COLD STORAGE OF SEED POTATOES IN INDIA

Thesis for the Degree of M. S.

MICHIGAN STATE COLLEGE

Bheru Jessaram Ramrakhiani

1948

This is to certify that the

thesis entitled

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B.J. Ramrakhiani

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of the requirements for

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Major professor

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COLD STORAGE OF SEED POTATOES
IN INDIA

By

Bheru Jessaram Ramrakhiani

A THESIS

Submitted to the School of Graduate Studies of Michigan
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in partial fulfillment of the requirements
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PREFACE

Perishable commodities show the greatest possible seasonal fluctuations in supply and prices. Adequate facilities for their cold storage forms an essential feature of a proper system of distribution, inasmuch as it insures controlled marketing, rational distribution and the elimination of waste and deterioration of quality. The need for proper cold storage is particularly felt in tropical countries like India. The Royal Commission on Agriculture in India reported in the following terms:

> "Cold storage is in other countries playing such a remarkable part in the marketing of goods, with results so generally profitable to the private enterprise, as well as to the farmer, that we do not doubt that sooner or later there will be a similar development in India."

Cold storage of seed potatoes is one of the urgent storage problems. In many parts of India, potato is a winter crop--planted in October and coming on the market in February. Seed potatoes for this crop are grown on certain hills in spring and harvested in May. At high summer temperatures, more than half the seed stock is damaged and the balance is not fit for seed. Therefore, cold storage plants of sufficiently large capacity to warrant their economical running, are necessary to be

set up either by private enterprise or sponsored by the Government.

In this manuscript, the writer has attempted to discuss the need of the problem, the physiological aspects of the potato storage, and application of engineering factors to the design of cold storage house, refrigerating machinery and auxiliary equipment. The discussion is more or less complete so that a horticulturist, an insulation contractor and a refrigerating engineer can all take advantage of the data collected, design factors used and methods involved.

The writer is very much indebted to Donald J. Renwick, Assistant Professor of Mechanical Engineering, under whose guidance this work was undertaken. The sincerest thanks are also due to Professor L. G. Miller, Head and other members of the Department of Mechanical Engineering for constant assistance and sympathetic attention.

East Lansing, Michigan B. J. RAMRAKHIANI
May, 1948

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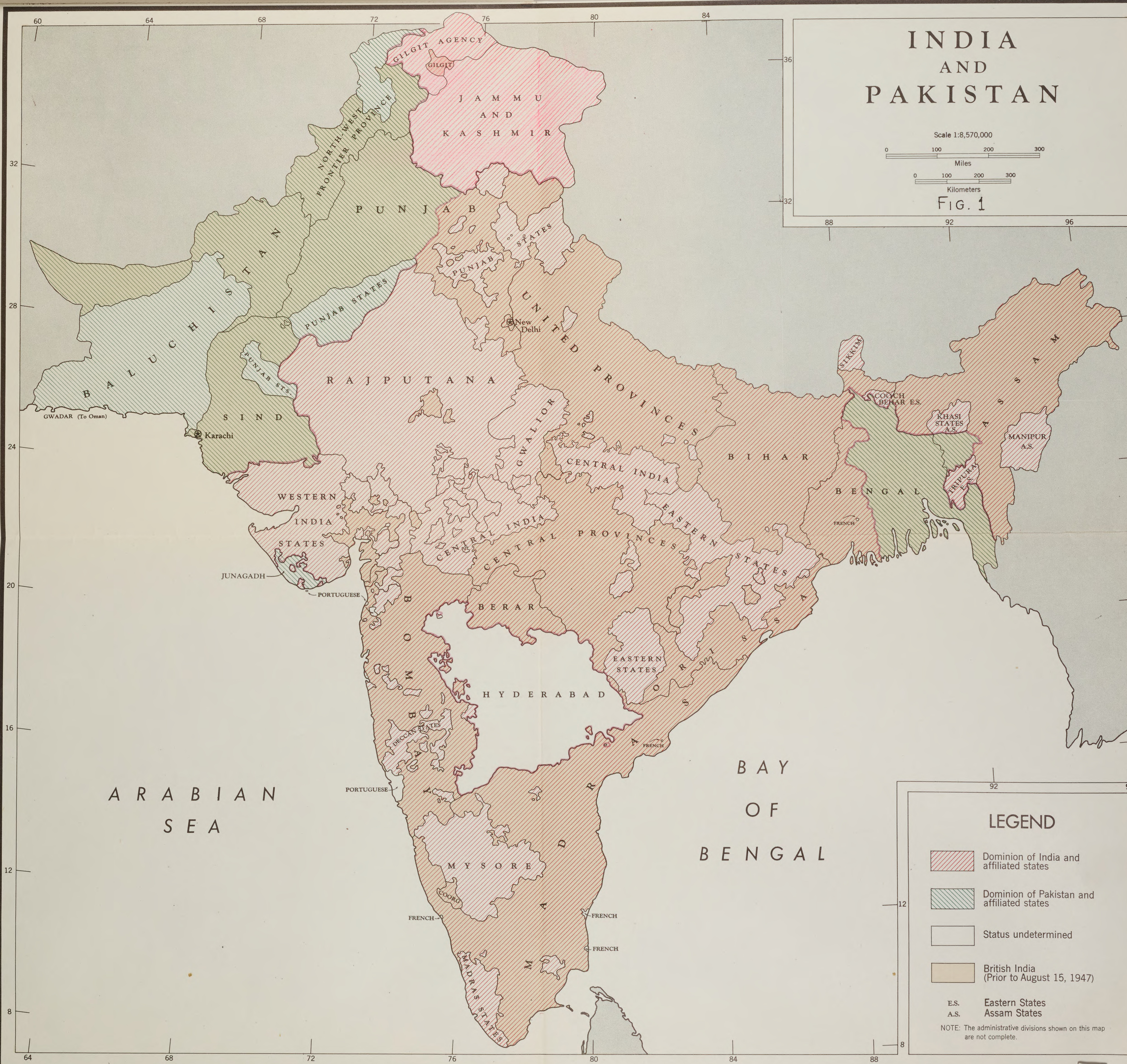
INDIA
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Dominion of India and affiliated states
Dominion of Pakistan and affiliated states
Status undetermined
British India (Prior to August 15, 1947)
E.S. Eastern States
A.S. Assam States
NOTE: The administrative divisions shown on this map are not complete.
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GILGIT
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NORTH-WEST FRONTIER PROVINCE
PUNJAB
PUNJAB STATES
BALUCHISTAN
GWADAR (To Oman)
SIND
Karachi
PUNJAB STS.
RAJPUTANA
New Delhi
UNITED PROVINCES
GWALIOR
CENTRAL INDIA
WESTERN INDIA STATES
JUNAGADH
PORTUGUESE
CENTRAL PROVINCES
BERAR
EASTERN STATES
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I. INTRODUCTION

A. Potato Production in India

Climate and Soil

The potato is an important commercial and cash crop in India* being worth 30 million dollars a year. The share of India both in world acreage and production is near one percent. The potato is a cool climate plant which grows best in regions where the mean temperatures are relatively low, generally not exceeding 70^{o}F. and where ample soil moisture is available during the growing season. Adequate moisture is especially important from the time tubers begin to form until shortly before the harvest. Well drained sandy, gravely or shale loam soils are well suited for potato production because they generally yield brighter skinned and better shaped tubers.

Areas

The main areas of concentrated production lie in the North, as the winter climate, sandy loam soil of rich Indo-Gangetic Plains and ample soil moisture due to monsoon rains make the area ideal for potato growth. The

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*Political map of India including Pakistan is shown in Fig. 1. The statistical information collected in this book is from the Report on the Marketing of Potatoes in India and Burma (1). These figures a re for the year 1940 and include those for Pakistan.

Note: The numbers in parenthesis indicate reference in the Bibliography.

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United Provinces, Bihar, East Bengal and West Punjab account for more than 80 percent of the acreage. Elsewhere in the South, the production is mainly confined to the Nilgiri Hills. An upward trend in the acreage of potatoes is noticeable. According to the available figures there has been an overall increase of about 12 percent from 1930-31 to 1937-38.

Crops and Seasons

The potato is the most widely grown of all vegetables in India and covers about 500,000 acres in cultivation, of which about 90 percent are estimated to be grown on plains and the remainder in the hills. Generally two crops, one in winter and the other in summer, are raised in the plains and submontane tracts. The winter crop is by far the most important and represents 90 to 95 percent of the total potato arma in the plains. Conversely, about 80 to 85 percent of the potatoes in the hills are produced in summer crop. The planting of the winter crop in the plains usually begins in the middle of September and continues until the end of November. In the hills, the summer crop is usually planted during February until March. The harvesting seasons vary through longer ranges, and generally speaking, harvesting of winter crop in the plains is during February-April and that of summer crops in the hills during May-June.

Production

India produces annually about two and one half million tons of potatoes of which about 8 percent are produced in the hills and the remaining in the plains. The United Provinces with an annual production of one million and one hundred and fifty thousand tons is the largest producer in India and accounts for 46 percent of the total production in the country. Bihar rates second and accounts for 20 percent. West Bengal produces 10 percent and the rest in West Punjab, Nilgiris and other areas. Areas of concentrated production are shown in the map. (Fig. 2).

Seed Supply

Production of seed in India is a relatively new enterprise. Originally, large quantities of seed stock used to come from foreign countries, particularly Italy. At present, total quantity retained for this purpose in India amounts annually to 400,000 tons or 16 percent of the total annual production. Bihar and the U. P. (United Provinces) are the most important provinces in this respect and account for 80 percent of the total quantity retained for seed. In addition to local supply, seed potatoes are still imported from Italy at Bombay, particularly for South India at an average of 10,000 tons per year during July-December. Centers of seed supply are shown in map (Fig. 3). It is reported that the Italian variety deteriorate in the course of a few years

