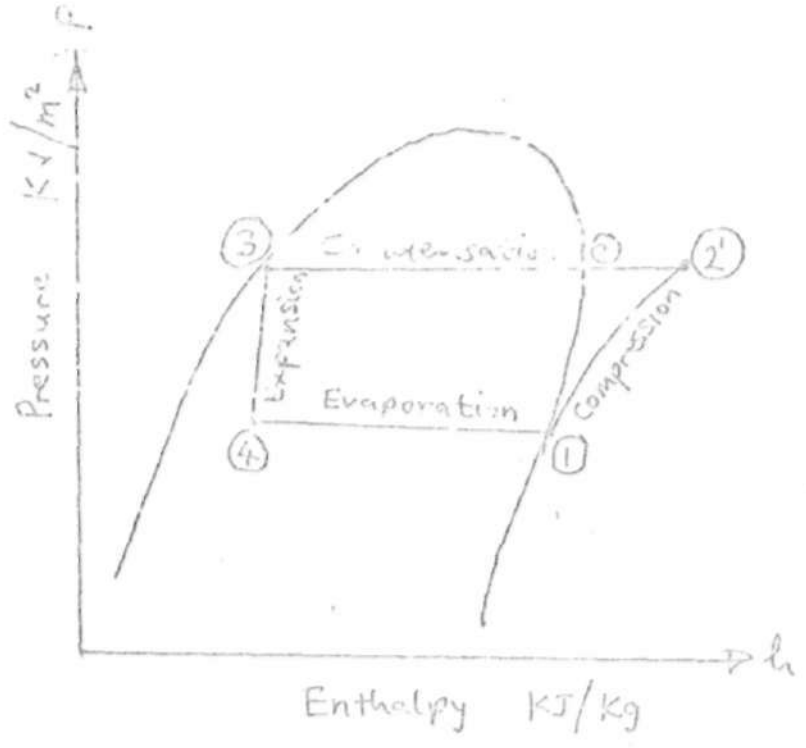
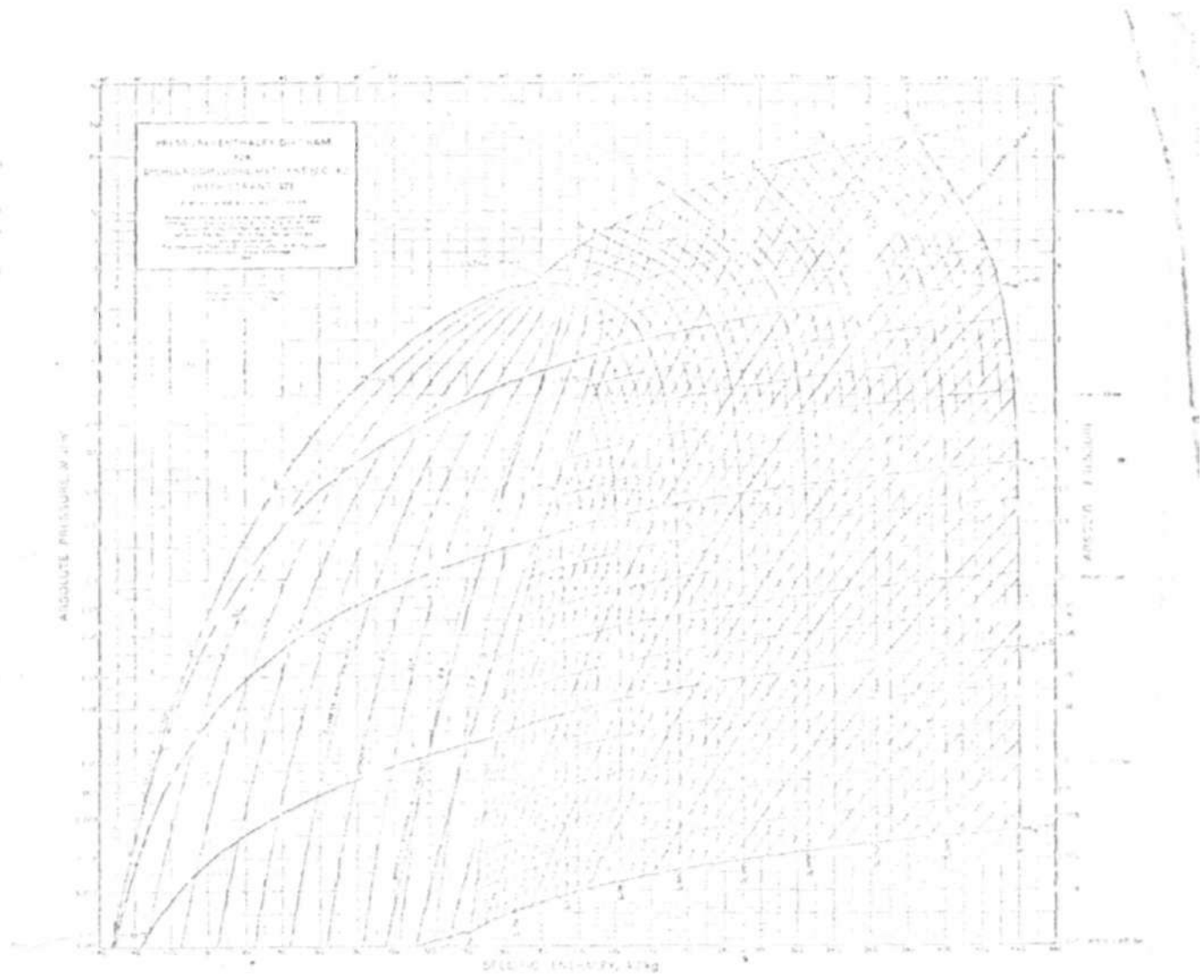


Fig 31: Pressure-Enthalpy Diagram



For this chart:
$h = 0$ for saturated liquid at -40 C
$s = 0$ for saturated liquid at -40 C

To convert to some zero state as tables 5 and 6
$h_{new} = h_{chart} + 365.40$ kJ/kg
$s_{new} = s_{chart} + 4.0760$ kJ/kgK

Reprinted with permission from *Thermodynamic Tables in SI (Metric) Units* by R. W. Haywood. Copyright © 1968 by Cambridge University Press, Cambridge, England.

TABLE 18:

Properties of Freon-12 (Refrigerant 12)

T(°C) / Press	294.2 x 10^3 N/m^2 (-1.5 C)				490.3 x 10^3 N/m^2 (15 C)				686.5 x 10^3 N/m^2 (32 C)				980.7 x 10^3 N/m^2 (52 C)				1.373 x 10^4 N/m^2 (55.5 C)				Temp
Temp C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	C
Sat liq	0.7551-3	117.049	117.298	0.2518	0.7634-3	132.582	132.946	0.2839	0.7744-3	148.137	148.627	0.3159	0.7893-3	169.330	169.964	0.3555	0.8029-3	179.458	180.227	0.3784	Sat liq
Evap	0.05944	339.107	355.161	0.8722	0.03579	329.813	347.360	0.8315	0.02553	320.006	340.131	0.7970	0.01758	308.189	330.340	0.7562	0.01520	299.785	320.596	0.7317	Evap
Sat vap	0.05938	353.345	372.359	0.9594	0.03553	342.394	360.306	0.8984	0.02530	332.154	353.125	0.8649	0.01835	322.304	343.104	0.8297	0.01584	313.230	333.123	0.8052	Sat vap
0	0.7496	376.23	373.76	0.9573																	0
5	0.7610	381.02	378.77	0.9693																	5
10	0.7624	381.47	379.83	0.9759																	10
15	0.7638	384.12	382.89	0.9793																	15
20	0.7651	386.83	385.98	0.9902	0.0377	384.98	383.47	0.9749													20
25	0.7664	389.56	389.08	0.9916	0.0385	387.77	386.65	0.9869													25
30	0.7677	392.30	392.22	0.9211	0.0393	390.57	389.84	0.9989	0.0158	380.94	387.04	0.9552									30
35	0.7690	395.10	395.40	0.9726	0.0401	393.36	393.02	0.9710	0.0137	382.47	383.75	0.9639	0.01189	380.67	380.16	0.9309					35
40	0.0703	397.90	398.59	0.9840	0.0409	396.22	396.28	0.9830	0.0126	386.66	388.85	0.9772	0.01089	383.45	385.65	0.9513	0.01315	395.23	398.34	0.9744	40
45	0.0716	400.70	401.77	0.9951	0.0417	399.10	399.55	0.9950	0.0136	389.85	392.05	0.9892	0.01007	386.22	389.17	0.9713	0.01238	382.29	401.94	0.9579	45
50	0.0730	403.50	404.95	0.0061	0.0425	401.98	402.82	0.0070	0.0127	393.03	395.24	0.0011	0.00935	389.00	392.65	0.9913	0.01169	386.35	405.58	0.9688	50
55	0.0743	406.29	408.12	0.0171	0.0433	404.85	406.08	0.0189	0.0119	396.21	398.42	0.0129	0.00869	391.78	396.13	0.0112					55
60	0.0756	409.08	411.52	0.0282	0.0441	407.72	409.31	0.0307	0.0111	399.38	401.59	0.0246	0.00810	394.55	399.61	0.0310					60
65	0.0768	411.86	414.93	0.0392	0.0449	410.58	412.56	0.0424	0.0104	402.54	404.76	0.0362	0.00757	397.31	403.09	0.0507					65
70	0.0780	414.63	418.34	0.0501	0.0457	413.43	415.80	0.0540	0.0098	405.70	407.93	0.0477	0.00710	400.06	406.57	0.0703	0.01023	400.63	410.23	0.0823	70
75	0.0793	417.39	421.75	0.0609	0.0465	416.27	419.04	0.0655	0.0093	408.85	411.09	0.0591	0.00668	402.80	410.04	0.0898	0.00961	403.40	413.98	0.0938	75
80	0.0806	420.14	425.16	0.0716	0.0473	419.10	422.27	0.0769	0.0088	411.99	414.24	0.0704	0.00630	405.53	413.50	0.1092	0.00902	406.16	417.72	0.1052	80
85	0.0819	422.88	428.56	0.0822	0.0481	421.92	425.49	0.0882	0.0083	415.12	417.38	0.0816	0.00594	408.25	416.96	0.1285	0.00847	408.91	421.45	0.1165	85
90	0.0831	425.61	431.94	0.0927	0.0489	424.73	428.70	0.0994	0.0079	418.24	420.52	0.0927	0.00561	410.96	420.42	0.1477	0.00795	411.65	425.17	0.1277	90
95	0.0843	428.33	435.31	0.1031	0.0497	427.53	431.90	0.1105	0.0075	421.35	423.65	0.1037	0.00531	413.66	423.87	0.1668	0.00746	414.38	428.88	0.1388	95
100	0.0855	431.04	438.67	0.1134	0.0505	430.32	435.10	0.1215	0.0071	424.45	426.77	0.1146	0.00503	416.35	427.31	0.1858	0.00700	417.10	432.58	0.1498	100
105	0.0868	433.74	442.02	0.1236	0.0513	433.10	438.28	0.1324	0.0068	427.54	429.88	0.1254	0.00477	419.03	430.74	0.2047	0.00657	419.81	436.27	0.1607	105
110	0.0880	436.43	445.36	0.1337	0.0520	435.87	441.45	0.1432	0.0064	430.62	432.98	0.1360	0.00453	421.70	434.16	0.2235	0.00617	422.51	439.94	0.1715	110
115	0.0892	439.11	448.69	0.1437	0.0528	438.63	444.61	0.1539	0.0061	433.69	436.07	0.1465	0.00431	424.36	437.57	0.2422	0.00580	425.20	443.60	0.1822	115
120	0.0904	441.78	452.01	0.1536	0.0535	441.38	447.76	0.1645	0.0058	436.75	439.15	0.1569	0.00411	427.01	440.97	0.2608	0.00546	427.88	447.24	0.1928	120
125	0.0916	444.44	455.32	0.1634	0.0543	444.12	450.90	0.1750	0.0055	439.80	442.22	0.1672	0.00392	429.65	444.36	0.2793	0.00514	430.56	450.87	0.2033	125
130	0.0928	447.09	458.62	0.1731	0.0550	446.86	454.03	0.1854	0.0052	442.84	445.28	0.1774	0.00375	432.28	447.74	0.2977	0.00484	433.23	454.49	0.2137	130
135	0.0940	449.73	461.91	0.1827	0.0558	449.59	457.15	0.1957	0.0050	445.87	448.34	0.1875	0.00359	434.90	451.11	0.3160	0.00456	435.89	458.10	0.2240	135
140	0.0952	452.36	465.19	0.1922	0.0565	452.31	460.26	0.2059	0.0048	448.89	451.39	0.1975	0.00344	437.51	454.47	0.3341	0.00430	438.54	461.70	0.2342	140
145	0.0963	454.98	468.46	0.2016	0.0573	455.02	463.36	0.2160	0.0046	451.90	454.43	0.2074	0.00331	440.11	457.82	0.3521	0.00406	441.18	465.30	0.2443	145
150					0.0581	457.72	466.45	0.2260	0.0044	454.90	457.46	0.2172	0.00318	442.70	461.16	0.3700	0.00384	443.81	468.89	0.2543	150
155					0.0589	460.41	469.53	0.2359	0.0042	457.89	460.48	0.2269	0.00307	445.28	464.49	0.3878	0.00364	446.43	472.47	0.2642	155
160					0.0597	463.09	472.60	0.2457	0.0041	460.87	463.49	0.2365	0.00297	447.85	467.81	0.4055	0.00346	449.04	476.04	0.2740	160
165					0.0605	465.76	475.66	0.2554	0.0039	463.84	466.50	0.2460	0.00287	450.41	471.13	0.4231	0.00329	451.64	479.60	0.2837	165
170					0.0613	468.42	478.71	0.2650	0.0038	466.80	469.51	0.2555	0.00278	452.96	474.45	0.4406	0.00313	454.24	483.16	0.2933	170
175													0.00270	455.50	477.76	0.4581	0.00298	456.84	486.72	0.3028	175
180													0.00262	458.03	481.07	0.4755	0.00284	459.44	490.28	0.3123	180
185													0.00255	460.56	484.38	0.4929	0.00271	462.04	493.84	0.3217	185
190																	0.00259	464.64	497.40	0.3311	190
195																	0.00247	467.24	500.97	0.3404	195
200																	0.00236	469.84	504.54	0.3497	200