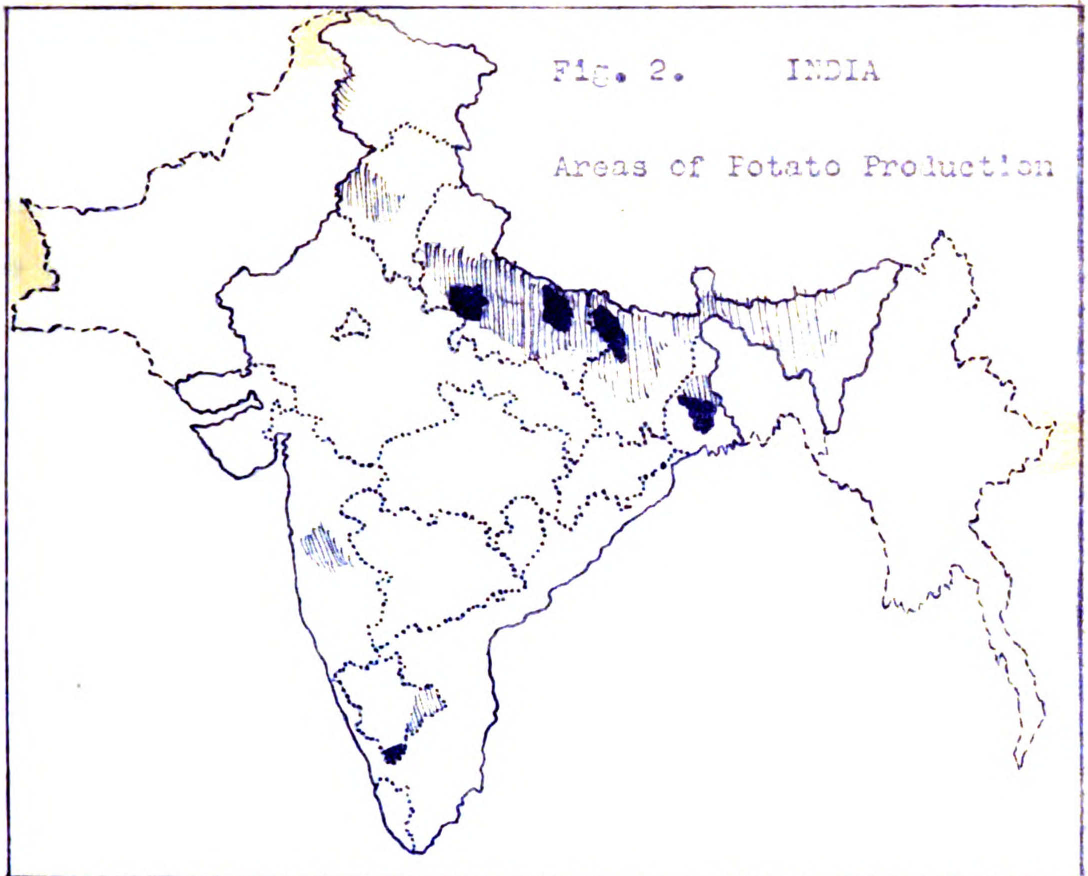


Fig. 2. INDIA

Areas of Potato Production

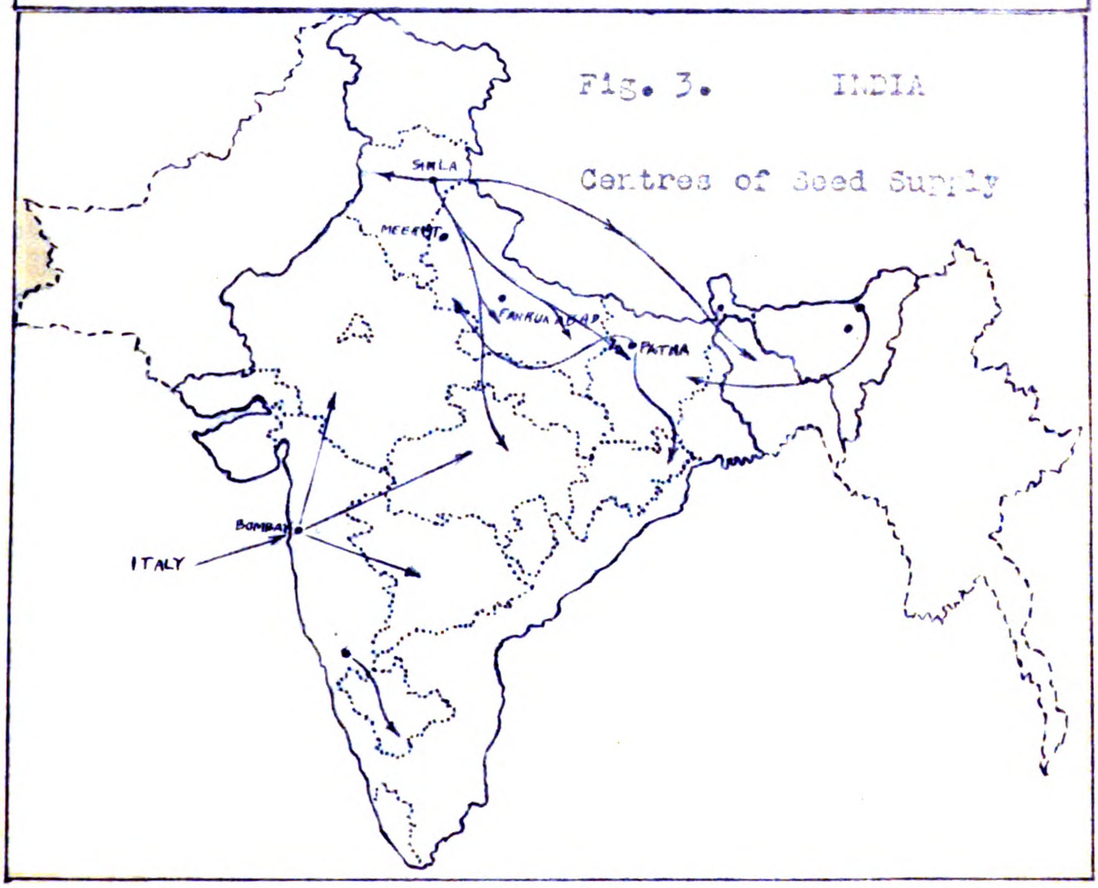


Fig. 3. INDIA

Centres of Seed Supply

under Indian conditions and so fresh supplies have to be obtained to renew the stock. Deterioration of quality of agricultural production can only be prevented by continuous renewal of seed of the highest quality from stock. The solution of the problem, therefore, lies in the production of fresh seed stock within the country, though for some time, the seed grown in India will have to depend on imported foundation stock of highest quality and suitable for tropical climate. Production of satisfactory certified seed requires special skill and care. But additional expense of producing certified seed in India is generally justified. This would ultimately help in increasing table stock production to a very great extent. The Principal requirements of seed certification will be discussed later.

Diseases and Insects

Potato diseases (12) in general are caused by fungi, bacteria or viruses. Important fungus diseases are early blight, late blight, black scrub and common scab. Dry rots may occur during storage. Bacterial diseases are ring rot, brown rot and black leg. Virus of degeneration diseases are mosaic, leaf roll, spindle tuber, yellow dwarf and spindling sprout. Other diseases are hopper burn, black heart and wilt or blue stem. Occasionally, diseases may be due to physiological causes during storage, i.e. to wrong storage temperatures and long exposure of the tubers to sunlight.

Again there may be insect species that are found feeding on the potato plant like potato moth and beetle, and they may be carried from one crop to the other.

Seed Selection

Control of insect pests and diseases is a big problem in producing more potatoes. But, nothing is so important in the control of potato diseases as the sources of supply and quality of seed. The best way is to buy nothing but certified seed. The soil may be of suitable type, with sufficient moisture and fertility, but if the seed is of poor quality, maximum yields will not be produced despite seed treatment, spraying and good cultural practices. It has been found almost without exception that high yields are obtainable with certified seed. The principal advantage of certified seed is the assurance that it is practically free from ring rot and such virus diseases as mosaic, leaf roll and spindle tuber. Hill selection is also of some value: by selecting seed from high yielding hills, a better grade of stock can be obtained. This shows how important it is to produce good certified seed and take greatest care during its storage.

As a result of extensive inquiries, it has been calculated that the weighted average yield for India is 109 mounds or 150 bushels per acre--largest yield per acre being 200 mounds or 275 bushels in Bihar. In other countries of the world Belgium tops the list with an

average yield of 224 mounds or 305 bushels per acre. The comparatively poor yield in India is mainly attributed to a relatively low standard of cultivation and to the fact that growers do not attach sufficient importance to the selection of seed used.

Inspection and Certification

Seed potatoes should be uniform size a nd quality, true to variety and free from diseases and mechanical defects. Commercial seed stocks in India are badly mixed, often wrongly named, unproductive and infected. Although seed certification and inspection has not yet been made statutory responsibility in India, these regulations, when adopted, would follow the same pattern as that in other countries. In the U.S.A., this extends from the time of planting until storing or transporting the crop.

Conditions required for inspection service (27) cover the selection of soils and plots, seed treatment and care during cultivation. In general, three inspections are made, two on the field and one after harvest. The first field inspection is made as early as possible to identify diseases; the second between the time of flowering and the time just before the vines mature; and the third after the crop is graded. Then they are sometimes bin or crate inspected after storage and shipping inspection of seed sold in lots. As, in this work, we are concerned with storage of seed potatoes,

the reproduction of storing regulations will be worth:

1. Potatoes grown for certification must be stored under conditions where there is no danger of their becoming mixed with uncertified stock. A certain section of the warehouse or cellar should be devoted to the exclusive storing of certified seed lots.
2. Certified seed potatoes must be stored in cool, well-ventilated cellars or houses. Where certified potatoes are stored in large piles, provision must be made for properly aerating the piles.
3. The grower will be held responsible for any injury due to frost, storage rot, etc., which may develop in storage.

To bring the seed production in India in line with other progressive countries, provincial and state governments should take steps, under the auspices of Indian Potato Research Institute (which is being set up and is described later) to improve the quality of seed potatoes. All the existing varieties grown in each zone should be collected and studied carefully. A detailed description of the characteristics of every variety should be drawn up, so that it might be easily recognized. Suitable varieties giving the best yields in different localities should be selected and classified.

In every case pure stock should be raised and kept

in good condition in storage for next planting. Pure stock of different varieties should also be raised in Government Farms and Experiment Stations and made available for distribution to cultivators who undertake to maintain purity of stock. It is only when sufficient seed stocks have been raised that seed certification system as prevalent in the U.S.A. and the U.K. can be adopted with success.

Grades and Size Classification

Potatoes are usually standardized according to their grades and size classifications. The certified seed potatoes are required to fulfil certain standards of grades and sizes. In India, standardization of grading has not been extensively adopted. Grading varies with different areas. In the absence of definite grades and standards it is difficult to compare prices in different markets, give estimates of weights contained in certain containers and is hard to make transactions without personally seeing the stock. Position is worse in case of seed stock, because factors such as the presence of other varieties and of diseased tubers, their condition, uniformity of size are of great importance. Due to this need, tentative grades for seed potatoes have been drawn up under the Agricultural Produce (grading and marketing) Act, 1937. The potatoes are being graded to some extent since 1939.

Grading can be done either on the farms or at the shipping or storage points. In the case of seed potatoes, the bulk of the preparation for market is usually done by hand on the farms. This would minimize handling due to grading or sorting at the terminal point and would also leave the storage of the product at the sole responsibility of the producer. But, in case farmers wish to avail of the grading facilities at the storage house, they may do so, preferably when the stock brought in from the field is all intended for storage; otherwise owner of the storage house may charge higher premium on grading.

Potato Research

Until a few years ago, very little attention had been paid to potato research. In 1940, the work of potato research was stimulated under the aegis of the Indian Council of Agricultural Research. Under the cold storage scheme at Poona financed by the Council, considerable work has been done on the effect of storage conditions on germinating capacity and the chemical changes that take place during storage of seed.

India has vast areas of agricultural land suitable for the production of potatoes, but in spite of such potentialities, it has been stated earlier, its annual production is very low, production per acre extremely low, and wastage due to disease and other factors very high. Problems relating to nature, variety and quality of seed requirements under different soil and climatic conditions, improvement in yield and prevention of diseases and loss during storage should form important items in the program of research work.

A proposal (17) to set up a Central Potato Research Institute for the development of co-ordinated fundamental research on all aspects of potato production and utilization, including a central seed certifying station for seed supply to the Provinces and States, is now under examination by the Government of India.

The Central organization will consist of (i) a main institute to be located probably in Bihar supported by three complementary research sub-stations in the hills in North India; (ii) three research sub-stations situated, one each in Bengal, Western U.P. and the Nilgiris and (iii) a seed organization for initial multiplication of improved seeds in a disease-free state and supply to the Provinces and States. The main institute will be located in Bihar, as it is an important seed-potato growing area, which exports more than 60 percent of its produce as seed to other Provinces and represents typical conditions for potato growing.

The primary object of the seed certification organization will be to insure that no diseased seed is used for potato growing. This will help in maintaining the purity of improved varieties. In order that the high yielding potentialities of improved varieties be maintained, for a number of years, it is necessary to use certified seed. The possibilities of improving and popularizing the potato crop in India lie in the direction of the evolution of higher yielding varieties for different climates and soils, the production of seed potatoes, the elimination of diseases which attack the crop in the field by producing disease-resistant varieties and the improvement of seed storage so as to prevent loss of rot and inspect pests.