Table 19: Properties of Freon-12 (Refrigerant 12)



Reprinted with permission from *Handbook of Thermodynamic Tables and Charts*, by K. Raznjevic. Copyright © 1976 by Hemisphere Publishing Corporation, Washington D.C.

Table 20: *Properties of Saturated Vapor 12 (Refrigerant 12)*

t	T	P	v_{sat}	v_{sup}	h_{sat}	h_{sup}	h_{sat}	h_{sup}	h_{sat}	s_{sat}	s_{sup}
C	K	cm^2	cm^3/g	cm^3/g	J/g	J/g	J/g	J/g	J/g	J/g K	J/g K
-30	243.15	1.51	0.6	1.1	30.1	60.1	330.9	375.4	345.0	1.5	1.7
-25	248.15	1.81	0.5	0.9	31.4	60.9	329.9	374.9	343.5	1.4	1.6
-20	253.15	2.16	0.4	0.8	32.8	61.6	328.9	374.4	342.1	1.4	1.6
-15	258.15	2.56	0.4	0.7	34.1	62.4	327.9	373.9	340.7	1.3	1.5
-10	263.15	3.02	0.3	0.6	35.5	63.1	326.8	373.3	339.3	1.3	1.5
-5	268.15	3.55	0.3	0.5	36.8	63.9	325.8	372.8	337.9	1.3	1.4
0	273.15	4.16	0.2	0.5	38.2	64.6	324.7	372.2	336.5	1.2	1.4
5	278.15	4.85	0.2	0.4	39.6	65.4	323.6	371.6	335.1	1.2	1.4
10	283.15	5.63	0.2	0.4	41.0	66.2	322.5	371.0	333.7	1.2	1.3
15	288.15	6.52	0.2	0.4	42.4	66.9	321.4	370.4	332.3	1.2	1.3
20	293.15	7.52	0.2	0.3	43.8	67.7	320.3	369.8	330.9	1.1	1.3
25	298.15	8.64	0.2	0.3	45.2	68.4	319.1	369.1	329.5	1.1	1.3
30	303.15	9.89	0.1	0.3	46.6	69.2	317.9	368.5	328.1	1.1	1.2
35	308.15	11.28	0.1	0.3	48.0	69.9	316.8	367.8	326.7	1.1	1.2
40	313.15	12.82	0.1	0.2	49.4	70.7	315.6	367.1	325.3	1.1	1.2
45	318.15	14.52	0.1	0.2	50.8	71.4	314.4	366.4	323.9	1.0	1.2
50	323.15	16.39	0.1	0.2	52.2	72.2	313.2	365.7	322.5	1.0	1.2
55	328.15	18.44	0.1	0.2	53.6	72.9	312.0	365.0	321.1	1.0	1.1
60	333.15	20.68	0.1	0.2	55.0	73.7	310.8	364.3	319.7	1.0	1.1
65	338.15	23.12	0.1	0.2	56.4	74.4	309.6	363.6	318.3	1.0	1.1
70	343.15	25.78	0.1	0.2	57.8	75.2	308.4	362.8	316.9	1.0	1.1
75	348.15	28.67	0.1	0.2	59.2	75.9	307.2	362.1	315.5	1.0	1.1
80	353.15	31.80	0.1	0.2	60.6	76.7	305.9	361.4	314.1	1.0	1.1
85	358.15	35.19	0.1	0.2	62.0	77.4	304.7	360.6	312.7	1.0	1.1
90	363.15	38.85	0.1	0.2	63.4	78.2	303.4	359.9	311.3	1.0	1.1
95	368.15	42.80	0.1	0.2	64.8	78.9	302.1	359.1	309.9	1.0	1.1
100	373.15	47.05	0.1	0.2	66.2	79.7	300.8	358.4	308.5	1.0	1.1
105	378.15	51.62	0.1	0.2	67.6	80.4	299.5	357.6	307.1	1.0	1.1
110	383.15	56.53	0.1	0.2	69.0	81.2	298.2	356.8	305.7	1.0	1.1
112.5 (crit.)	385.65	59.15	0.2	0.2	69.7	81.6	297.5	356.4	305.0	1.0	1.1

Reprinted with permission from *Handbook of Tropical Refrigeration Tables and Charts*, by K. Raznjević. Copyright © 1978 by Hemisphere Publishing Corporation, Washington, D.C.

Already the refrigeration effect (or refrigeration capacity/required has been determined.

Q = 44.3 KW (14 tons)

Refrigerant = · Freon - 12 (CCL_2 F_2)

Dichlorodifluoromethane

Temperature limits:

Condesning temperature of the refrigerant

T_2 = 20°c

Evaporation temperature (or temperature of the cold chamber)

T_1 = -28°c

The work of compression in KJ/kg of refrigerant is equal to the change in:

1. Enthalpy in process 1-2 of fig 31

or $W_s = h_2 - h_1$

where W_c = Work of compression

h_2, h_1 = Enthalpy of the refrigerant at condensing and evaporation temperature respectively.

From the properties of the saturated refrigerants tables and charts.

h_1 = hg at - 18°c (evaporation temperature

= 174.8 KJ/kg

h_2 = hg at 20°c (Condensation temperature)

= 273.3 KJ/kg

h_4 = 69.6 KJ/kg

RE = Refrigeration effect = $h_1 - h_4$

Wc = Work of compression = $h_2 - h_1$

= 273.3 - 174.8 = 98.5 KJ/kg

2. Weight of refrigerant circulated per ton

$$W_R = \frac{3.16}{RE} \frac{\text{tons}}{KW} = \frac{3.16}{174.8-69.6} = 0.03 \text{ kg/sec per ton}$$

3. Referring to table 5.3, Density of vapour at -18°c

= 6.25 kg/m^3

∴ Total volume of vapour leaving evaporation

$$= \frac{\text{Mass}}{\text{Density}} = \frac{W_R}{6.25} = \frac{0.03}{6.25} = 0.005 \text{ } m^3/\text{sec per ton}$$

∴ For the rated 44.3 KW (14.02 tons)

Mass flow rate = 0.005 x 6.25 x 14.02

= 0.43 kg/sec

Volume flow rate= 0.005 x 14.02

= 0.0701 m^3/sec

Since the plant is a 44.3 KW refrigerator, the size of the compressor is such that can handle 0.005 m^3/sec per ton.

i.e. Capacity of compressor

$= 0.005 \times 14.2$

$= 0.07\ m^3/min$

To specify the capacity of the condenser, there is need to compute the amount of heat q to be dissipated by the condenser.

Theoretically, the heat dissipated in the condenser q_c is the total of the heat pumped from the evaporator and the heat of compression.

$q_c = Wc + RE$

where q_c = heat dissipated by the condenser

Wc = heat of compression

RE = refrigeration effect

$Wc = h_2 = h_1 = 273.3 - 174.8 = 98.5\ KJ/kg$

$RE = h_1 - h_4 = 174.8 - 69.6 = 105.2\ KJ/kg$

$\therefore q_c = WC + RE$

$= 98.5 + 105.2 = 203.7\ KJ/kg$

$= 203.7 \frac{KJ}{kg} \times 0.43 \frac{kg}{sec}$

$= 87.59\ KJ/sec$

$= 87.6\ KW$

5.5 Summary of Specifications of Components

Compressor rating:	0.07 m^3/sec
Condenser rating:	87.59 KW
Evaporator rating:	0.43 kg/sec
Expansion valve:	0.43 kg/sec
Refrigerant flow rate:	0.43 kg/sec

CHAPTER SIX

HEATING, VENTILATION AND AIR CONDITIONING

6.1 Desired, Condition of Spaces

The condition of air in the different spaces in the dairy industry depend upon the function of the particular space. Essentially the functional spaces include:

1. Yoghurt Quality Control Room - a 3.5m x 6.25 m room maintained at 45°c and 65% RH.

2. Laboratories, offices and finished product stores - a total of 318.75 m^2 spaces maintained at 8-10°c and 75% RH.

3. Cold Room - a 50cm x 100cm space maintained at -4°c.

4. Ice cream freezing room - a 4m x 6m space maintained at -20°c.

5. Processing rooms, packaging rooms, passages, stores, workshops and spare parts rooms, boiler and cooling equipment rooms, - maintained at 20-25°c and 75-85% RH.

A total of 1400 m^3 space maintained at a temperature range of 20-25°c and 75% relative humidity. The air must be filtered of all dust.

6.2 Cooling Load Estimation

The cooling load is the sum of all the sensible and latent heat gains. The sources of heat gains are:

1. Direct radiation from the sun
2. Infilteration from windows, doors and roofs.
3. Heat from warm walls and ceilings
4. Sensible and latent heats from people, lights, motors, machines and materials.

The design conditions of air are as specified on section 6.7 and the ambient temperature of Kaduna is 30°c at 75% RH.

From the 1965 ASHRAE guide and Data book we have the following heat gains calculations:

Source	Sensible Heat KW	Latent Heat KW	Total Heat KW
Heat gains per occupant	0.2 KW	0.1 KW	10.3 KW
Heat gains from electric motors	-	-	0.9 KW
Heat gains from gas burners in the lab.	0.9 KW	0.60 KW	1.5 KW
Heat gains from pies per metre	-	-	0.03 KW
Heat gains from Infiltration (per square metre) of window or door	-	-	0.007 KW
Average Director Solar radiation and diffuse sky radiation and convection per square metre of window or door glass	-	-	0.05 KW

Table 21: Sources of Heat Gains

Source		Heat Gain KW
Number of occupants	= 65 @ 0.3 KW	19.5
Number of motors (2.7 KW each)	= 18 @ 0.9 KW/motor	16.2
Number of Burners	= 4 @ 1.5 KW/Burner	6
Total length of steam and hot water pipes	= 27m @ 0.03 KW/m	0.81
Total area of external windows and doors	= 14.4 m^2 @ 0.05 KW/m^2	0.72
	TOTAL	43.23

Table 22: Estimation of Cooling Load
Hence Cooling Load - Q = 43.23 KW

6.3 Determination of the Mass Flow Rate of Chilled Water (kg/hour) required to Maintain the Air at the Desired Conditions

Let m = mass flow rate of chilled water in kg/hour

Initial temperature of the chilled water = $4^{o}c$

Final temperature of the chilled water after cooling the air = $8^{o}c$.