B. Storage

Although the seasons of harvesting of potatoes in different parts of India overlap to some extent, there remain considerable gaps, particularly during long summer, and it is therefore necessary to store potatoes to meet the demand in the off-season. In the plains, the produce of the winter crop is generally stored as it keeps better. In the hills, the bulk of the summer crop which comes in April-June is stored. Since in this manuscript, we are concerned with storage of seed stock, we would not deal with the subject of general potato storage in detail, but would confine our attention to the need and methods of storage of seed.

Present Method and Losses

In Bihar, seed potatoes are stored in large quantities for a period of five to six months in the hot and rainy season (April to October). In the U.P., large quantities are stored for seed purposes in the districts of Farrukhabad, Jaunpore and Merrut. Present methods of storage are very crude. The godowns used for storage are built on a higher plinth and are usually surrounded by other buildings to keep them dry and cool. Before storing, potatoes are kept in the open for some time so the surface moisture may dry up. Potatoes are stored in sand as they keep better and are protected to some extent from the potato moth. During the period of storage, sorting is carried out many times and diseased and damaged tubers, due to warm temperatures of storage and heat of respiration, are removed. In the hills, seed potatoes are buried in pits dug out for the purpose.

Losses in storage are very high and quality of seed is very low due to deterioration and shrinkage. It is generally believed that losses due to decay and shrinkage from March to June, i.e. after harvesting and upto the outbreak of monsoon, vary from 25 to 40 percent. If potatoes are stored until September-October, as in the case of seed potatoes in the U.P., Bihar and Bengal, losses are usually 50 to 65 percent. In certain cases, losses are as high as 60 to 80 percent and what is left is not worth seed, not to talk of certified seed. The annual losses in storage and in the process of marketing and handling, at a very conservative estimate, amount to

500,000 tons valued at 6,000,000 dollars.

Need of Cold Storage

It was stated that in India, potatoes cannot be successfully grown in the plains during the summer and hence seed stock is not available for main crop. A possible solution would be to produce greater quantities and better varieties in the hills during the summer. But, the difficulty is higher cost of transportation from the hilly areas. Bombay imports potatoes mainly from Italy for seed purposes. It has already been mentioned that the Italian seed deteriorates in the course of a few years under Indian conditions and so fresh supplies have to be obtained to renew the stock. Further, it is neither safe to depend on foreign sources, nor are the imports large enough to meet the seed requirements of the main crop. The solution of the problem, therefore, lies in the production of fresh seed stock within the country during the main winter crop or in spring on the plains and carry it over to the next winter crop.

The percentage of losses in seed potatoes stored in the plains under present methods is very high, particularly for long periods of storage. In spite of high transportation costs, the U.P. Government are trying to have supplies of seed from hills where conditions for storing spring crops are more favorable. Still, much of the seed stock can only be grown on the plains in the U.P., Bihar and Bengal. It can be kept under controlled conditions, such as low temperature, high humidity and good aeration, only under refrigerated warehouses, located in the centers of production. Storage can be carried satisfactorily for 5 to 6 months of summer until planting of the main crop, that is, until September-October.

Losses in cold storage are very slight if the storage stock is of good quality. According to experiments and observations made by the commercial concerns, the loss in cold storage for five to six months may vary from 2 to 6 percent only.

As already pointed out, the actual quantity of potatoes for seed retained every year is about 400,000 tons. But the average seed requirements of seed per acre in India is 8 cwt. which gives total seasonal requirements for total acreage (500,000) to be 200,000 tons, which is half of the total quantity retained at present at harvest. The reason is the great loss of potatoes during the long storage period. If the losses in ordinary storages averaged 45 percent, this quantity can be saved for next season's requirements; or only 55 percent of the normal stock stored need be taken in storage, if refrigerated houses are installed. The prices of potatoes during March and April range from 45¢ to 75¢

per mound (82 lbs.), while during the period of July to November they range from $1.50 to $6.25 per mound. Thus, there is a big difference in price during the two seasons and there is an additional advantage for the farmer or merchant to make substantial profits at the time of planting the next crop. The potatoes will be of better quality which will give greater yield per acre to the cultivator who purchases his seed requirements from the cold storage depot. Therefore, there is three-fold advantage of installation of cold storage plants in Northern India.

Economical Size of Cold Storage

The size of the warehouse is determined mainly from two considerations: firstly, the capacity should be large enough to warrant cost of refrigeration machinery installation with all its maintainance, management and incidental expenses; and secondly, the warehouse should have continuous and sure supply of the commodity to be stored in sufficiently large quantities, every season during proper loading season. The more the prospects of getting large quantities of product to be stored, the larger will be the size of the storage and the more economical will it be in its first cost and running cost. Hence, areas of concentrated production can afford to have bigger storage houses. There are a few other considerations which may count: For instance, a cold-storage plant near the city may prove costly in running and, hence need be quite large, if a supply of stored product is assured; a small plant with combined facilities of storage of products other than seed potatoes may also prove economical; or cold storage plant forming a small part of a big factory may carry on with its existing power and water resources and other facilities.

In case of potato storage, there is one more consideration which seems quite important. During the loading season (April-May), refrigeration load will be at its peak value; but during storage period (June-October) which is probably hottest and the most oppressive season in India, the refrigeration load will hardly be one-third to one-half of the peak load (computations given later). Hence, ice manufacturing can profitably be carried on during this hot period, if there is ample demand for ice in that locality and nearby towns. Ice will come as a by-product with a little extra equipment like ice tank and other accessories, and will make the entire plant very economical.

All of these considerations should be taken into account in the order of their local importance while planning the size of the cold storage plant for seed potatoes. It would seem from past experience that a cold storage plant having capacity of 660 tons of seed potatoes with an ice plant would be the minimum size that could run economically. Larger plants of about 2000 tons capacity could be installed in areas of concentrated production and 1000 tons potato storage capacity may be taken as the average size.

Number of Cold Storage

About 20 potato and general cold storage houses are existing in India, but they are not generally availed of due to their location being either in big cities or port towns (for imported stock), which are far from farms and transportation is not so easy and cheap. Previously, people had not realized the advantages of cold storage and few had seriously thought of utilizing the cold storage facilities for a cheap and semi-perishable commodity like potatoes. The importance of cold storage is, however, being realized gradually, and in the case of long storage necessary for seed stock, the cold storage is almost unavoidable.

Taking total seasonal requirements of seed potatoes for main (winter) crop to be 200,000 tons, about 180 more cold storages of 1000 tons average capacity are still required throughout India, but concentrated at centers of seed potato production. This is, of course, a target figure based on present acreage under potato crop in India, but it is very evident that there is a good scope for many cold storages at or near the farms in the near future. We have not taken into account that some storages may even store potatoes for table use to be carried to later part of summer, or some general storage houses may offer other services like storage of apples, fish or poultry in separate rooms, if the storage houses are suitably situated to carry on that job too.

Location of Plants

Location of cold storage houses usually should be determined by marketing practices. They should be in proximity to users, public markets, produce dealers, commission merchants and the like. Location should be such that cost of trucking is a minimum and storage is easily accessible to motor trucks. If produce is shipped by railroad, a storage on a railroad siding may be desirable.

Other important factors are: nearness to the farms, topography of the locality, the availability of power and the availability of water supply for cooling purposes.

Taking all these factors into account, potato storage houses cannot be built on northern India hills due to the difficulty of transportation and market facilities, even though good qualities of seed potatoes can be grown there. Similarly, storage near the big cities will generally be far from production areas and may prove costly. Farm storages are also not practicable due to small sizes of farms under potato production and due to unavailability of power and market facilities. The best proposition would be to locate the storage houses in medium sized towns near the production areas, which have electric power and water supply, have importance as potato trade centers with railroad facilities, and in addition fair public demand for ice in the locality.

II. PHYSIOLOGICAL ASPECTS

A. Prerequisites for Storage

<u>Harvest Time and Maturity</u>

Much of the investment in a potato crop can be lost through carelessness in harvesting and in storage. Tuber development in late varieties continues over a period and ceases only when the plant is mature. As the potato tuber approaches maturity its keeping quality improves. Also, immature potatoes do not store so well as do mature potatoes because of a thinner skin which results in more rapid loss of water and hence in greater shrinkage. Injuries are more prevalent in immature than in mature potatoes because of their thin skins and brittleness. But to eliminate current-year infection by virus diseases, potatoes for seed are sometimes harvested early before disease-carrying insects become numerous. The claim for superiority of immature potatoes for seed is not justified on any chemical or physiological basis but is due to greater freedom from degenerative diseases in the immature seed (3). A better practice adopted on the hills, where the temperatures through day and night are not very high, is to pull the tops and leave the tubers in the ground for some two weeks for the natural maturing process, during which the skin becomes toughened. In hot, dry weather this practice may prove harmful and it would be enough to confine the digging to early morning or late afternoon and leaving the tubers on the field overnight.

Harvesting and Handling

Seed potatoes required for storage should be free from mechanical injuries. Usually potatoes for table use are picked in long narrow baskets fastened to the back of a person. They are then transported to the storage house, graded and either dumped in the pit storage or transferred into crates for cold storage. Careful handling of the seed crop at harvest time will be bound to pay for its efforts. Mechanical injuries to tubers incurred during the harvesting and handling processes have a very marked influence on the shrinkage and decay loss during sufficient storage. For these reasons, hand picking is preferable to harvesting even with a padded digger, in case of seed tubers; and handling in crates preferable to baskets or barrels as less bruising results (35).

Typical losses in weight from shrinkage and decay during storage following various methods of handling are shown in the table below (35):

Treatment	Percentage weight loss in storage 1 month	7 months
Immature Tubers:		
Carefully handled	3.05	8.07
Normally handled	5.04	11.70
Mature Tubers:		
Carefully handled	2.08	5.49
Normally handled	3.78	8.45

Grading

Before going into storage, potatoes should be graded in a grading machine or otherwise. Managers of potato warehouses should encourage growers to do more hand sorting in the field at the time of picking, as this would eliminate extra handling, reduce the need for so much mechanical sorting and mechanical injury caused by grader. Grader should be equipped with a roller type picking table to facilitate the removal of scabby, off type, cut and otherwise undesirable potatoes. All oversized, rough or growth cracked potatoes should be removed.

B. Storage Behavior and Conditions of Storage

Introductory

A potato in storage is a living, breathing plant tissue. As it breathes, carbon dioxide, moisture and heat of respiration are given off accompanied by loss in weight or shrinkage. Perhaps the most thorough studies of potato storage have been made at Cornell University Agricultural Experiment Station in 1933. The data collected from storages in two houses in Ithaca, New York (35) show that:

(a) The greatest shrinkage loss of potatoes occur during the first month and the sixth and seventh months of storage. Shrinkage during the first month appears to be due largely to excessive evaporation and a high rate of respiration through the injured and uncorked surfaces of the tuber; whereas the increase in loss during the last two months occurs simultaneously with excessive sprout growth and increase in temperature.

(b) The highest rate of respiration of the tubers, similarly occurs immediately after harvest. The rate then decreases rapidly, even if the tubers are placed at a comparatively high temperature. The rate again increases with resumption of growth subsequent to the dormant period.

(c) Loss in weight of potatoes due to respiration is very small in comparison with the loss in weight caused by water.

(d) Immature tubers lost more weight than did mature during storage periods of one, three and five months duration. During seven months storage period, however, physiological loss in weight was greater in mature than in immature lots. This increase in loss of weight of the mature over the immature lots therefore occurred during the sixth and seventh months of storage. It was concluded that the mature tubers terminated their dormant period sooner than did the immature tubers, depending largely on the internal environmental conditions of storage house.

<u>Storage</u>

From the above discussion, it is clear that the whole storage period can be divided into three parts: (1) curing period, (2) the holding period, and (3) warming-up period.

(1) Curing Period: It was shown above that immediately after harvest, potatoes even if not injured lose weight faster than during any subsequent period. At harvest time, the periderm of the tuber is only partly formed. The process continues for sometime depending on the temperature and humidity of the air surrounding the potatoes. Ideal temperature for rapid growth of this "thick skin" seems to be between 60^{0} and 70^{o} F. and the best humidity is near saturation. This "healing" can be accomplished by providing these storage conditions for the first week or two after harvest time, and is reported to have materially reduced weight losses throughout the remainder of the storage season. Curing

period temperature is usually easy to maintain; but when several weeks are required to fill large storage rooms, as may be the case in seed stock from the hills, it is better not to cure all the potatoes because the first lot stored may be held too long at 60°F. temperature or above (14).

(2) Holding Period: Experimental results (37) show that dormancy of the tubers could be maintained indefinitely at temperatures of 36°F. or below, but that at 40°F., some germination was almost certain to occur if the storage period was prolonged much beyond six months. In the same experiment, average loss of weight for Irish cobbler variety for normal storage was found 3.7% at 32°F., 1.55% at 36°F., and 3.42% at 40°F., for storing in barrel. Results for other varieties were similar. It has been suggested that the greater weight loss of sound tubers stored at the lower temperatures is due to slower "corking" of the periderm. The general conclusion to be arrived at from numerous widely separated tests is that seed stored at 36° - 40° F. are almost invariably more productive than those stored at 32° - 35° F. (44). For the normal potato storage period, room temperature of 36° - 38° F. with 85 to 90% humidity would prevent germination and at the same time insure as low a transpiration and respiration loss as is desirable. Under

36° - 37° F. storage with high humidity, seed stock can be stored from six to eight months, depending upon variety. Earlier varieties cannot be stored as long as late ones. Again, it is better to store seed stock at a little higher temperature, say 38°F. for early planting. Seed desired for late planting should be moved to about 36°F. after about three to five months.

(3) Warming Period: It is known that respiration is greatly accelerated when potatoes are moved from a low to a higher temperature. This indicates that a rather abrupt increase in physiological activity occurs when potatoes are taken out of cold storage. Because of this and also to give seed potatoes an opportunity to begin sprouting before planting, potatoes are warmed up for certain periods. A fairly good increase in yield has been obtained from seed removal from storage and held for 12 days in a room temperature of approximately 70°F. as compared with the yield from similar stock planted direct from storage.

Ventilation and Aeration

Ordinarily, ventilation is used only as a means of controlling the temperature and the humidity of the air in the storage, as in common storage or forced circulation refrigerated system. If the temperature and the humidity are maintained at the proper level, as in direct expansion or brine coil system, usually no further ventilation is needed to provide for air exchange for the respiratory process. Little attention need be given to the Carbon Dioxide, as storage construction is seldom so tight that the gas accumulates in quantities where it would either be beneficial in retarding respiration and sprouting, or where greater concentration would be detrimental in promoting black heart. On the contrary, ventilation has a disadvantage in that the ventilating air also removes and carries away moisture from the stored product and atmosphere. Therefore refrigerated seed storages are constructed so as to permit the least leakage or infiltration of outside air. But, continuous air circulation is needed in the storage, for the purpose of equalizing the temperature and humidity in different parts of the storage, thus preventing freezing or condensation and development of mold and sprout growth.

Condensation

If humidity is maintained at high level, no moisture will condense on cold surfaces or on the top layers of potato boxes or crates. If the temperature is kept low, say 34^{o}-36^{o}F. usually no harm will result from development of decay even if slight moisture condenses. Still it is good practice that the chance of condensation be avoided by maintaining high humidity and uniformity of temperature throughout the room.

Sweating

This results in condensation of atmospheric moisture on cold products removed suddenly from storage. It is higher when relative humidity of atmosphere is higher. It should be prevented as it favors the development of decay. This can be done by allowing the product to warm up gradually before taking out of storage. In the storage of seed potatoes, it has already been suggested to warm up stock in a room to 70°F. for about 12 days for reason of increased yield; so, sweating is out of the question.

Freezing Injury

It is of great importance to the commercial cold-storage man to know the exact freezing points of the product that he handles. Furthermore, knowledge of the freezing point of a commodity may be of special advantage to the warehouseman in case of alleged freezing damage. Freezing-point determination made on 14 different commercial varieties of potatoes averaged 28.9°F. (47). But freezing or freezing injury does not always occur when the fruits or vegetables are exposed to temperatures at or below their true freezing point. This is shown in investigations on potatoes in which tubers were cooled as much as 10°F. below their freezing point and were again warmed up without injury and without having become frozen. In view of this fact, the average freezing point given above should be considered only as the danger point at or near which, either above or below, there is a possibility of freezing injury if exposed for a sufficient length of time. In a cold storage, there is great likelihood of temperature of storage falling and remaining low for quite a long time unnoticed. Special care and constant supervision should be given to maintain constant temperatures well above average freezing point. This care is particularly important for seed stock. Even slightly damaged tubers are not fit for seed because of the rapidity of rotting of the seed piece by killing their tissues. Freezing injury on the field is rarely possible in India in the winter crop, as the temperatures rarely go below 29°F.

Storage Sanitation

Potatoes should be placed in a clean, disinfected storage. The storage interior should be free from refuse, dirt and mold. Empty storage should be disinfected with sodium hypochlorite or copper sulphate solution. The rooms should be closed for a few days following the application, before next fresh supply is brought in. Rooms should be washed out and cleaned once every three months. Potatoes in storage should be handled carefully to avoid bruises and cuts; otherwise they are likely to be damaged by various forms of decay before the end of storage period. Potatoes should not be stored in the same room with fruits, nuts, eggs or dairy products because of the objectionable flavor they will impart. For instance, potatoes stored with apples will cause the latter to absorb the potato odor and taste to the extent of making the apples unsaleable. Hence, if storage of any other commodity is intended, it should be made in exclusively separate room. For the purpose of this thesis, we will consider that only potatoes are handled, mainly for seed purpose.

Inspection

Potatoes should be inspected by a qualified inspector when received for storage and again within 30 days. The frequencies of inspection thereafter will depend upon the condition of the stock as determined by previous inspections. Big warehouses can afford to employ their own inspectors or inspecting agencies. But small storage houses handling only one product like potatoes cannot afford to seek such service. If need be, government may keep an inspecting staff on a non-profit basis. The inspection certificate is an authoritative document for both parties and would, in most cases, avoid disputes on the quality of the product before and after storage.

C. Conditions After Storage

It was pointed out under "Warming Period" in this chapter on Conditions during storage that respiration is greatly accelerated when potatoes are moved from a low temperature to a higher one. Again, it has been shown by Kimbrough (22) that after a storage period which is long enough to cause the maximum respiration rate, the period in the storage life of the tubers seems to have no appreciable influence on the respiration rate when potatoes are removed from storage. The rate of respiration after removal is therefore independent of the length of storage.

To avoid sudden changes from cold temperatures to higher temperatures seed should be warmed up in storage by shutting off a portion of refrigerating load, until the whole stock is either disposed of or transported. But when sudden changes do occur, or when first lots of seed stock are removed from cold storage for transit, special attention must be given to the ventilation of potatoes, as otherwise the period of abnormally high respiration following the change of temperature may cause the potatoes to heat and sweat, resulting in favorable conditions for the growth of decay organisms. They are more susceptible to black heart at this time.

Transportation

It is important that when the seed stock is removed from storage for planting, it should be transported as quickly as possible to the farms where they are to be planted. For long distances, railroads remain the largest means of carrying potatoes from one place to another, though motor lorries are also used in Northern India. For seed potatoes, railroad authorities allow certain commission and charge one-quarter of the usual parcel rate. But, there is no provision in the Indian railways for refrigerated vans for the transportation of perishable commodities other than fish. The existing steel wagons are not properly ventilated and insulated and iced wagons are necessary to keep the commodity from heat of the summer. Some railroads have provided wooden vans for summer transport of fruits and vegetables.

III. INSULATION

A. Insulating Materials

Function

The function of a heat insulating material is the maintainance of an unbalanced condition of heat transfer against the unremitting efforts of natural forces to restore the balance. It is essentially a material having a large percentage of relatively small voids containing dead air. Little, if any, convection can take place within such a material and the solid portions effectively screen off the radiation, so that only conductivity plays the part in heat transfer and the low conductivity of dead air is utilized to a much greater extent than in open air. There is no insulating advantage in having air spaces more than 3/4 inches wide because above this width the insulating value remains practically constant. Narrower spaces have less insulating value, and, according to the U.S. Bureau of Standards, below one-half inch the insulating value is approximately proportional to the width. If these air spaces are filled with any good conducting medium like moisture, much of the insulating value of the material is lost.

Properties

An ideal thermal insulating material for a cold-storage work would possess the following properties:

Thermal Quality	(1) A high thermal resistivity or low conductivity.
	(2) A low moisture absorbing capacity and resistant to air infiltration.
Physical Quality	(3) Fire resistant and no spontaneous combustion.
	(4) Vermin or rodent proof
	(5) Odorless
Structural Quality	(6) Light in weight
	(7) Elasticity with rigidity
	(8) Not deteriorate easily
Cost	(9) Reasonable in cost
	(10) Easily and cheaply applied

<u>Classification</u>

Commercial insulating materials are now available in a variety of forms and a satisfactory material may be had for almost any building situation that may arise. Some of them possess a certain amount of structural strength while others do not contribute to the structural strength of the buildings. These can be divided into two general groups: (1) fibrous materials either in loose form or fabricated into soft flexible quilts confined between relatively thin layers of paper or textile, and (2) more or less rigid boards in which the components are bonded together in some way. Stiff fibrous insulating boards having considerable structural strength are poorer insulators than lighter and coarse material. These materials from structural standpoint can also be classified as: (1) Rigid or board type, (2) Semi-flexible, (3) Soft flexible or blanket form, (4) dry fill, and (5) Wet fill.

Types

(1) Rigid or board type insulating materials include corkboard and various kinds of lumber substitutes, in which fibres of sugar cane (canite), wood or corn stock are bonded together. They combine insulating value with structural rigidity.

(2) Semi-flexible insulating materials are made of belted fibres of flax or rye and are furnished in sheets of various thickness. They are sufficiently flexible to conform to slight surface irregularities but still are rigid enough to be self-supporting.

(3) Soft flexible or blanket form insulators consist of quilted fibres of such materials as eel grass, kapok, hair, asbestos, wood, jute or flax. Some of these materials have good insulating value.

(4) Dry fills are granular, powdered, shredded or expanded materials such as granulated or regranulated cork, mineral or rock wool, shredded bark, sawdust, planer shavings and expanded mica.

(5) Reflectories usually consist of aluminum foil and similar reflectors which may be applied as single

sheets or as successive layers of foil. They reflect the radiant heat from surfaces which are kept away from other materials.

Though cheap filler type materials are generally used for cheap farm storage, they are generally less moisture resistant and are not packed uniformly for low conductivity. From this point of view and for structural stability, rigid insulating materials like corkboard, mineral wool board and other fibre boards are recommended for refrigerated storage houses. Besides, board type insulating materials are more adaptable to brick, concrete block or other masonry construction and for use under concrete floors.

Comparison and Tests

Therefore, we will compare the board type insulating materials. Series of tests were conducted by Pittsburgh Testing Laboratory, Pittsburg, Pa., during 1936 on corkboard and 5 other different types of low temperature insulating boards commonly used in cold storage (15). Tests were meant to yield information concerning the following properties:

(1) Thermal conductivity

(2) Susceptibility to moisture absorption by both total immersion and by capillarity from partial immersion.

(3) Resistance of materials to compression for direct loading and the degree of recovery upon the removal of load. In the case of bulk material, tendency to settle under vibration.

(4) Transverse strength of the materials.

(5) Ability of materials to withstand handling likely to be encountered before installation.