The specific heat capacity of water at (--100^{o}c)

$$S = 1.289 \text{ KJ/kg}^{o}\text{c}$$

Now the chilled water in order to cool the air the desired condition must remove heat Q, equivalent to the cooling load, Q = 43.23 KW

The heat absorbed by the chilled water is given by:

$$Q = ms\ (t_{final} - t_{initial}) = mst$$

where Q = quantity of heat removed by the chilled water in KW

M = mass flow rate of the chilled water in kg/hour

t = $(t_{final} - t_{initial})$ = difference between the initial temperature of the chilled water, and its temperature after removing quantity of heat

Q = 43.3 KW

i.e. $t = t_{final} - t_{initial} = 8^{o}c - 4^{o}c$

$= 4^{o}c$

$\therefore \quad Q = m \times 1.289 \times 4 = 43.2$ KW

$\therefore \quad m = \frac{43.2}{1.289 \times 4} = 8.3$ kg/sec

i.e. Mass flow rate of chilled water

= 8.3 kg/sec or 498 kg/min

Hence the volume flow rate = 29880 litre/hour

6.4 Determination of Chilled Water Capacity and the Chilled Water Pump Capacity

6.4.1 Tank Capacity

The chilled water tank should hold sufficient quantity of chilled water to last for at least 12 hours.

Hence volume of the tank

$$V = 29880 \ \frac{\text{litres}}{\text{hour}} \times 12 \text{ hours}$$

$$= 358560 \text{ litres or } (3586 \ m^3)$$

For convenience of observation and routine servicing of the evaporator and the ceiling height of the building, the depth of the tank should not exceed (H = 4m).

$\therefore$ Cross sectional Area of the tank

$$A = L \times B = \frac{V}{H} = \frac{358.6}{4} = 89.5 \ m^2$$

A = Area of cross section of the tank (m^2)

where L = Length of tank in m

B = Breadth of tank in m

H = Height or depth in m

V = Volume of tank in m^3

For a sqaure cross section

$$L = B = \sqrt{9.46} = \underline{9.46} \text{ m}$$

6.4.2 Capacity of the chilled Water Pump Pc

The capacity of the pump is function dependent. Hence the pump is required to pump 8.3 litre of chilled water per sec and therefore capacity of pump is Pc = 8.3 litre/sec

∴ centrifugal pump of capacity 8.3 litre/sec is recommended.

6.5 Design of Stirrer for the Chilled Water Tank

In order to maintain uniform temperature in the chilled water tank, and to avoid formation of ice in the upper or any section of the tank, it is necessary that the water is continuously being stirred in such a manner that each molecule of water in contact with the evaporator of the refrigeration system moves away to the opposite

end of the tank and back again to the evaporator in every one hour. Therefore the stirrer must be capable of causing 358560 kg of water to be displaced through the perimeter of the tank (perimeter of tank = 37.84m) . That is the power rating of the electric motor of the stirrer will be:

$$P_s = \frac{358560 \times 37.84}{60 \times 60} \text{ Watt} = 3.77 \text{ KW}$$

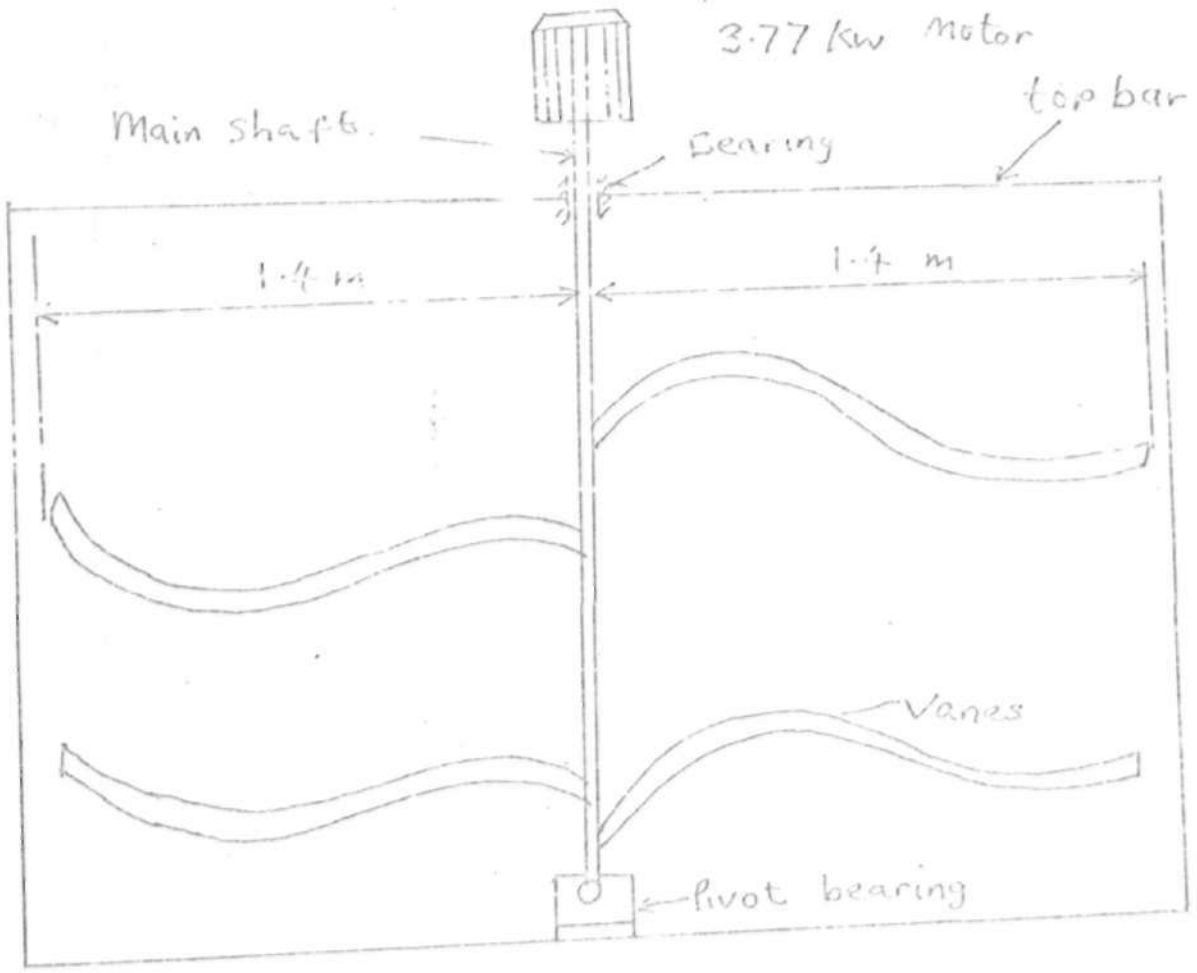


Fig 32: Chilled Water Tank Stirring System

6.6 Component Specifications of the Airconditioning System

The design out-door temperature in Kaduna is $30^{o}c$ and the minimum desired comfort airconditioned air temperature in the factory is $8^{o}c$.

Detailed estimation of the spaces, and therefore volume of air required per hour to maintain the desired conditions as given in section 6.1 and subsequently, the cooling load has been determined to be:

$Q = 43.2$ KW

The recommended refrigerant is freon 12 (CCL_2F_2) Dichlorodifluoromethane.

Now with the aid of the pressure-enthalpy diagram fig 31, all the significant quantities necessary to size the components of the airconditioning system can be determined. The quantities are:

1. the work of compression
2. the heat rejection
3. the cooling load Q
4. the coefficient of performance
5. the volume rate of flow per ton; and
6. the power per ton.

Already the cooling load ($Q = 43.2$ KW) is known. And condensing temperature of the refrigerant ($T_2 = 20^{o}c$)

is also known. It is desired to maintain the evaporator temperature at ($T_1 = 0^{\circ}c$) to effectively chill the water to the desired $4^{\circ}c$.

The work of compression in KJ/kg of refrigerant is equal to the change in enthalpy in process 1-2 of fig 31.

1. $W_c = h_2 - h_1$ = Work of compression

h_1 and h_2 = Enthalpy of the refrigerant at condensing and evaporation temperatures respectively. See fig 31.

From the properties of the saturated refrigerants, tables and charts (see table 18 and fig 31)

h_1 = h_g at $0^{\circ}c$ (evaporation temperature)

= 249.4 KJ/kg

h_2 = h_g at $20^{\circ}c$ (condensation temperature)

= 275.5 KJ/kg

$h_3 = h_4$ = 69.6 KJ/kg

$\therefore$ W_c = 275.5 - 249.4 = 26.1 KJ/kg

2. Refrigeration effect (RE) = $h_1 - h_4$

= 249.4 - 69.6 = 179.8 KJ/kg

Weight of refrigerant circulated per ton

$$W_R = \frac{3.16}{RE} = \frac{3.16}{179.8} = 0.02 \text{ kg/sec}$$

= 1.2 kg/min

Referring to the tables:

Density of the refrigerant vapour at $0^{\circ}c$

$= 11.8\ kg/m^3$

$\therefore$ Total volume of vapour leaving evaporator

$$= \frac{\text{Mass}}{\text{Density}} = \frac{W_R}{11.8} = \frac{0.02}{11.8}\ m^3/\text{sec per ton}$$

$$= 0.1\ m^3/\text{min per ton}$$

$\therefore$ for the rated capacity of 43.2×0.3165 tons

volume flow rate $= 0.1 \times 13.67$

$= 1.39\ m^3/min$

Mass flow rate $= 1.39 \times 11.8$

$\therefore$ Capacity of compressor $= 1.39\ m^3/min$

3. To specify the capacity of the condenser, there is need to compute the amount of heat q_c dissipated by the condenser. Theoretically, the heat dissipated in the condenser q_c is the total of the heat pumped from the evaporator and the heat of compression.

That is $q_c = W_c + RE$

where q_c = heat dissipated by the condenser

Wc = Work or (heat) of compression

RE = Refrigeration effect

$\therefore$ $q_c = 26.1 + 179.8 = 205\ KJ/kg$

$$= 205\ \frac{KJ}{kg} \times 16.4\ \frac{Kg}{min}$$

$= 3376.76\ KJ/min$

$= 56.28\ KW$

6.7 Summary of Specifications of Components

Compressor rating = 1.39 m^3/min

Condenser rating = 56.28 KW

Evaporator capacity= 16.42 kg/min

Expansion valve capacity = 16.42 kg/min

Refrigerant flow rate = 16.4 kg/min

Chilled water tank capacity = 8.3 litre/sec

Chilled water tank capacity = 358.6 m^3

Chilled water pump capacity = 8.3 litre/sec

Power rating of the electric motor driving stirrer of the chilled water tank = 3.77 KW

6.8 Distribution of Air

The most economical method to satisfy the different condition of air requirement of the various spaces is to centrally circulate filtered air at 20-25^{o}c and 65% RH to all parts of the factory and to heat or cool as the case may be at particular spaces. This is best achieved by the means of Ducts and fans. Hence the first phase of this design shall commence with the determination of the specifications of the Ducts and fans after the distribution pattern has been fixed. The air distribution pattern is worked out as shown in figure 33.

The air distribution pattern is worked out as shown in figure 33 below:

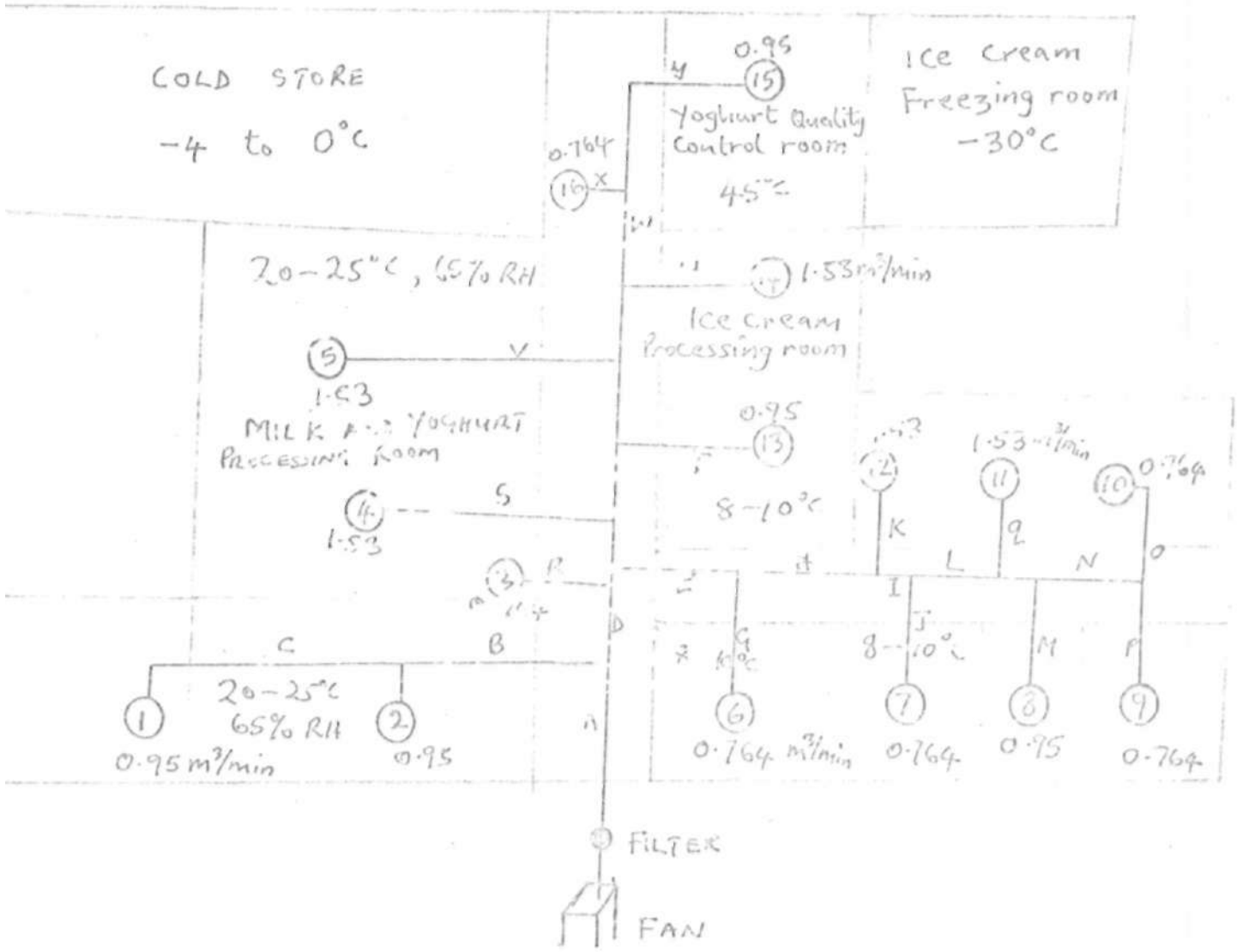


Fig 33: Distribution Pattern of air

Note that the cold store and the ice cream freezing room have been excluded from supply from this central distribution because they are maintained at sub-zero temperatures and therefore require heavy refrigeration: they are already covered in Chapter Five.

6.9 Designing Duct Work (Using Velocity Method)

Three methods are widely used in designing duct work:

1. Velocity
2. Equal friction
3. Static regain

The choice of design method depends almost entirely upon the size of the duct arrangement. Small duct systems are commonly designed by the velocity method. Large high pressure systems are most frequently designed by the static regain method. Duct arrangements between these two extremes are nearly always laid out by the equal friction. Sometimes a duct arrangement will be designed by a combination of two methods. For instance, the trunk duct will be laid out by the static regain method and the branch ducts system designed by the equal friction method.

6.9.1 Velocity Method

In the velocity method, a velocity is arbitrarily assigned to each section of the ductwork. The velocity is highest at the far outlet and is reduced as various branches are taken off the main duct. The velocity will be lowest at the end of the duct since the air quantity for each section of the duct is already known, assuming the velocity permits the area to be calculated. Taking the duct size of each section to the friction chart, along with the air quantity, the unit friction is read The term "unit friction" means the friction loss per (30m) of equivalent length. The total friction is then found by adding friction losses of individual sections making up the longest equivalent length.

To analyse the duct system, we write the air quantity on each section starting from the last ceiling Diffuser (14). Hence section y carries 0.95 m^3/min section u 1.53 m^e/min and section v 1.53 m^3/min. Section T must carry the sum of outlets (4), (13), and (14) or 3.99 m^3/min. Section D must carry sum of sections E and R or 14.5 m^3/min. And section A carries the sum of sections D, B and C is equal to 16.24 m^3/min. Table 23 is a guide for selecting appropriate velocities of air at various sections of the duct.

Designation	Recommended Velocities m/min
Outdoor air intakes	15.0
Filters	10.5
Heating coils	18.0
Cooling coils	18.0
Air washers	15.0
Fan outlets	48.0 - 72.0
Main Ducts	36.0 - 84.0
Branch Ducts	24.0 - 30.0
Branch risers	24.0

Table 22: Recommended Duct Velocities of the Air Conditioning System taken from 1965 ASHRAE Guide and Data Book

Hence a velocity of 78 m/min can be assumed for section A and then reduced to 72m/min for section D and 30 m/min for Section T and 24 m/min for Section Y.

6.9.2 Area of Sections

The area of any particular section can be calculated from:

$$Q = AV$$

where Q = Air quantity, m^3/min

A = Cross sectional area of duct in m^2

V = Velocity of air in m/min

Solving this equation for area

$$A = \frac{Q}{V}$$

For Section A,

$$A_A = \frac{16.42}{78} = 0.21 \ m^2$$

For Section D

$$A_D = \frac{14.5}{72} = 0.20 \ m^2$$

For Section T

$$A_T = \frac{3.99}{30} = 0.133 \ m^2$$

For Section Y

$$A_Y = \frac{0.962}{24} = 0.04 \ m^2$$

Hence the diameter of the sections can be determined using the formular $A = \frac{\Pi d^2}{4}$

$$\text{or} \quad d = \sqrt{\frac{4A}{\Pi}}$$

Therefore, for Section A

$$d = \sqrt{\frac{4 \times 0.21}{\pi}} = 0.52m$$

For Section D

$$d_D = \sqrt{\frac{4 \times 0.20}{\pi}} = 0.50m$$

For Section T

$$d_T = \sqrt{\frac{4 \times 0.133}{\pi}} = 0.41m$$

For Section Y

$$d_y = \sqrt{\frac{4 \times 0.04}{\pi}} = 0.23m$$

Section	Capacity m^3/min	Air Velocity m/min	Area of Section m^2	Diameter of the Seotion m	Air Delivery at the Ceiling Diffusers m^3/min
A	16.42	78	0.21	0.52m	-
D	14.5	72	0.20	0.50m	-
T	4.00	30	0.133	0.41m	-
Y	0.96	24	0.04	0.23m	0.95
	G,J,O, P,R,X,	-	-	-	0.764
	B,C,F, M,Y	-	-	-	0.95
	K,Q,S, U,V	-	-	-	1.52

Table 23: Summary of Duct Specifications (See fig 3.)

6.10 Equivalent Rectangular Duct

Generally ducts are of rectangular cross-section and it is not feasible to design the ducts in circular sections.

1965 ASRAE Guide and Data Book given equation for circular equivalent of a rectangular duct as:

$$d_e = \frac{1.30(ab)}{(a+b)\,0.25} = 1.30\sqrt{\frac{(ab)^5}{(a+b)^2}}$$

where a = length of one side of rectangular duct in inches (converted)

b = length of adjacent side of rectangular duct inches

d_c = circular equivalent of a rectangular duct for equal friction and capacity in inches.

The process is shorter if the Ductulator or the tables are used.

Hence for Section A

d_e = 0.52m = 1.30 inches

$\therefore$ = 130 = $1.30\sqrt{\frac{(ab)^5}{(a+b)^2}}$

for square section a = b

hence $d_c = 1.3 \sqrt[8]{\frac{q^{10}}{(2a)^2}} = 1.3 \sqrt[8]{\frac{a^8}{4}}$

$$= \frac{1.3\ a}{\sqrt{4}} = \frac{1.3\ a}{1.189}$$

$$\therefore \quad a = b = \frac{dc \times 1.189}{1.3} = \frac{dc}{0.915}$$

$\therefore$ when dc = 0.52m; dc = 0.50m; dc = 0.41m;
dc = 0.23m; a = 0.5685m; a = 0.5469m; 9 = 0
a = 0.4483m; a = 0.2515; if a = 0.75b

$$dc = 1.30 \sqrt[8]{\frac{(\frac{a^2}{0.75})^5}{a + \frac{a}{0.75})^2}}$$

$$= 1.30 \sqrt[8]{\frac{(\frac{a^2}{0.75})^2}{(\frac{1.75a}{0.75})^2}}$$

i.e. $$dc = 1.3 \sqrt[8]{\frac{a^{10} \times 4.2139917}{a^2 \times 5.4444}}$$

$$= 1.3 \sqrt[8]{0.773998\ a^8}$$

$$1.3 \times 0.968484\ a = 1.259\ a$$

$$\text{or } a = \frac{dc}{1.259}$$

Hence for dc = 0.52m

$$a = \frac{0.52}{1.259} = 0.413m$$

$$\text{and } b = \frac{a}{0.75} = \frac{0.413}{0.75} = 0.55m$$

For Section D

dc = 0.50

$$a = \frac{0.5}{1.259} = 0.397m$$

$$\text{and } b = \frac{0.397}{0.75} = 0.5295m$$

For Section T

dc = 0.41m

$$a = \frac{0.41}{1.259} = 0.3257$$

$$b = \frac{0.3257}{0.75}$$

For Section Y

dc = 0.23m

$$a = \frac{0.23}{1.259} = 0.1827m$$

$$b = \frac{0.1827}{0.75} = 0.2436m$$

6.11 Summary of Equivalent Rectangular Duct

Section	Equivalent Circular Duct Dimension (dc)m	Equivalent Square Duct Dimension (a) m	Equivalent Rectangular Duct Dimensions	
			Length (b) m	Width (a) m
A	0.52m	0.5685	0.55	0.413
D	0.50m	0.5467	0.5295	0.397
T	0.41m	0.4483	0.4342	0.3257
Y	0.23m	0.2515	0.2436	0.1827

6.12 Fan Size and Duct Layout

Since all the air in the system must pass through section A, and infact the sum total of the air flow in the different sections of the system equals the air flow rate at Section A, the size of the fan depends upon the air flow rate at Section A. Hence the fan must be such that can handle 580 cubic feet of air per minute.

6.13 Distributing Cooling and Heating Media

Each coil, washer, or heat exchanger in this system must have an arrangement of piping that will convey the cooling or heating fluid from it to the boiller or boiler and back again.

In this system water is used, d the arrangement takes the form of supply and return water piping along with the pumps to circulate the water through the piping and other equipment.

Water piping arrangements, for heating and cooling can be divided into four general classifications:

1. One-pie arrangement
2. Two-pipe arrangement
3. Three-pipe arrangement
4. Four-pipe arrangement

The first two are arranged so that either hot or cold water can be circulated through the piping. However, when hot water is being circulated in one-pipe and two-pipe arrangements, only hot water is available to each coild connected to that circuit. Conversely, when cold water is being circulated, only cold water is available to each coil.