(6) Ability of materials to withstand impact.

(7) Fire resistance or combustibility.

(8) Uniformity of dimensions--effect of humidity.

A summary of the results is given in the accompanying Table.

Selection

The tests show that corkboard and Mineral Wool insulating board compare favorably to each other, except for the fact that the latter is not as fire resistant and not as efficient as insulating material. As in tropical summer temperatures, fire hazard constitutes a main factor in building construction, mineral wool is not recommended. Other fibrous boards like canite (cane Product) and Uniflex (Jute Flax) available in India are cheaper than Corkboard, and though an installation using one of these competing materials may show a lower first cost, in the longer run they deteriorate easily and cause trouble and inconvenience. Refrigerating a space is an expensive process and in the high range of

Test	Corkboard (Density) High	Corkboard (Density) Low	Corkboard (Density) Average	Mineral Wool	Vegetable Fibre	Wood Fibre	Shredded Redwood Bark	Animal Hair
K. at 60°F.	.283	.251	.270	.336	.302	.288	.318	.279
Water Absorption (15 days)% by vol.	13.2	9.1	10.9	.7	59.0	11.7	15.1	23.0
Capillarity (15 days) Inches	.5	.25	.5	.5	5.5	.75	0	.75
Compression (10 Days) Load 10 lbs. sq.in.-inches	.278	.140	.200	.226	.098	0.930	.630	--
Recovery (10 days) after 10 lbs.sq. in.load-%	72.8	66.4	69.5	46.3	49.0	48.5	22.4	--
Transverse Strength Dry-lbs.	27	14	19	101	128	28	28	--
Transverse Strength-Wet-lbs.	19	9	16	112	16	11	10	--
Handleability (500 rev.) % remaining intact	90.0	11.8	37.6	19.9	99.5	99.6	3.7	--
Impact Indentation-inches	0.10	.04	.06	.38	.31	.21	.92	--
Fire Test-20 min. % remaining	79.6	55.3	70.7	16.3	3.8	50.3	70.2	4.0
Linear change with humidity	0.99	0.81	0.91	0.17	0.3	1.73	0.37	--

temperature differential to be maintained, first cost is often the least important cost of an installation which is to be in constant service for say, twenty to twenty-five years. It is the operating cost of the installation which goes on year after year that ultimately is of much greater importance than the first cost. Therefore considering only efficiency and performance of the insulating material, Corkboard is almost extensively used in commercial storage installations.

Specifications of Corkboard

Cork is found in the bark of a live oak tree which grows in commercial stands surrounding the western end of the Mediterranean Sea. The bark of the Cork Oak, stripped from the trees every ten years roughly, is cellular in structure. Insulating Corkboard is made by first grinding the cork into coarse granules, carrying them by an air blower system to screens and finally the packed moulds are baked in ovens or by dry superheated steam. No foreign binder is added. The baking melts the natural resins in the cork and these bind these granules together in a homogeneous mass of pure cork. Corkboard is light in weight, yet structurally strong; a flexible and resilient board of low conductivity that is a fire-retardant and moisture resistant; a practical and easily erected form of insulation which will not settle, warp, support bacterial growth, attract vermin, absorb or emit odors--all these qualities amply exemplified by several tests. The standard board or slab comes in sheet sizes of 12" x 36", 18" x 36", 24" x 36" and 36" x 36", and with standard thicknesses of 1", 1-1/2", 2", 3", 4" and 6". Though average thermal conductivity may be taken 0.27 Btu. the hour, the inch thickness, the square foot area, the degree temperature difference (F.), for computing the overall coefficient of heat transfer of composite walls, the safe value of 0.3 Btu./hr./sq.ft./deg.F. for inch thickness will be assumed. Average density may be taken 8.5 lbs. per cu. ft.

B. Vapor Barriers

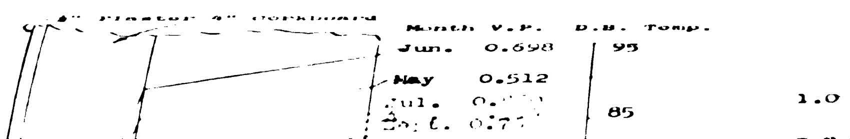
Need

We already know that the value of insulating materials lies in their cellular structure containing dead-air spaces. If these spaces are filled with atmospheric vapor condensed into water, much of the insulating value is lost. Condensation results when the dew point temperature of this vapor exceeds the temperature of any inside surface it contacts. No insulating material is waterproof. Vapor barriers are therefore employed in insulated walls and ceilings to prevent the entrance of moisture in the form of vapor.

The rate of moisture movement across the walls without vapor barriers and the direction of movement is determined by the difference in vapor pressure on the two sides of the wall. Accompanying Figure 4 shows the calculated average monthly temperature gradients through a typical masonry wall with corkboard insulation for summer months of Delhi (India) during the storage season, i.e. April-November inclusive, with corresponding vapor pressures shown in another column. A graph (Fig. 5) showing these vapor pressures above that maintained in the storage room is also shown. This shows very clearly that throughout the potato storage, outside vapor pressures are much higher and vapor movement will be from outside to the inside of the walls and the rate of movement will be very fast unless prevented by the barriers on outer surfaces of the insulating material. In the case of outside walls, this is often augmented by wind pressure.

During these hot summer months the temperature in attics above ceiling may be 10^{o} - 30^{o}F. higher than outdoor air temperatures. Thus, both the temperature and vapor pressure differences for ceilings will be much greater than those for walls and this might present a serious problem of moisture accumulation in ceiling insulation. Therefore a good vapor barrier must be applied above the insulation, on the top of the beams or joists.

Vapor barriers should be applied only on one side of insulated walls or ceilings, for, if by chance, the vapor does enter from one side, it should not be entrapped in the insulating or masonry construction and should be


Month V.P. D.B. Temp.
Jun. 0.698
95
May 0.512
Jul.
85
1.0

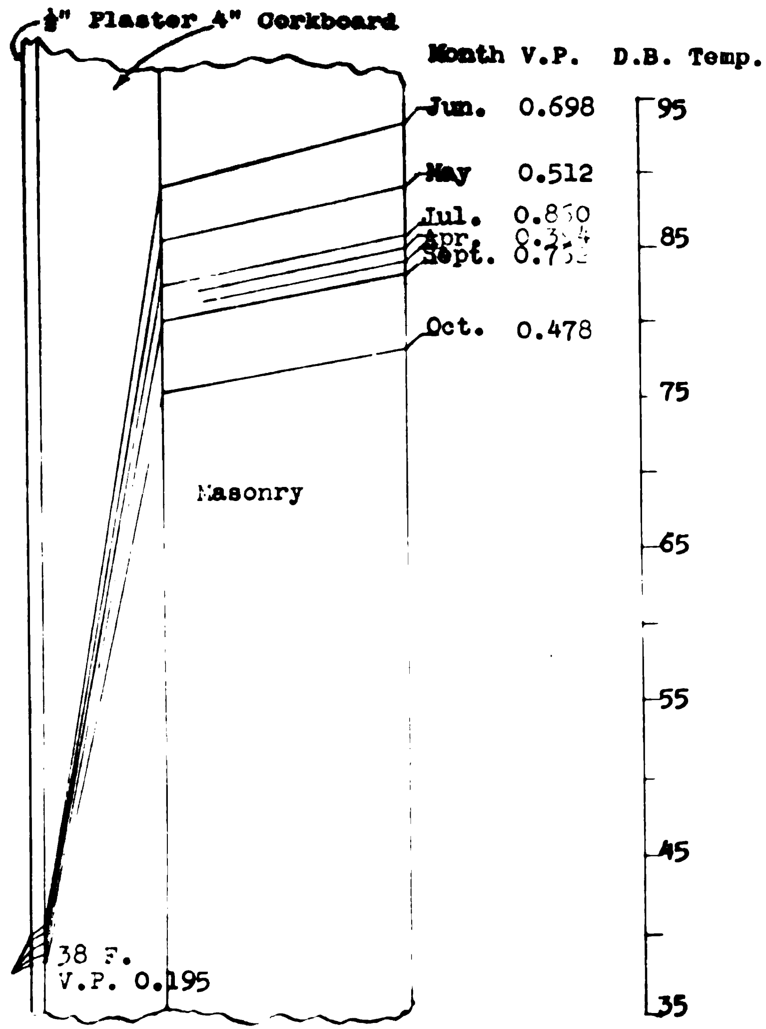


Fig. 4. Summer Temps.Gradients for Insulated Wall in Delhi.

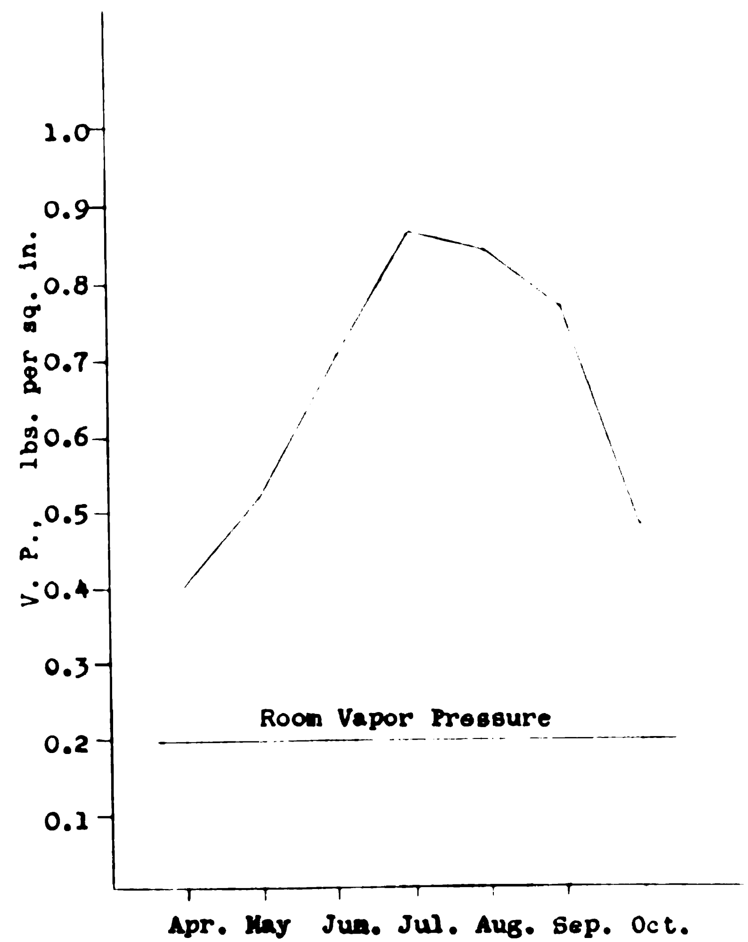


Fig. 5. Graph of Air Vapor Pressures during Summer in Delhi.

allowed to escape in the normal course of vapor movement or aeration from the other side.

Materials

Cement mortar or ashphalt are used as binding as well as air and moisture proof materials. Cement is, however, not a good vapor barrier and does not bond well with cork. Any little movement may develop cracks. It is sometimes used along with asphalt bonding.

Asphalt is almost invariably used as a binder and vapor barrier in cold-storage installations. Commercial asphalt is a bitumenous substance, secured from natural deposits or from petroleum distillation. It is generally felt that asphalt in a solidified form is most effective of all practical materials in protecting insulation for disintegrating effects of air infiltration and moisture. To be useful, in this respect, it must have special characteristics:

1. A high melting point so that it will not run, sag or fail as a bonding medium when exposed to high summer temperatures.