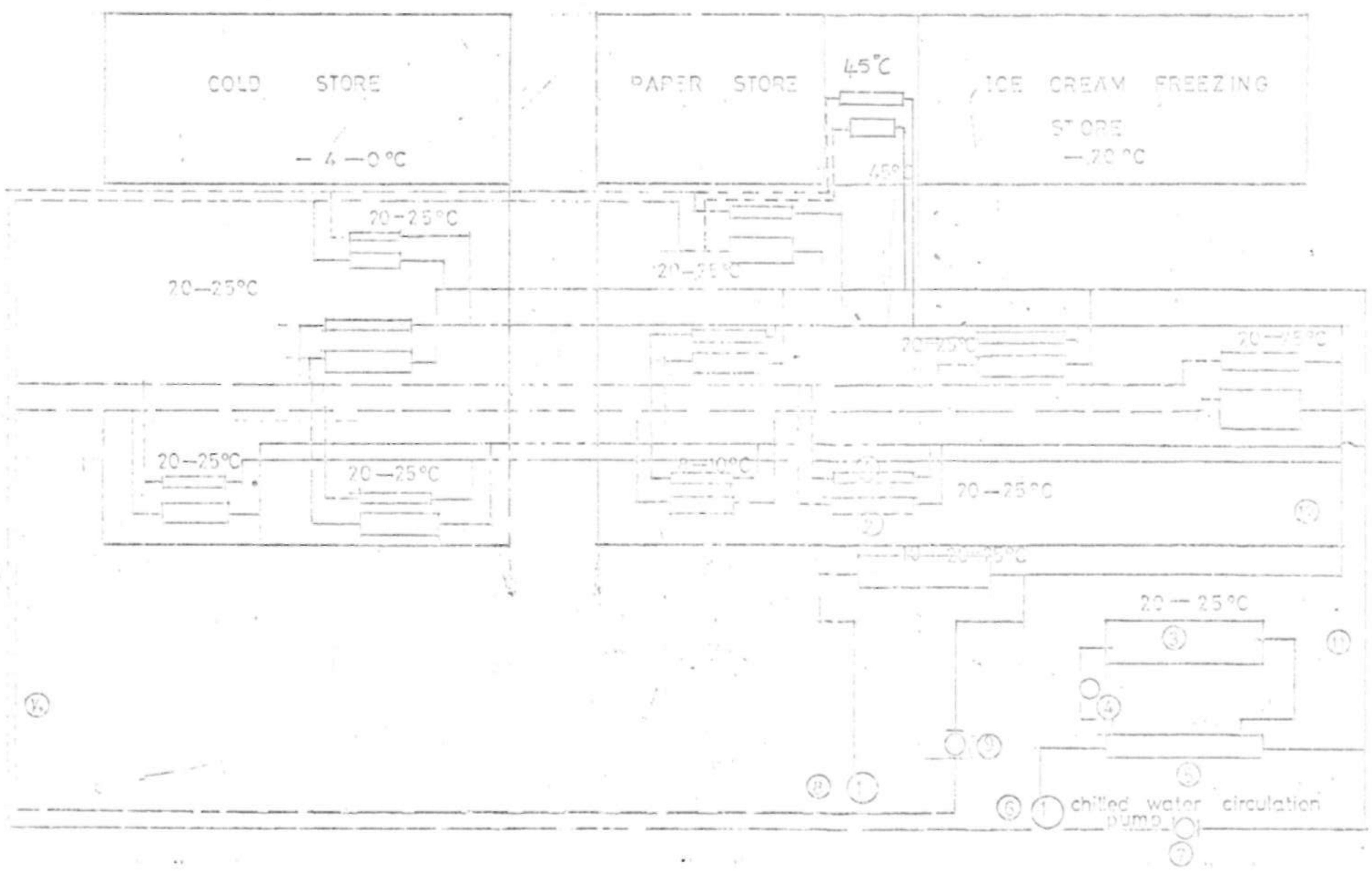
The three-pipe arrangement has two supply pipes and one return pipe connected to each coild. One supply

pipe carries hot water; the other supply pipe carries cold water. The return pipe carries a mixture of hot and cold water back to the chilling or heating equipment where it is diverted to the chiller or heater, depending on its temperature.

The four-pipe arrangement is actually two, two-pipe arrangements. One of the two supply pipes carries cold water; and the other supply pipe carries hot water, to the coils connected to the circuit. There are also two return lines connected to each coil - one return carries the hot water tank back to the boiler or heat exchanger, and the other carries the cold water back to the chiller. Thus, hot and cold water do not mix.

This work employs the four-pipe arrangement in order to achieve the desired flexibility required to satisfy the varying heating and cooling requirements of the various spaces in the dairy plant. Fig 34 shows four-pipe distribution with two coils.

Fig 34. FOUR PIPE DISTRIBUTION WITH TWO COILS & SPACE TEMPERATURE RANGES.



Key to Fig 34

1. Heating coils
2. Cooling coils
3. Chiller
4. Pump
5. Header
6. Chilled water circulation pump
7. & (9.) Pressure relief valves (for hot and cold lines respectively).
8. Hot water circulation pump
10. Heat exchanger
11. Cold supply line
12. Hot supply line
13. Warm water return line
14. Cold water return line

CHAPTER SEVEN

THERMAL INSULATIONS

7.1 Need for Thermal Insulations

In order to minimize heat losses and heat gains in the thermal system, it is necessary to insulate the system. To accomplish this, the system is lagged.

To be lagged are:

1. The cold room, maintained at $-4^{o}c$.
2. The freezing room maintained at $-28^{o}c$.
3. The steam pipes
4. The boilers
5. The Hot water pipes
6. The chilled water pipes
7. The air Ducts

A heat insulating material will be a material which has a low thermal conductivity (or high thermal resistivity). Apart from its heat resisting properties it must also be able to withstand high or low temperatures. It must be relatively easy to apply, it should have a long life, be able to stand up to a moderate degree of rough handling and its costs should not b e excessive. A further derirable property is that there should be no fire risk. The American Society of Heating, Refrigeration and Airconditioning

Engineers recommended the following materials as good insulators:

1. Asbestos (K = 0.167 KJ/(hr)(m)^{o}c
2. Diatomaceous earth, earth, earth formed by dead water plants called diatons (k = 0.3654 KJ/(hr) m ^{o}c
3. Glass and rock wool (fribre glass), formed by making wool of the fine fibres made from silica minerals (k=0.046)
4. Magnesium carbonate k = 0.385
5. Cork (k = 0.048)
6. Expanded polystyrene (k = 0.04).

These insulating materials, and others, generally owe their low thermal conductivities to their cellular structure. Such that air is locked in their cellular structures and hence the property of the materials as thermal insulators really belongs to their static air content rather than the material itself.

Based on the lowest conductivities. Expanded Polystyrene (k=0.04), fibre glass (0.046), cork (k=0.048) and Asbestos (k=0.167) are considered for selection. But viewed along with fire resistance properties.

Fibre glass and Asbestos top the list. Fibre glass can be applied with so much ease to any structure and can withstand rough handling to a good extent.

Hence fibre glass is chosen as the insulating material in this project.

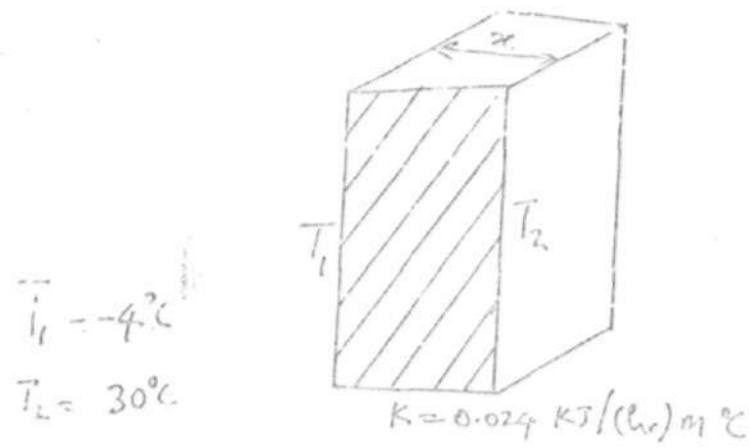
7.2 Calculation of Optimal Thickness of Insulation of the Cold Room

Minimum inside temperature (T_1) = $-4^{o}c$

Maximum outside temperature (T_2) = $30^{o}c$

Laging material = Fibre glass

Let = Optimal thickness of the insulating material, such that the heat flow per unit area q , across the outer boundary of the insulator is minimum.



$$q = K\Lambda (T_2 - T_1) \quad - 7.1$$

where q = heat flow across the insulation

k = conductivity of insulation

T_2, T_1 are outside and inside temperatures

= thickness of the insulator

But for the pipes, polystyrene is considered good

for the reason of cost. The insulation of the pipes are as shown in figures 35, 36 and 37. The American Society of Heating, Refrigerating and Airconditioning Engineers considers heat flow rate of 1 to 21 KJ/hour as permissible in a good insulation.

This project aims to achieve heat loss rate of around 10 to 15KJ/hour.

Hence puting $q = 10$

and $A = 1$

$$= \frac{KA\,(T_2 - T_1)}{10} \quad \frac{0.046x1(30^{o}-(-4^{o}c)}{10}$$

$$= 4.4 \text{ cm}$$

and puting $= 15$

$$= \frac{0.046 \times (30-(-4)}{15}$$

$$= 2.9\text{cm}$$

7.3 Calculation of Optimal Thickness of lation of the Ice cream freezing Room

Minimum inside temperature (T_1) $= -28^{o}c$

Maximum outside temperature (T_2) $= 30^{o}c$

from equation

$$= \underline{KA\,(T_2 - T_1)}$$

where q = rate of heat flow across to the insulation

k = conductivity of insulation

T_2, T_1 = Outside and inside temperatures respectively

= thickness of the insulation

Puting q = 10

and A = 1

$$= \frac{KA\ (T_2 - T_1)}{10} = \frac{0.046 \times 1 \times (30-(-28)}{10}$$

= 7.5 cm

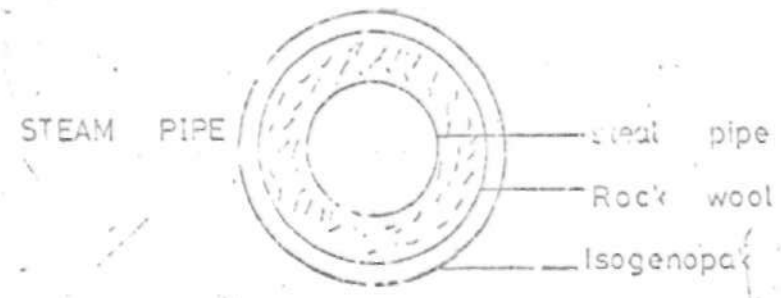
And puting q = 15

$$\frac{0.046 \times 1 \times (30 - (-28))}{15}$$

= 5cm

7.5 Pipe Insulation are as shown in figures 36, 37, and 38

PIPE INSULATION

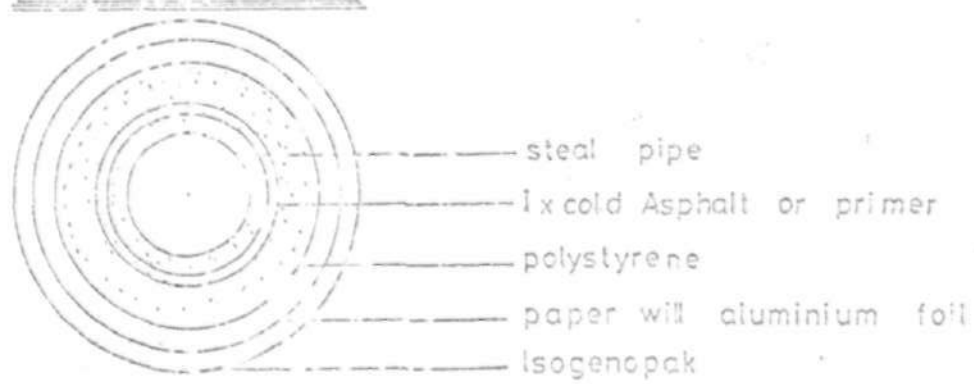


INSULATION THICKNESS

Diam of pipes inch	mm	steam pipes 1 – 10 atms	steam pipes 11 – 15 atms	condensate
½"	21	20	20	20
¾"	27	20	20	20
1"	34	30	30	20
1¼"	42	30	40	30
1½"	48	40	40	30
2"	60	40	50	30
2½"	76	40	50	40
3"	89	40	50	40
4"	114	50	50	40
5"	140	50	50	40

ICE WATER PIPE

PIPE INSULATION

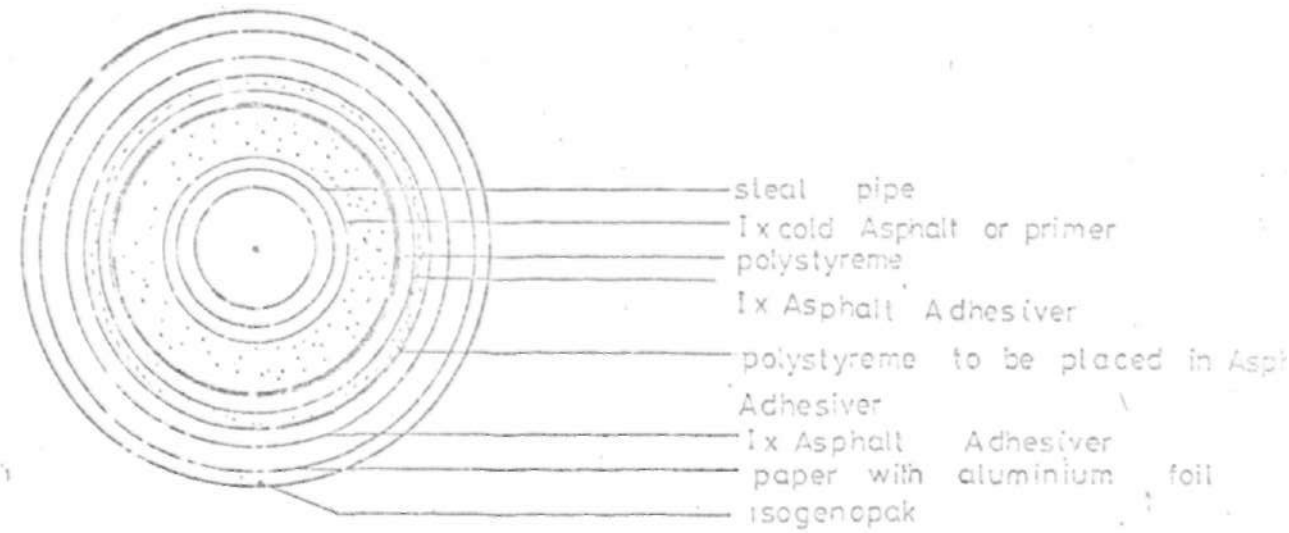


Isulation thickness

Diam of pipe Inch	mm	chill water
½	21	20
¾	27	30
1"	34	30
1¼	42	40
1½	48	40
2½	76	40
2"	60	40
3"	89	40
4"	114	40
5"	140	50

PIPE INSULATION

REFRIGERANT PIPE



THICKNESS OF INSULATION

Diam of pipes	Isulation thickness for various pipe temp		
	0 to −10 °C	− 1 to − 35 °C	− 36 to −45 °C
1/2 21 3/4 27	20 + 20mm	30 + 30mm	30 + 30mm
34	30 + 30mm	40 + 40 mm	40 + 40 mm
1 1/4 42			
1 1/2 48			
2 60			50 + 50mm
2 1/2 76			
3 89			
4 114	40 + 40mm	60 + 60mm	60 + 60mm
5 140			

cold humid days of the year reheat may be required. Therefore, a reheat coil must be placed along with the cooling in the air stream or the system will not be able to maintain the conditions desired.

Just as the components of an airconditioning system must be properly arranged, so must each component be selected so that the capacity can be varied to meet the variable room load.

Basically, capacity control can be divided into two types:

1. Two position; and
2. Modulating

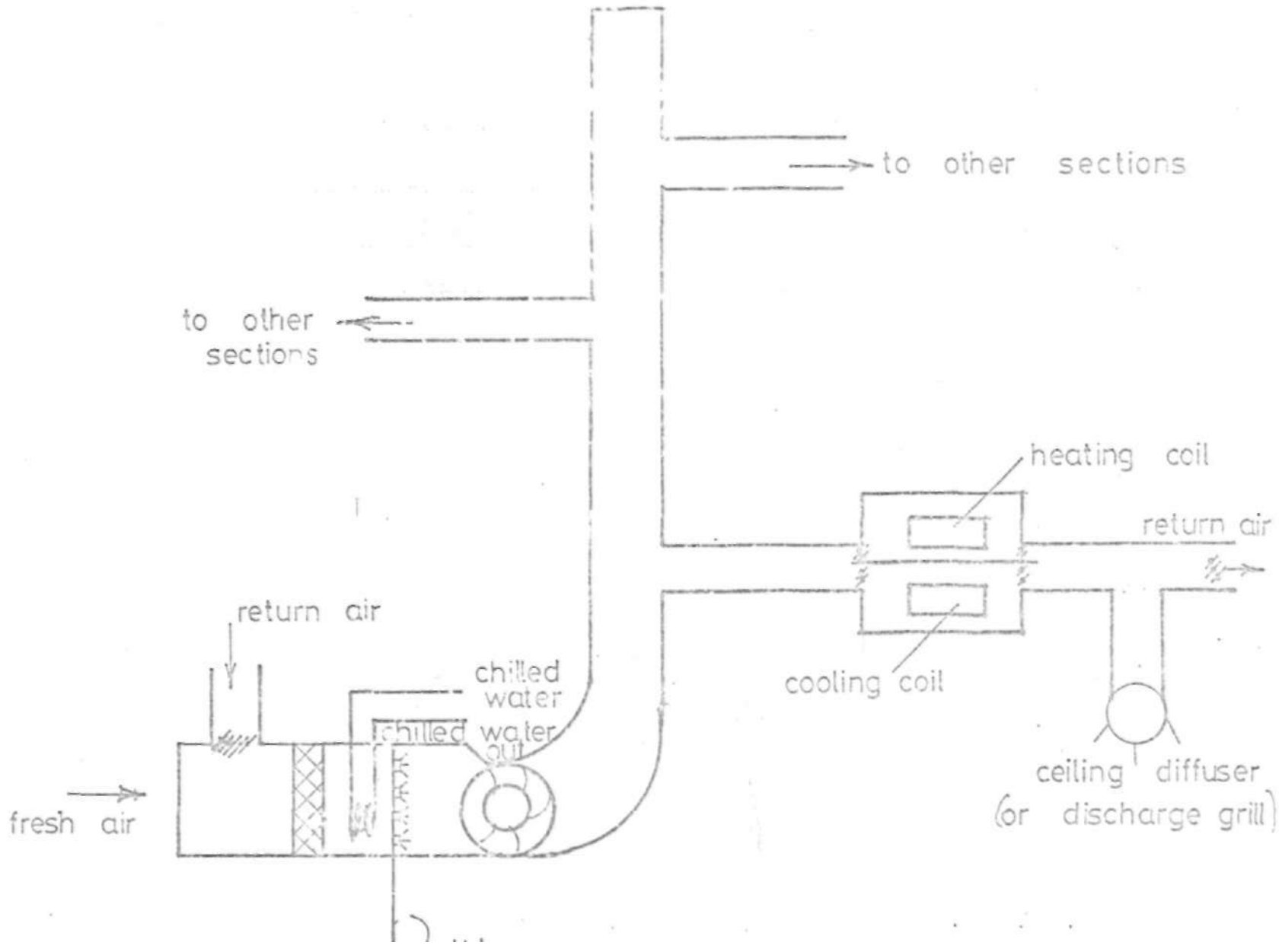
8.1.1 Two Position Control

Two position capacity control means that either full capacity of the component is available, or no capacity is available. A coil may be producing full capacity or no capacity. A pump may be running or not. A damper may be fully or open or fully closed, compressor running or stopped, a fan circulating the full quantity of air or fully stopped.

8.1.2 Modulating Capacity Control

Modulating capacity control signifies that the equipment can be producing capacities between full capacity and no capacity, either at several points or an infinite number of points.

FACE AND BYPASS DAMPERS CONTROL



To obtain better control on the central fan arrangements, bypass dampers and face dampers have been employed. See figure 8.1. A two position thermostat shuts the dampers during minimum temperatures and opens them when the temperatures reach maximum values.

8.1.3 Humidity Control

To maintain a desired humidity in the conditioned space, the proper rate of flow of air should enter the space with a condition on the load-ratio line. Problems of high-moisture loads may be handled by:

1. decreasing the evaporating temperature,
2. using reheat, and
3. using face-and-bypass dampers.

Decreasing the evaporating temperature of a coil will lower the sensible-heat factor of the coil. In fig 8.4 the load ratio is not satisfied by the coil with condition curve 1 because the condition curve does not intersect the load-ratio line. By lowering the refrigerant temperature and shifting the coil condition curve to 2, an intersection with the load-ratio line is achieved at point Y. The big problem however is how can severe latent loads be handled if the load-ratio

Fig 39: Psychrometric Chart of Air Entering the Conditioner at Point n and Leaving at Point y Along the Load-ratio Line

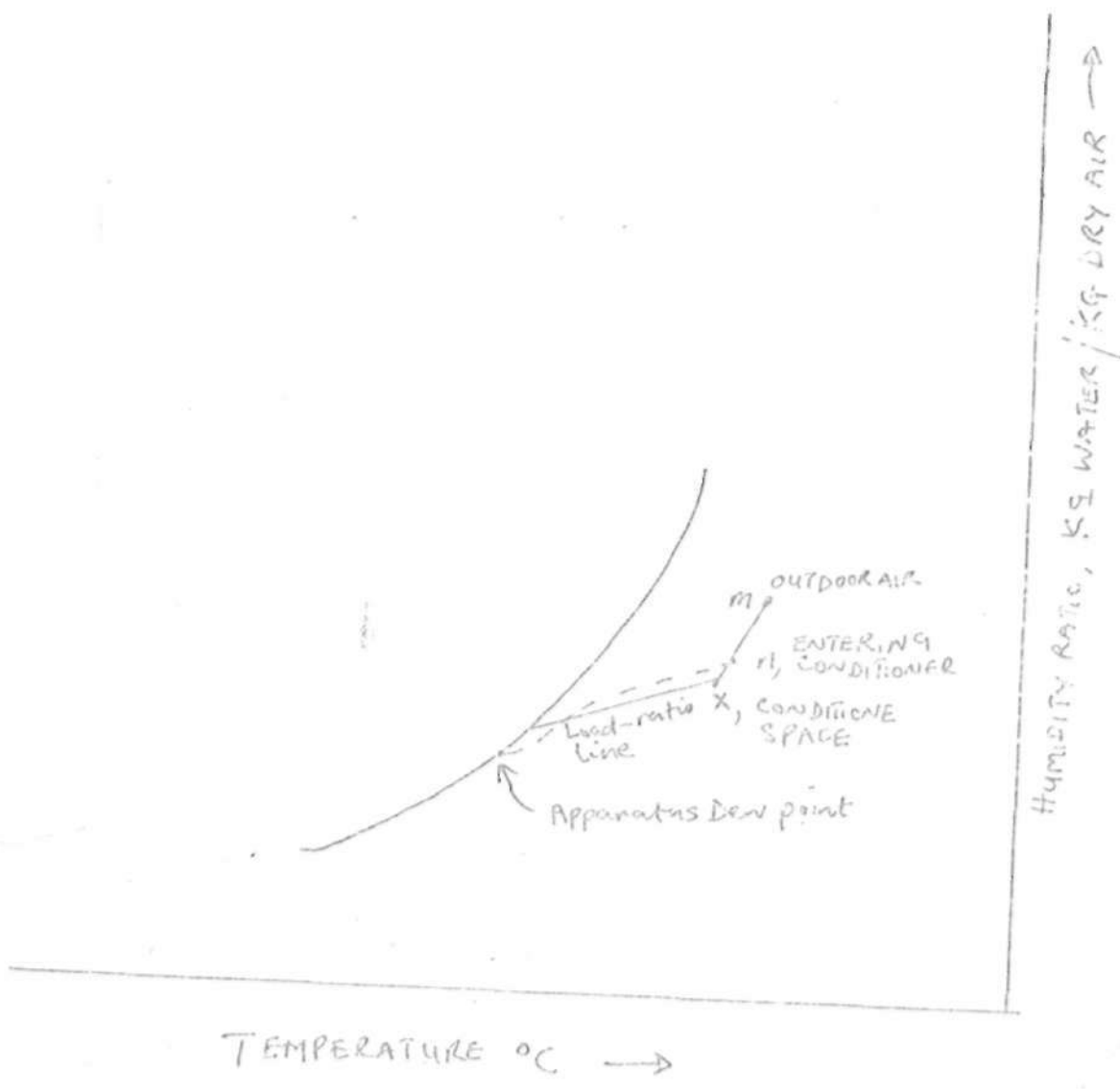


Fig 40: Meeting the Requirements of Latent-Heat Removal by Lowering the Refrigerant Temperature