2. Ductility when exposed to low temperatures, so that it may not crack.

3. A hard, tough surface at ordinary temperatures to withstand damage from abrasion or impact.

4. Freedom from odor which might contaminate or affect the product in storage.

A normal (hot) erection asphalt should have the following specifications:

Specific gravity at 60°F	1.065
Melting Point	200°F
Flash Point (open cup test)	460°F

Asphalt must be applied hot mixed with approximately 3 percent by weight of cork dust. Methods of applying insulation and vapor barrier are given in the latter section. Masonry surfaces from inside may be primed with asphaltic priming paint where corkboard is to be erected in hot asphalt.

C. Amount of Insulation

Economical Thickness

There should be sufficient insulation in the walls and ceilings of a cold storage building to reduce the heat load of the plant, but if insulation is too thick, cost of insulation will be extraordinarily heavy as compared to reduction in the plant load obtained. Hence, there is optimum limit to the economical thickness of the insulation, when the total of all the overall operating costs is minimum. No rigid rule, however, can be given depending on a difference of temperature at outside and inside conditions, as the duration of storage in a year has special bearing on the result. Again the most economical thickness of insulation would be different for loading period than for storage season, when the conditions are more steady and no product load would come into the picture to an appreciable degree.

MacIntire (24) has summed up the various costs of operation of a plant and by differentiating the total with respect to thickness of insulation, taking it as the only variable, he has found the most economical thickness of insulation in inches during storage as:

$$L = 1.743\sqrt{\left[\frac{A(t_a-t)F + \frac{0.329P}{U(t-tp)}\left(I'+R'+\frac{100}{Y'}\right)(t_m-t)}{B'\left(I+R+\frac{100}{Y}\right)+8.3S}\right]k} - \frac{K}{U},$$

Where the notation has following significance and values:

K = Conductivity of the insulating material = 0.3 Btu/hr/sq.ft/deg.F per 1" thickness;

F = Yearly load factor = 0.5, as storage season will be from Middle of May to Middle of October;

B' = Cost of the insulation per sq.ft. per 1" thickness delivered to the job in the equation $B = B' + \frac{C'}{L}$ (where C' is the cost of finish, plaster, nails, labor and overhead per square foot and B being the total cost of insulation applied) = 0.065 dollars;

A = The cost in dollars per ton of refrigeration per 24 hours delivered,

= cost of electric energy plus cost of cooling water plus incurring costs of refrigerants;

If unit runs with two B.H.P. input per ton, electric energy consumed in motor = 0.746 x 2 = 1.5 K.W.hr./hr. = 36 K.W.hr./24 hrs. If unit of energy for industrial area is at the rate of 1-1/2 annas (equivalent to 3¢), cost of electric energy per day = $1.08;

Water consumption may be 1000 gallons per day at the rate of $0.2 per 1000 gallons;

For replacement of brine or primary refrigerant, we may add $0.32 more per day.

So the value of A = $(1.08 + 0.2 + 0.32) = $1.6 per day;

I = the interest rate, as a percentage, for the insulation investment = 6 percent per annum.

R = the repair cost per year as a percentage of the insulation investment taken as uniform throughout the life of insulation = 3;

Y = the life of insulation in years = 15;

I', R', Y' = corresponding values applied to machinery, etc. = 6, 5 and 10 respectively;

t_a = the temperature of the outside air, degrees F., as the average for the period of operations = 85 for outside walls; *

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*This 85 deg.F. average air temperature has been taken for places in Northern India. The necessary Meterological data for important places in India where seed storage houses may be built is shown in the accompanying table.

TABLE GIVING METREOLOGICAL DATA FOR SOME PLACES IN INDIA FOR SUMMER MONTHS

	April	May	June	July	Aug.	Sept.	Oct.	Average
Ludhiana	78.4	85.6	90.7	86.2	85.7	82.8	74.8	83.5
(E.Punjab)	64	69	73	77	78	75	63	71
30°55'Lat.	49	45	49	68	69	67	54	57
75°54'Long	0.432	0.514	0.637	0.854	0.849	0.786	0.489	0.651
811.86 ft.Alt.								
Simla (E.Punjab)	58.7	64.2	67.3	64.2	63	61.3	55.7	62
31°6' Lat.	49	53	60	62	61	58.5	47	55.8
77°12'Long.	50	47	65	87	90	83	53	68
7110.61 ft.Alt.	0.276	0.307	0.434	0.543	0.536	0.483	0.265	0.406
Delhi	84.6	89.9	93.3	87	86.2	84.2	78.5	86.2
28°40' Lat.	64.5	70.5	77	78	77	75	65	72
77°16' Long.	32	38	47	67	67	65	49	52
717.81 ft.Alt.	0.364	0.512	0.698	0.860	0.833	0.762	0.478	0.644
Meerut (West U.P.)	83	89	97.6	86.3	85.1	83.4	76	85.8
29°0' Lat.	66.5	71	80.5	77.5	77.5	75	65	73.3
77°41' Long.	39	41	48	71	72	68	56	55
737.48 ft.Alt.	0.432	0.514	0.68	0.864	0.850	0.758	0.499	0.657
Patna (Bihar)	86.8	88.6	87.9	84.8	84.1	84	79.5	85.1
25°37' Lat.	69.5	76	79	80	79.5	78.5	72.5	76
85°14' Long.	41	56	68	81	82	79	71	67
182.84 ft. Alt.	0.469	0.704	0.877	0.949	0.949	0.916	0.697	0.794
Purneah (Bihar)	83.9	83.2	84.0	84	83.7	83.3	78.8	83
25°50' Lat.	71	75	79	80	79.5	79	72	75.2
87°34' Long.	55	70	79	85	85	83	77	76
625.00 ft.Alt.	0.592	0.782	0.92	0.978	0.97	0.944	0.758	0.850
Berhampore	85.2	84.7	83.9	82.7	82.9	80.3	72.9	81.8
(W.Bengal)	74	77	79.5	79	79.5	77.5	68	76.3
24°6' Lat.	60	71	82	86	86	85	78	78
28°17' Long.	0.543	0.809	0.924	0.953	0.939	0.929	0.781	0.854
66.45 ft. Alt.								
Coimbatore	83.2	81.3	78	76	77	77.2	76.9	78.5
(Nilgiris-	72	72	70.5	69	70.5	70	70.5	70.6
S.India)	58	66	70	70	73	70	74	69
11°Lat.	0.668	0.695	0.672	0.655	0.673	0.656	0.684	0.672
77°Long.								
1347.64 ft.Alt.								

For every place in each column:

First figure indicates average monthly Dry Bulb Temperature; in deg.F.
Second " " " " Wet " " ; in deg. F.
Third " " " " percentage humidity;
Fourth " " " " Vapor pressure, in lbs. per sq. in.

Note - The figures given above have been taken from Dr. Blanford's report on Meterology of India in 1884; published by the Superintendent of Government Printing, India, Calcutta.

t_m = the maximum outside temperature, degrees F. = 115, say;

t = the temperature of the cold storage room, degrees F. = 35 giving average round figure;

t_p = the temperature of the refrigerant in the evaporator piping, degrees F. = 10° for indirect cooling by chilled brine;

P and u are the functions in the equation $G = \dfrac{12000\ P}{u(t - t_p)}$,

where, P is the cost in dollars per sq. ft. of refrigerating piping in the cold storage room = 0.5 including all pipe accessories,

u = the overall coefficient of heat transfer for the piping = 1.5 BTU/sq.ft/hr/deg.F.,

and G = total cost of piping, etc. in dollars per ton of refrigeration, and can be worked out from the above values;

S = the value per year of cubic foot space in the cold storage room = $0.15;

U = coefficient of heat of the wall for the materials of construction and air films other than insulation material as given by the usual formula:

$$\frac{1}{U} = \frac{1}{f_i} + \frac{1}{C_1} + \frac{x'}{K'} + \frac{1}{C_2} + \frac{1}{f_o}$$

For masonry wall of solid brick, Indian style, 12" thick or hollow concrete block, light gravel, 8" thick with 1/2" cement plaster on the walls, U = 0.31 Btu/hr/sq.ft/deg.F. or Resistance $\frac{1}{U}$ = 3.22

Applying these values in the equation for economical thickness of insulation for outside walls, we get

L = 4 inches.

For roof t_a may be taken as 115 instead of 85, hence

L becomes = 5.07 inches.

For floor, on the ground level, t_a is about 75 deg.F. which is the ground soil temperature. This gives corkboard thickness

L = 4 inches, same as that of outside walls.

Recommended thicknesses

Many corkboard or refrigeration machinery manufacturers recommend the economical thickness of corkboard insulation for different temperature range or temperature differential in a tabulated form, as an approximation, one such table given by Veneman (42) is given below:

Range of Temp. in Storage Areas (F)	Walls ins.	Ceilings ins.	Floors on Ground ins.	Floors Above Ground ins.	Roofs of Attic Ceilings ins.
-5 to 10	6	6	5	6	7
10 to 25	5	5	4	5	6
25 to 40	4	4	3	4	5
40 to 50	3	3	2	3	4
50 to 60	2	2	-	2	3

So taking the temperature range of 25°F to 40°F for storage room, we get exactly the same values as arrived at by elaborate formula, except that for floor, for which the table gives 3 in. This deviation is probably due to soil temperature in India, which is in tropical region, higher than taken in many parts of the U.S.A. For Michigan, value is 55 deg.F. Hence, we will take the 4 in. corkboard insulation for floor.

Partition walls in the storage house are between any two of the storage rooms maintained at the same temperatures during storage season. But it may often happen that while one storage room is being loaded with seed potatoes after harvest, other two rooms might be left unrefrigerated. Hence, to protect against heat transmission during any such period, partition walls should also be insulated by 4" corkboard.

For flexibility in laying the insulation and for elasticity corkboard in sheet size 12" x 36" is mostly used. Floors and walls will take two layers each 2" thick, while roofing will have two layers, one 2" thick and the other 3", the 2" layer being on the outside, as will be shown later.

<u>Overall Coefficient of heat transfer of the Composite Walls</u>

Walls may be either 12 in. common bricks or 8 in. hollow concrete (Sand, gravel or limestone) with 4 in.

corkboard and 1/2 in. plaster. Conductance C for 12 in. brick wall and plaster is 0.34 Btu. per Sq. ft. per hr. per deg. F.

Taking K = 0.3 for Corkboard $U = \dfrac{1}{\frac{1}{0.34} + \frac{4}{0.3}} = 0.0615$

C for 8 in. Concrete block and 1/2 in. plaster = 0.52

U for Composite Walls $= \dfrac{1}{\frac{1}{0.52} + \frac{4}{0.3}} = 0.0655$;

Average for transmission calculations may be taken as 0.065, on higher side for safety.

Similarly,

for partition wall with 4 in. corkboard and 1/2 in. cement finish on both sides U = 0.069;

for ceiling with 5 in. corkboard 4 in. concrete slab on top and 1/2 in. cement finish at the bottom side U = 0.056;

for floor 5 in. slab, 4 in. corkboard and 3 in. cement finish, U = 0.066.

D. Erection of Corkboard

We will assume, throughout, masonry construction of the building--concrete or common bricks. For outside walls, 8" hollow concrete blocks or 12" brick wall will be used; ceiling will have concrete slabs over the cork-board insulation; and concrete flooring will be made.

Wall Construction

There are two methods of erecting corkboard insulation on masonry walls. One method, which is rather expensive, consists of Masonry surfaces being primed with hot asphalt before applying insulating board. The other economical method which is quite suitable for storage temperatures above $35^{o}F$. is laying first insulating course laid in cement mortar. This latter method is explained below:

The masonry walls are roughened by hacking or constructed rough from inside for cement bond. Each sheet of corkboard composing the first layer is covered evenly on one side with Portland cement mortar 1/2 in. thick, this mortar being mixed in one part of cement to two parts of clean sharp sand. Each sheet thus covered is slapped into place against the wall as near as possible to proper location. Buck stays 3 ft. apart are to be erected for security on walls, as the walls are sufficiently high and may not hold the first course of corkboard otherwise.