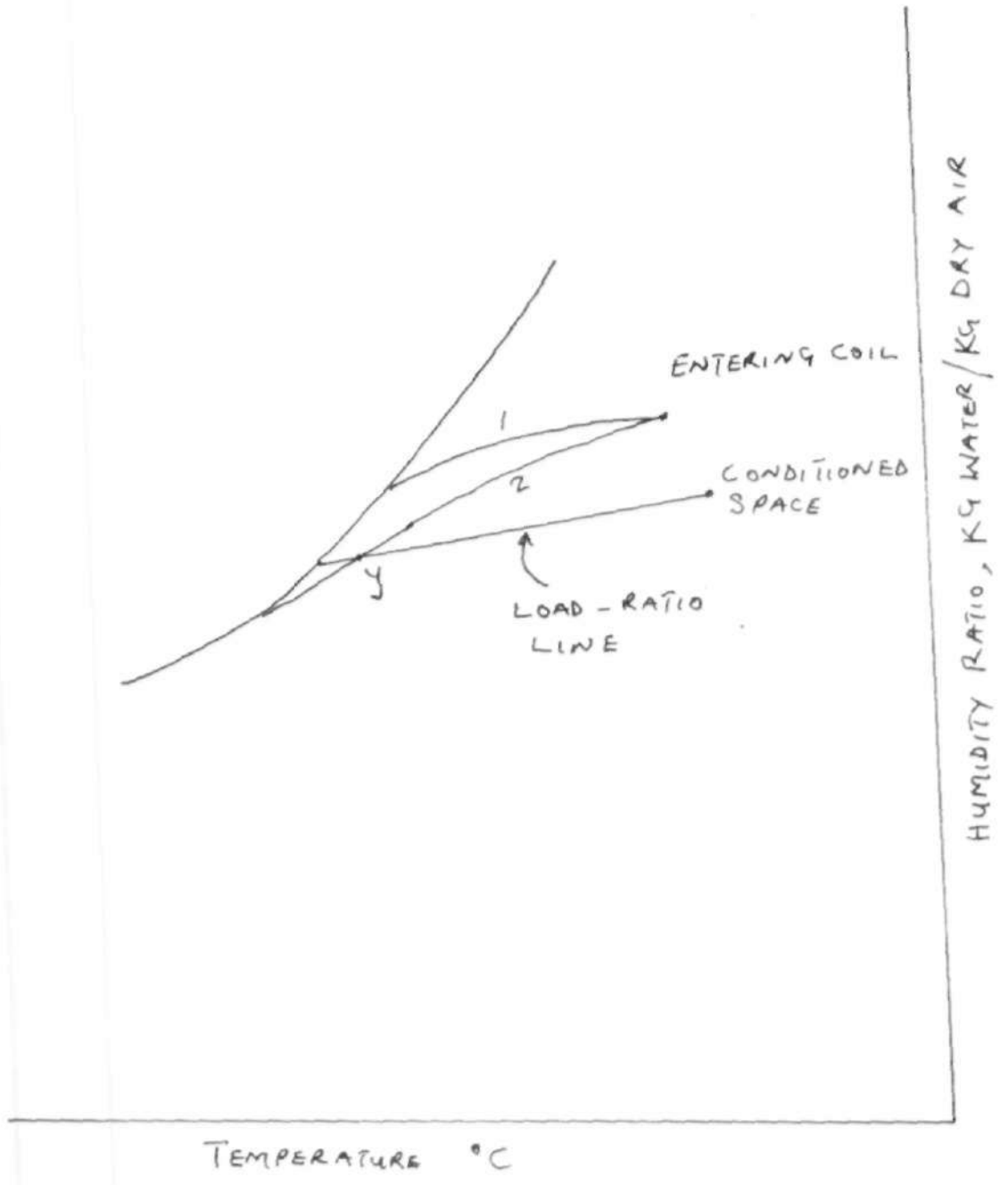


Fig 41: Using Reheat to Satisfy a Heavy Latent Load

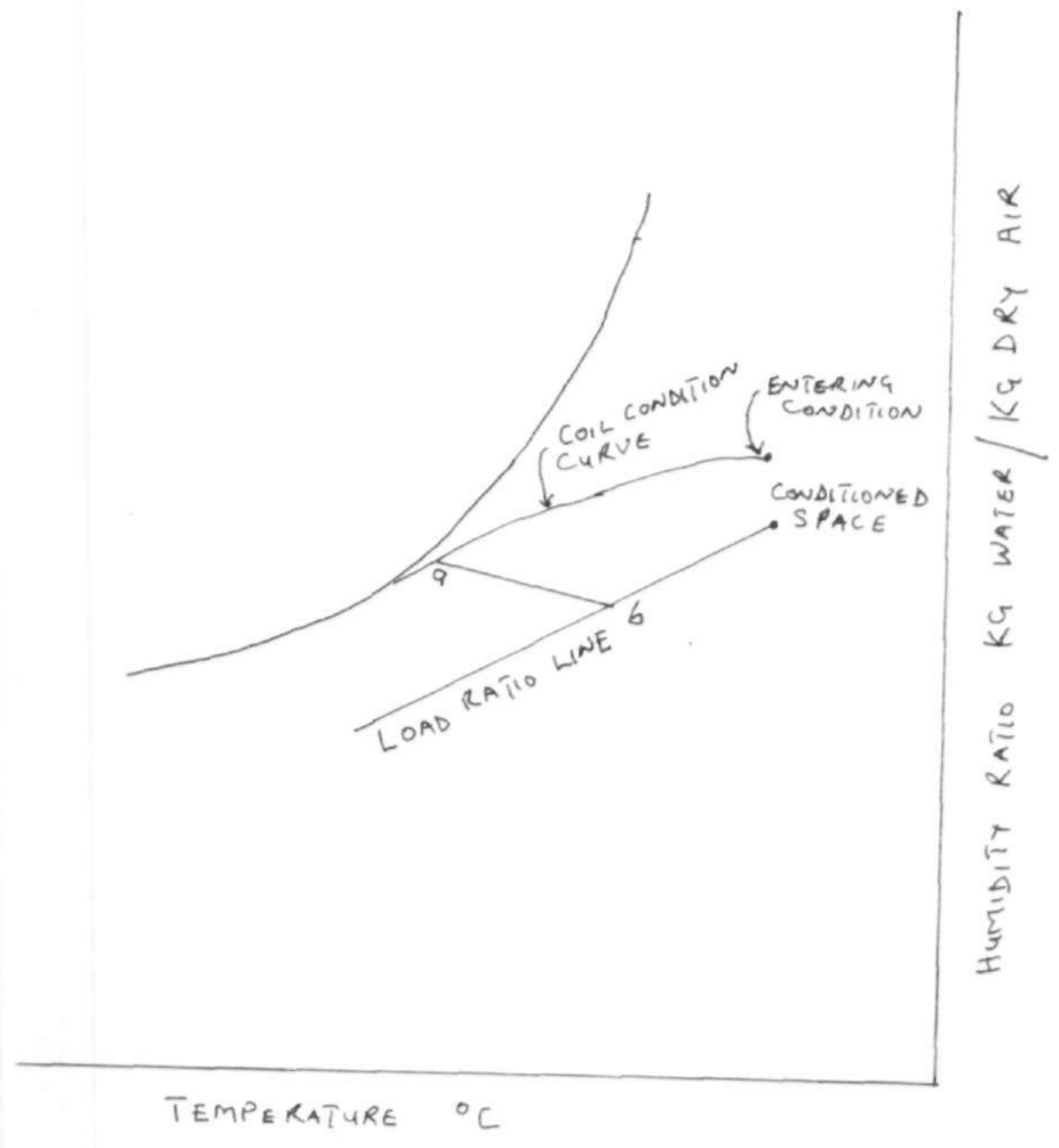


Fig 42(a): Face-and-Bypass Dampers

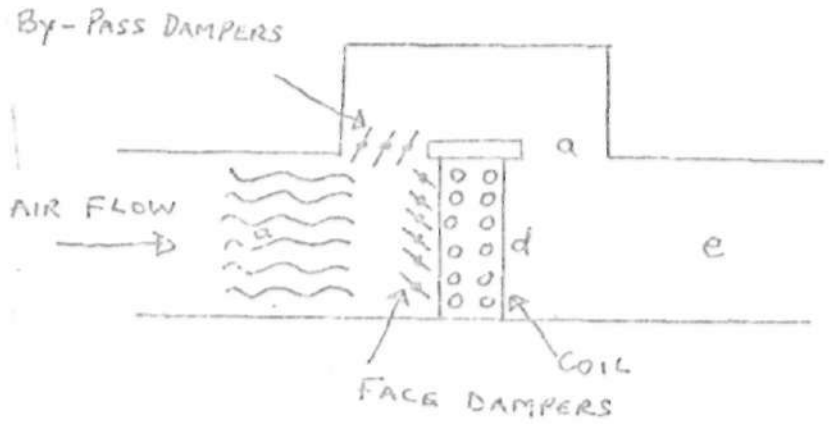
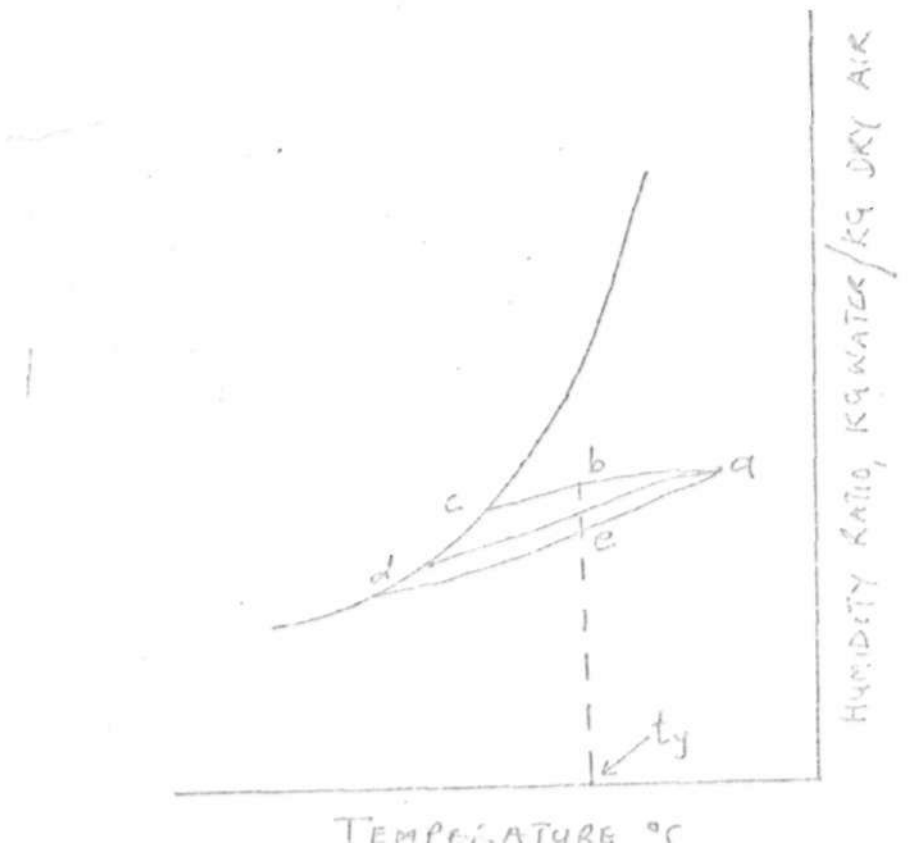


Fig 42(b): Psychrometric Chart of Air Conditions When Using Face and Bypass Dampers



line does not intersect the saturation line? In fig 41 the load ratio line is so steep that it is not intersected by the coil condition curve. Regardless of how low the evaporating temperature is reduced, no intersection occurs. This dilemma can be solved by reheating the air, process ab, after it leaves the cooling coil.

The third scheme for adjusting the humidity is to use face and bypass dampers. Fig 43a shows the equipment and fig 42b the points on the psychrometric chart. For example to satisfy the sensible heat requirement of the laboratory. We see that a flow of 0.95 m^3/min must enter the conditioned space at temperature of t_y = 10°c and 65% RH as shown in fig 33 and 42b. The air could be sent through a coil and cooled to point b on coil condition curve abc. To achieve a greater removal of moisture, cool only a portion of the air to point d and mix it with bypass air at condition a coil condition curve ad is as at lower temperatures than abc because of the reduced air flow. The result of the mixing is point c at temperature t_y, but with a lower humidity ratio than point b.

8.1.4 Temperature Control

Various methods are available to control the temperature in the conditioned space. In this project for instance chilled water is used, hence to control the temperature, the quantity of chilled water can be reduced or its temperature can be increased when the temperature in the conditioned space drops below the control setting. When using a direct - expansion coil, the compressor can be stopped, the flow of liquid refrigerant interrupted, or the suction vapour throttled when the temperature in the conditioned space drops too low. As indicated in fig 33 and the control system in this project must allow individual temperature control in several zones. This is a case of multi-zone control unit in which both heating and cooling coils are in use. A pair of dampers for each zone regulates the proportions of warm and cool air to maintain the proper temperature in that zone (see fig 33).

8.2 Control of the Water Supply System

Automatic control of the water supply is by the use of Hydrofor. See fig (7). When maximum pressure is attained in the supply line, the hydrofor switches off the electric motor driving the submersible pump. Also when the pressure drops to the minimum due to much consumption of the water, the hydrofor switches

on the submersible pump. By this means the control is automatic. The non-return valve ensures no back flow of water in the supply line.

8.3 Control of the Disinfection System

The scheme of chlorine disinfection, discussed in Chapter Three is as shown in fig 8, the basic m unit is arranged for manual adjustment of the gas feed rate to the required value. But automatic stop/start control is simply achieved by the cinclusion in the water supply line to the injector of a valve operated in conjunction with the main pump. Details are shown in fig. 43.

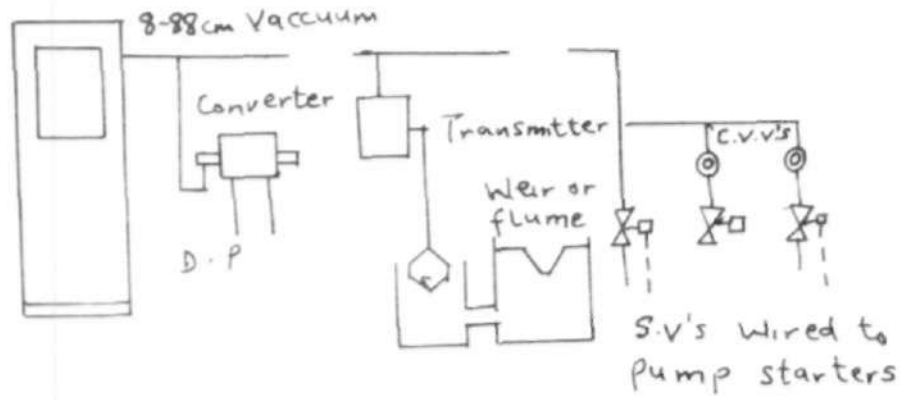


Fig 43: D.P. - Differential Pressure, C.V.V. Constant Vacuum Valve S.V. - Solenoid Valve

CHAPTER NINE

CONCLUSION AND RECOMMENDATIONS

Engineering services are a major and integral aspect of the building system as a whole, and as such, present the kind of challenge which makes design the most exciting activity to which man can aspire. As with all design endeavour, the need is to be systemic, systematic and immaginative.

This work concentrated on the design of water supply, sewage disposal, refrigeration and air condition systems in the building. This by no means exhausted the services system in the building, but for a dairy plant these constituted the most essential services for the industrial processes involved.

For efficient construction of the system, the following recommendations are made:

1. All the specifications made as to the size and capacities of the various equipments and components are based on theoretical computations It is therefore necessary in practice to incorporate into the these specifications certain factors:

(a) In selecting the pumps, the right capacity should be the theoretical capacity plus head losses in the pipes. Since head losses depend on I/d and L, losses are reduced by having

short runs of large diameter pipe with low velocity flow. Also from Bernoulli's principle, the lower the destination poin Z_2 in relation to the height of the source Z_1 the lower the losses. Losses can further be reduced by keeping bends and sudden enlargements and contractions in the pipe to a minimum.

(b) In specifying the air flow for the air conditioning system, the bypass factor should be incorporated. As air passes through a coil a portion is cooled to the surface temperature of the coil and the rest flows through the coil unaffected. The air subsequently mixes. The bypass factor is the x term in the following equations.

$$t_2 = t_1(x) + t_s\ (1 - x)$$

and $$W_2 = W_1\ (x) + Ws\ (1-x)$$

where subscript 1 refers to entering conditions of the air, subscript 2 to leaving conditions, and subscript s to coil surface conditions. If the cooling coil is a multitube system, then the bypass factor for the rows of the tubes will be derived as follows:

If t_3 is the outlet temperature from the second row of tubes

$$t_3 = X_1\ t_2 + t_s\ (1- x_1)$$

substituting the value of t_2 from eqn (9.1)

$$t_3 = X_1 (t_1 (X_1) + t_s (1 - X_1) + t_s (1 - X_1)$$

$$= (t_1) X_1^2 + t_s (1 - X_1^2)$$

or the Bypass factor (BPF) for two rows,

$$X_2 = (X_1)^2$$

and for n rows of tubes

$$X_n = (X_1)^n$$

If for example the BPF for one row of tubes is 0.67,

BPF for 2 rows = $(0.67)^2$ = 0.45

BPF for 4 rows = $(0.67)^4$ = 0.20

BPF for 8 rows = $(0.67)^8$ = 0.04 or nearly zero.

This work therefore recommends that the cooling coil should be composed of at least 8 rows of tubes.

There is the need for a stiff legislation to ensure environmental sanitation in all the dairy plants. Most industries dump industrial waste products into streams and rivers thus polluting drinking water for the neighbouring community. The Federal Government should promulgate a decree on environmental sanitation as well as improved efficient milking techniques.

The procedures involved in proper milking shall form part of the supervisory functions of the Food and Drug Administration Division of the Federal Ministry of Health. Since Modern milking facilities are relatively expensive, local industries should as a matter of compulsion establish these facilities at strategic locations in their catchment areas. This done, local dairy farmers will be encouraged to bring their milk for sale to feed the dairy plants thereby minimizing the amount of waste experienced in Nigeria today due to milks perishability and lack of suitable storage facilities on the part of the local dairy farmers.

20. Rogers, G.F.C., and Mayhew, Y.R. (1967) Engineering Thermodynamics, Work and Heat Transfer, London, Longmans, Green and Company Limited, pp. 12-19, 471-502

21. Stoecker, W.F. (1968), Refrigeration and Air Conditioning, London, Mc-Graw-Hill Book Company pp. 60-56.

22. Veal, T.H.P. (1966), The Disposal of Sewage, London, Chapman and Hall Limited, pp. 76-82.

23. Nigerian Meteorological Services, (1951-1983), Agro-Meteorological Bulletin of Nigeria.

REFERENCES

1. Advian, D.D., and Smith, J.E., (1972) "Applications of New Concepts of Physical-Chemical Waste Water Treatment", Progress in Water Technology, Tennessee, Vanderbilt University Press, Vol. I; pp. 273 - 286.

2. Aitchison, J. (1973), "The Estimation of Design Values", Journal of the Building Services Research Unit, University of Glasgow, Vol. 144, pp. 29 - 42.

3. American Society of Civil Engineers, "Design and Construction of Sanitary and Storm Sewers", Manual of Engineering Practice (ASCE) No. 37, New York, 1969.

4. American Society of Refrigerating Engineers, Airconditioning Refrigerating Data Book, Design Vol., 9th Edition, New York, 1966

5. Asimov, M. (1972), Introduction to Design, Prentice-Hall, New Jersey, pp. 120-132

6. Bockelmman, B.V. (1982) "Aseptic Packaging Systems" Aseptic Packaging Processing, A Collection of Lectures from Tetra Pak Seminars, Sweden, Tetra Pak International AB.

7. Davison, N. and Taylor, H. (1970), Revision Notes on Building Services, Kent, Newnes-Butterworth Group. pp 4-39.

8. Fair, G.M. Geyer, J.C., and Okun, D.A. (1971) Elements of Water Supply and Wastewater Disposal, New York, John Wiley and Sons, Inc

9. Fair, G.M. Geyer, J.C., and Okun, D.A. (1978) Water Purification and Waste Water Treatment, New York, John Wiley and Sons, Inc.

10. Ford, D.C., and Tischler, L.F., (1977), Guide to Waste Water Treatment", Chemical Engineering Magazine, New York, McGraw-Hill Publications Company.

11. Kinzey, B.Y. and Sharp, H.M. (1963), Environmental Technologies in Architecture, New Jersey, Prentice-Hall

12. Markus, T.A. (1976) "The Role of Building Performance Measurement and Appraisal in Design Methods," The Architects Journal, Vol. 146; pp. 28-31

13. Maver, T.W. (1971), Building Services Design, Riba Publications Limited, London

14. Maver, T.W. and Gratton, B. (1974), "Study of the Water and Energy Services in the Buildings", Journal of the Building Services Research Unit, University of Glasgow Vol. 144, pp 90-104

15. McGuineess, W.J. and Stein, B., (1964) Mechanical and Electrical Equipment for Buildings, New York, John Wiley and Sons, Inc. pp 17-28 and 138 - 248.

16. Merritt, F.S., (1966), Mechanical and Electrical Design of Buildings for Architects and Engineers, New York, McGraw-Hill.

17. Morse, F.B. (1965), Air Conditioning Manual, La Crosse, The Trane Company

18. Mulligan, J.T. and Fox, R.D., (1976), "Treatment of Industrial Waste Waters", Chemical Engineering Magazine, New York, McGraw-Hill Publications company.

19. Rayner, J. (1971), Basic Engineering Thermodynamics, London, Longmans Group Limited, pp. 648-655

QUESTIONNAIRE

COMPANY NAME AND ADDRESS: _____

_____

1. What is your product line _____

_____

2. What are the raw materials _____

_____

(a) How are they procured _____

(b) What is the optimal storage condition i.e. at what temperatures can these materials stored to remain good _____

3. What is your production capacity _____

4. What percent of your product is water _____

5. Details of the production process _____

_____

6. What is your source of water supply _____

(a) If source is borehole what is your pumping head _____

(b) Do you have water treatment scheme _____

(c) What is the method _____

7. Do you have problems with your pipeline Hot or Cold system _____

(a) Rusting _____

(b) Pipe blocking _____

(c) Product contamination _____

(d) Others (indicate) _____

8. What is the capacity of your water reservoir. How is the pumping station controlled _____

_____

9. What is the capacity of your boiler_____

10. How is hot water distributed and controlled?

_____

11. What is the cleaning procedure for:
(a) Floor and external parts of machines_____

_____

(b) Internal parts of pipes and heat exchanges

_____

12. What is your total electric power demand_____

_____

13. What is the capacity of your standby generator

_____

14. What is your method of refrigeration

_____

(a) What is your total cooling load requirement

_____

(b) And what is the actual capacity_____

15. Which different sections of the factory are refrigerated (To what temperatures)_____

_____

16. Do you use central refrigeration system for all these section_____

How is the system controlled_____

17. What sections of the factory are airconditioned?

_____

(a) What is the cooling load capacity_____
(b) Is this adequate_____
(c) What do you consider optimal capacity_____

_____

18. Do you use a central system of airconditioning?
How is the system controlled_____

_____

19. How are other spaces ventilated_____

_____

20. What is the fan duty (Vol. of air/min)_____

21. How do you dispose of solid wastes_____

22. How do you dispose of liquid wastes_____

23. Location of drainage points and manholes

_____

24. Storm drainage _____

25. All liquid wastes drainage system

_____ Factory wastes
_____ Urinals
_____ W.Cs
_____ Showers

26. Sewers system (size of the pipes)_____
Grandient of pipes and optimum flow rate

_____

27. Sewage treatment Plant (Type and capacity

_____

Is this adequate_____

What is the optimal capacity_____

28. What is your staff strength
(a) Administrative staff_____
(b) Factory staff (number in each section)

_____