Successive layer of corkboard is installed against the first in hot asphalt with joints broken in respect to those of previous layer, after the cement mortar has set. This layer is to be additionally secured with impregnated hardwood skewers, driven diagonally into the corkboard, at least two to the square foot, or with large head galvanized nails, used in the same manner. Nails or skewer should project into the underneath layers at least one inch. Cement finish is finally applied in two 1/4 in. layers, first layer being roughened to take the second. For strength, metal lath may be applied on the corkboard before applying cement finish. An isometric view of the insulated wall is given in Fig. 6.

Columns

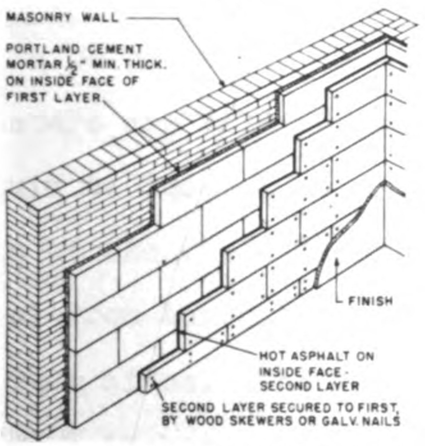
Columns which are in contact with the building structure and thus constitute conductors for the passage of heat should be insulated. Usually they are insulated with the same thickness of corkboard as that for the walls, up to three feet from floor or ceiling line, for 30° - 40° F storage temperature. When columns are part of an outside wall, they should be completely insulated. Location of these columns will be shown in the building layout. Method of insulating is the same as for walls.

Partition Walls

It will consist of two 2" corkboard sheets joined together in hot asphalt with 2 layers of 1/4 in. cement finish on both sides. As these walls do not support any load on the top, this 5" will will be stable and support itself.

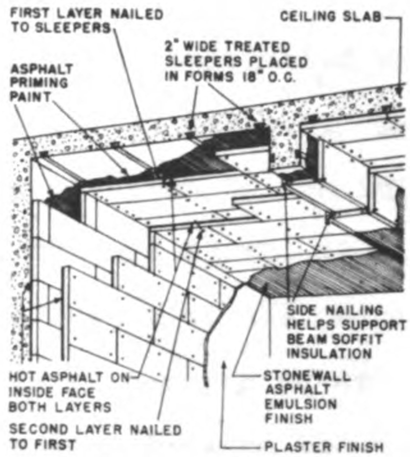
Ceilings on the Attic

Ceilings can either be laid in reinforced concrete



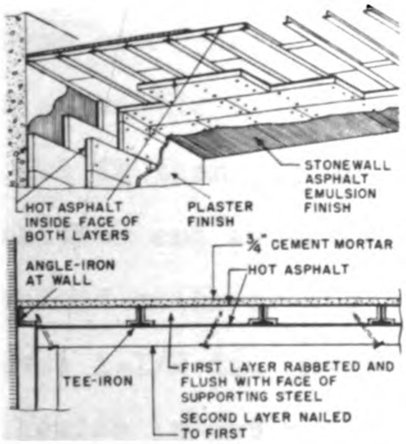
An economical method for rough walls of rooms where temperatures are above 35° F.

Fig. 6. Isometric View of Insulated Masonry Wall.



The sleepers widen from a 2 inch face to provide anchorage in the concrete

Fig. 7. Isometric View of Insulated Concrete Ceiling.



The weight and strength of the mortar provide a substantial backing for nailing succeeding layers

Fig. 8. Isometric and Sectional Views of Insulated Ceiling with Girders and T-s.

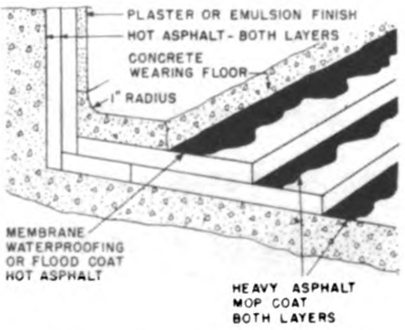


Fig. 9. Isometric View of Insulated Floor alongside a Wall.

in slabs or may be made up by a system of girders and tees with cement mortar or concrete laid over them. Method of applying corkboard insulation is different in either case.

In the former case (Fig. 7), it will be assumed that joist and wooden sleepers are in forms and concrete is to be poured on them. Wedge-shaped impregnated wood sleepers are laid on the forms 18 in. on centers. They should be located at a distance equivalent to the thickness of the first layer of the corkboard (3 in. in this case) from the walls, the top and bottom of the joist or beam sides. The face and depth of sleeper is at least 2 inches. Concrete is then poured on to the forms.

When concrete is sufficiently set and dry, the forms are removed and the surface is given two coats of asphaltic priming paint. The first layer of corkboard is then dipped in hot asphalt, quickly placed in position and additionally secured with large head galvanized nails driven obliquely into the sleepers at least 1-1/2 in. Subsequent layer (2 in.) is dipped likewise in hot asphalt and further secured with large head galvanized nails driven obliquely at least three to the square foot and entering the previous course at least 1-1/2 in. All joints in one layer are broken in respect to those of the previous one. Where horizontal and vertical insulation meet, tight neatly fitted joints should be both broken and staggered. Two layers of 1/4 in. cement finish is finally applied.

In the second method, Tee-irons of the size required by the span (14 ft. in this case) and loading on the top are located, say 18 in. apart between stems. They should be supported on channel iron or I-section beams anchored to the masonry wall by means of expansion bolts or other secure device. The first layer of corkboard (18 in. x 36 in. x 2 in.) is placed between the stems of the tee-iron and rest upon the flanges, the edges of the corkboard being rabbeted so that the underside will be flush with the face of the tee-iron flange. A flood coat of hot asphalt is placed on this first layer and allowed to harden. The subsequent layer of corkboard (3 in. thick) dipped in hot asphalt is quickly placed against the underside of the first layer and additionally secured with large head, galvanized nails, driven obliquely, at least three to the square foot, and entering the previous layer at least 1-1/2 in. As before, all joints in one layer should be broken in respect to those of the preceding layer, and where ceiling insulation meets that of wall, tight joints should be both broken and staggered. Two layers of 1/4 in. cement finish on the bottom side and 4 in. concrete unconditioned floor on the top should be applied, as shown in Fig. 8.

Floor Construction

Standard method of applying corkboard over concrete is generally used. The 5 in. concrete surface upon which the corkboard is to be installed should be clean, dry and reasonably smooth. Thereupon, a small quantity of hot asphalt is to be spread or mopped evenly on the floor and the first layer 2 in. thick laid therein with all joints tight and broken. The second layer also 2 in. thick is placed in like manner, due care being taken that the joints of any layer are broken in respect to those of the next layer. On the top surface of this layer should be spread or mopped evenly a heavy coat of asphalt or water proof sheeting to prevent ingress of moisture into the corkboard. Sand or gravel should be thrown into this top coating before it has set to facilitate the bond with the concrete wearing floor, 3 in. thick which is to be laid on it and smoothed round at the wall edges, with insulation also fitted neatly to join that of the wall with tight staggered joints. Isometric section of floor adjoining wall is shown in Fig. 9.

Brine and Ice-making Tanks

At least 6 inches of corkboard in two layers should be applied on the tank floor as described above, except that no sand or gravel is thrown or cement flooring made. This insulation should be placed over the area to be covered by the tank and by the side wall insulation, where the tank is to be adjacent to the masonry wall. Both layers of corkboard are supposed to have already been applied in a manner similar to that described under wall construction, previous to the placing of the tank. Any air space which might exist between the wall and the tank after it is placed shall be filled with granulated cork. A 2" x 4" wood plate is then fastened to the wall at the proper height by expansion bolts or some other mechanical device, in order to provide a support for the curbing.

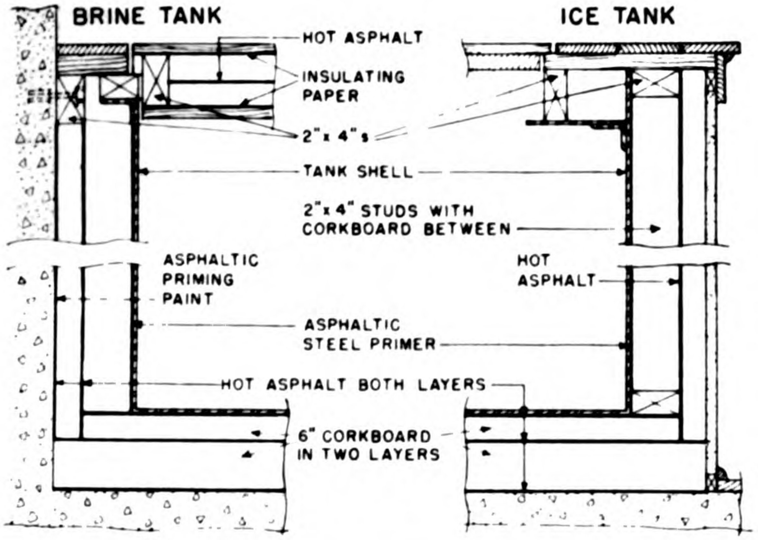
On the other side of the tank, corkboard is applied in two layers to a total thickness of 6 in. Studs equivalent in thickness to that of the first layer are placed tightly against the tank 24 in. apart and are secured firmly at the top to the tank, at the bottom to a wood plate of corresponding size. The first layer of corkboard, dipped in hot asphalt is tightly fitted between the wood studs and placed against the sides of the tank which is to be treated previously with a coat of steel primer. This layer is secured further by toe-nailing to the studs with large head galvanized nails. The second layer is dipped likewise in hot asphalt and so placed against the first la yer that all joints are broken. At the studs, this layer is secured further with large head galvanized nails and to the first layer with impregnated hardwood skewers. The outer layer of corkboard is to be covered with 1/2 in. cement plaster as usual and top of tank covered by wooden slippers. Cross section of Ice and brine tank is shown in Fig. 10.

Cylindrical Brine Cooler

Insulation for horizontal shell and tube brine cooler with flanged ends is one layer of 6 in. cork covering all around the cylindrical wall tightened by 1-1/4 in. band iron with clip and bolt; 6 in. plane corkboard on the two ends fastened all around by cork covering and tightened similarly. Space left at the bulging ends is filled with fine granulated cork. Fig. 11 is cross section of two types of flanged end brine coolers, one requiring a corkboard filler strip, as shown.

Insulation of Refrigerator Doors

Standard cooler or refrigerator doors are insulated with 4 in. of loose fill insulation, such as granulated cork or mineral wool. Before the insulation is installed, the box-like structure formed by the front panel and stiffners, made of conditioned lumber, is lined with a layer of moisture proof insulating paper. After the insulating material has been thoroughly packed and tamped into place it is covered with a final layer of insulating paper thus completing a moisture proof envelope around



The finish on exposed wall tank insulation may be sheathing, plaster or emulsion.

Fig. 10. Cross Sections of Brine Tank and Ice Tank.

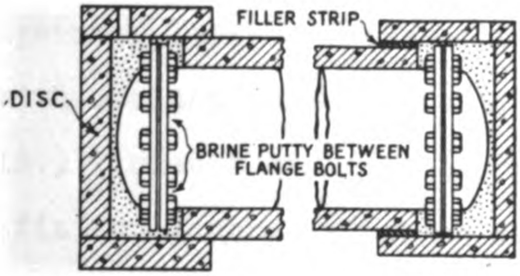


Fig. 11. Cross Sections of Two Shell and Tube Brine Coolers.

the entire insulation.

Cold storage doors are furnished hung in the frame by the manufacturers complete with necessary hardware, ready to set in the wall. Double seal is generally provided for flush type doors. The maintainance of these gaskets is important to prevent air leakage. A concrete sill, with a 3/8 in. tapered rise, is provided beneath the door to prevent sill-seals being dragged against the floor when the door is opened.

Pipe Covering

All refrigerant suction lines, cold water lines, if any, wastes, valves and fittings should be covered to prevent sweating due to carrying chilled or cold refrigerants and other liquids through areas of relatively high temperatures.

Suction lines for low temperature systems carrying refrigerants (brine, ammonia, etc.) should be covered with moulded cork, brine thickness 1.70 in. to 3.00 in., finished with one heavy coat of asphaltic emulsion finish, generally put on at the factory. Many times covering on suction piping at more than 10 deg. F. refrigerant temperature consists of hair felt in two layers each one in. thick, alternated with waterproof paper and finished with standard canvas jacket.

Cold water lines for cooling water should be covered with 3/4" wool felt covering, tar paper lined and finished with 8 oz. canvas jacket pasted on.

IV. REFRIGERATION LOADS

A. Storage Rooms and Building Layout

Storage Space

In the U.S.A., a rough figure of 3 cubic feet of storage space per bushel basket or crate of product is commonly employed to reckon the total space required, in case of packed storage rooms of 10 ft. height of ceiling in a multi-storied building. In India, during hot summer days, sunlight radiation on top of the ceiling may be so high that in spite of enough insulation provided for average conditions the upper part of the storage space may show higher temperatures than the average prevailing elsewhere. Hence a minimum of 3 feet open space for air movement may be left above the top of the stacks. Further, for leaving about 6 ft. aisles between stacks, additional cubic space is required. Hence we may take about 4 cu. ft. space per bushel of potatoes.

A bushel of potatoes having 1-1/2 in. average size would weigh about 60 pounds. Hence total cubic space for 1000 tons storage capacity would be

$$1000 \times 2000 \times 4/60 = 133{,}000 \text{ cu. ft.}$$

For flexibility of loading operations and providing for slack trade during certain seasons, it is better to have more than one room, say, three rooms of same dimensions.

In that case, capacity of each room would be 333 tons with cubic space of $\frac{133000}{3}$ = 44400 cu. ft. per room, say, 45,000 cu. ft.

Room Dimensions

From the standpoint of economical use of insulation, space and machinery and for minimizing the transmission loss through walls, ceilings and floor for a given cubic space, the ideal shape of storage room is a cube. But the height of the ceiling is limited to the extent up to which product containers could be stacked easily and economically. Using a telescopic high lift pallet truck which could stack tiers of potato crates one above the other upto 12 ft. and allowing 3 feet free air space on the top as already suggested, the height between the floor and ceiling would be 15 feet.

Again in order to utilize maximum ground space for product stacking and to achieve easy flow of material for loading with ample aisle spaces, the floor area may not be square, as it should be for minimum transmission heat loss. We may therefore choose to keep the floor area to rectangular dimensions 70 ft. x 43 ft., so that total cubic space may be 70 ft. x 43 ft. x 15 ft. = 45,150 cu. ft., nearly what is required. Taking approximately one ft. thickness of walls and ceiling, the outside dimensions would be about 72 ft. x 45 ft. x 16 ft.

Containers

Containers are of vital importance in the satisfactory storage, marketing and distribution of commodities. In the United States, containers are standardized by law. In view of climatic conditions of India and long distances over which much product has to be transported before it is stored at a terminal storage or marketed, the question of suitable containers assumes special importance. A standardized container would lower the cost of packages, tend to reduce damage in transit or unnecessary handling and provide definite basis for sale at the terminal storage house or market. If the same containers are used as those in storage, this may even minimize further handling of the product at the storage house, if the potatoes do not need any sorting or grading. Work is being carried on on the standardization of containers in India on information collected by the Empire Marketing Board.

Containers can be divided into four principal classes (5): baskets, crates and boxes, barrels and sacks. Crates and boxes are grouped together because in the trade consistent distinction between them has not been recognized. Whether constructed rotary-cut or sawn material, the ends, sides and bottoms may be solid, paneled or slatted. Crates are now bound together by means of encircling wires stapled to the units and closed with interlocking wire loops. Sacks are generally made of jute or hemp.

Though for table stock, bin storage or storage in jute bags (sacks) would be permissible; for potato seed, special care and handling is necessary. Investigations carried out by Werner (43) show that crate storage is more effective in preventing weight loss than sack storage, principally for decreasing the amount of decay, which probably results from the less humid atmosphere immediately around the tubers in crates during spring and summer seasons. It was found that there was less sprout growth and rot with crate storage than with sack storage, the difference being probably due to difference in aeration. In view of this air circulation needed during storage, wooden crates are perhaps the only containers suited for the purpose.

A well filled "weight" bushel crate 12-5/8 in. height, 12-7/8 in. width and 16-3/8 in. length (12 in. x 12 in. x 15 in. inside) contains 60 lbs. net potatoes of uniform size. Another crate 12-3/4 in. by 14-3/4 in. by 18-1/4 in. (approximate outside dimensions) would take about 80 to 85 lbs. of potatoes of 1-1/2 in. to 2 in. size. As standard weight in India is a mound and is equivalent to 82-2/7 lbs., the latter size of crate would be well adopted for storage and marketing in that country. The empty container would weigh approximately 10 lbs. The gross weight of the product in the container would be in the neighborhood of 90 lbs., which is not too heavy for an Indian laborer to handle.

One more point to consider in adopting the crate for storage of seed potatoes is the stacking of the containers one above the other. In this respect, one consideration is to leave a little air space between two containers, stacked one above the other, allround and further, the stacking should be well stable. Keeping this in mind, the standard crate with four triangular cross sectional supports, as shown in Fig. 12, could be designed and adopted. This crate has an advantage that no strapping between tiers is necessary and bottom of wodden supports of the upper crate rests on top of these supports of the lower one and leaves ample air space between the tiers. These supports have ample bearing area for bottom most crate to support the total load up to the top.

It is intended that the manager of the storage house will take all possible means to use the same type of crates in his storage houses for uniformity and stability. He may either provide his own containers or encourage customers to use same type of standard dimensions. If irregularities should occur, suitable means of stacking the product may be found, depending on local conditions.

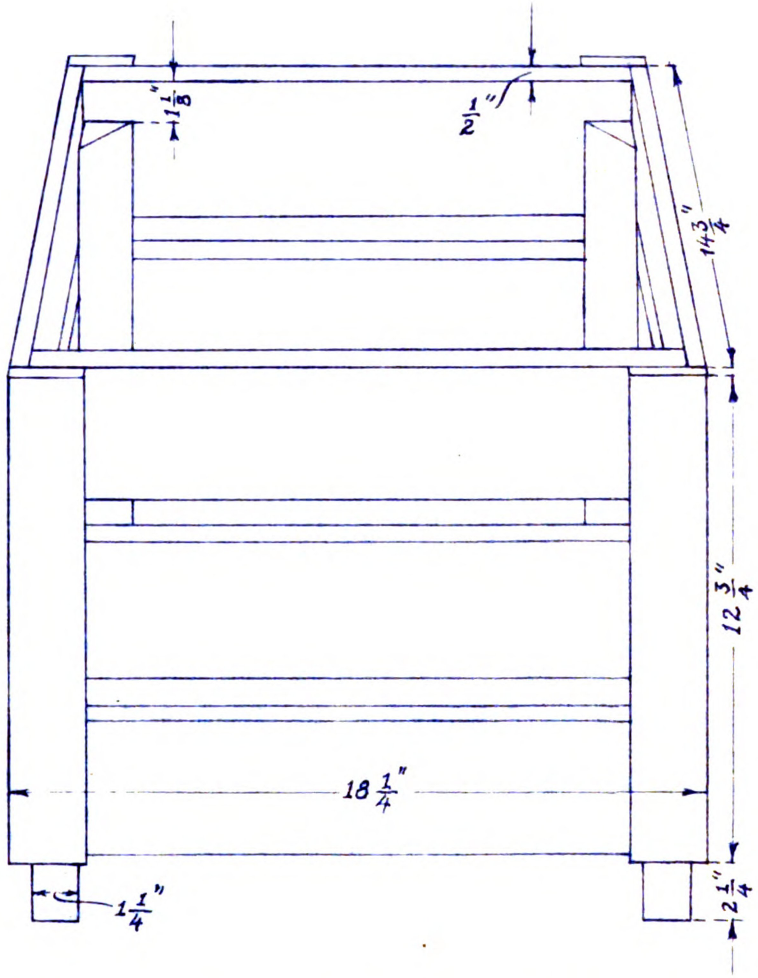


Fig. 12. Standard 80 lbs. Crate for Seed Potato Storage.

Crate Storage

A slat floor raised 2 to 4 in. from the main floor should be provided in order to allow air circulation under the crates, which should be stacked in even rows to the proper height. A satisfactory method is to stack each bottom row of crates on two 2 in. x 4 in. strips or racks set on edge, parallel to each other, and about one foot apart, thus permitting ventilation under the crates. In this way, the pieces act as a substitute for a slat floor.

We have already mentioned keeping ample space between crates' rows and tiers. Allowing 2-1/2 in. all-round, the overall space occupied by each crate will be 19-1/2 in. in length, 16 in. width and 14 in. height. These overall dimensions will be taken to have satisfactory stacking arrangement on the floor space.

Number and Size of Doors

The more doors there are in the storage rooms, the greater is the cost of construction and the greater the air infiltration and consequent difficulties with temperature and humidity control. The number of doors should be reduced to a minimum consistent with the practical management of the storage. In the single-story house, more than one door-way to a room is seldom needed.

The minimum size for an entrance door is 3-1/2 ft. by 6 ft., though 4 ft. by 6-1/2 ft. is more desirable for

these large rooms. One more consideration in the size of the door is that trucks with overall collapsed height and width of pallets for loading the product can easily pass through the doorways.

Location of Room Coolers

For uniform air distribution and air velocity in the product zone, favorable location of unit coolers is very important. Irregular shaped spaces, beamed ceilings, columns and various methods of stacking will often complicate the problem of air distribution. In such cases, proper air distribution and velocities can often be secured by the use of more than one units located at suitable places.

In general, the cooling units should not be installed in front of, or close to door openings, as moist warm air will be drawn directly into the unit each time the door is opened, causing excessive frost, loss of refrigeration capacity and load.

In the large space of regular shape requiring more than one cooling unit, the same care in locating the units in relation to the doors must be observed. With a door at the end of a side wall in a common vestibule for all the three storage rooms, care should be taken not to place the units back to back in the center of the space; but side wall location as shown in the room layout in Fig. 13 is to be preferred, so that they blow across the width of the room.

Stacking Arrangement

Lines are ordinarily painted on the floor of the storage rooms to indicate the spaces for placing rows of boxes and to facilitate even stacking. Stacking packages in contact with outside walls or floors should be avoided, as there is some heat through conduction on hot summer days that affects the temperature of the product in outside or bottom packages. When crates are being stacked, spacing between the walls and the packages may be insured by using side rails or by fastening wooden triangular planks to the floor around the walls of the room. A space of about 6 inches allround would not only safeguard against the outside heat conduction but would also provide passage for convection currents for cooled air.

Number of crates, each carrying 80 lbs. of potatoes, in a room of 33 tons storage capacity is

$$\frac{330 \times 2000}{80} = 8250$$

and piling 10 tiers deep would give 825 crates occupying floor space. Taking consideration of all the factors for good spacing, location of unit coolers and leaving 6 ft. aisles, the stacking arrangement of 825 boxes would be as shown in Fig. 13. It will be noted that the total number of crates on the floor area are 6 x 25 + 2 x 14 x 20 + 6 x 20 = 830 which is more than what is required.

Building Layout

The rectangular space 9 ft. x 8 ft. on right hand

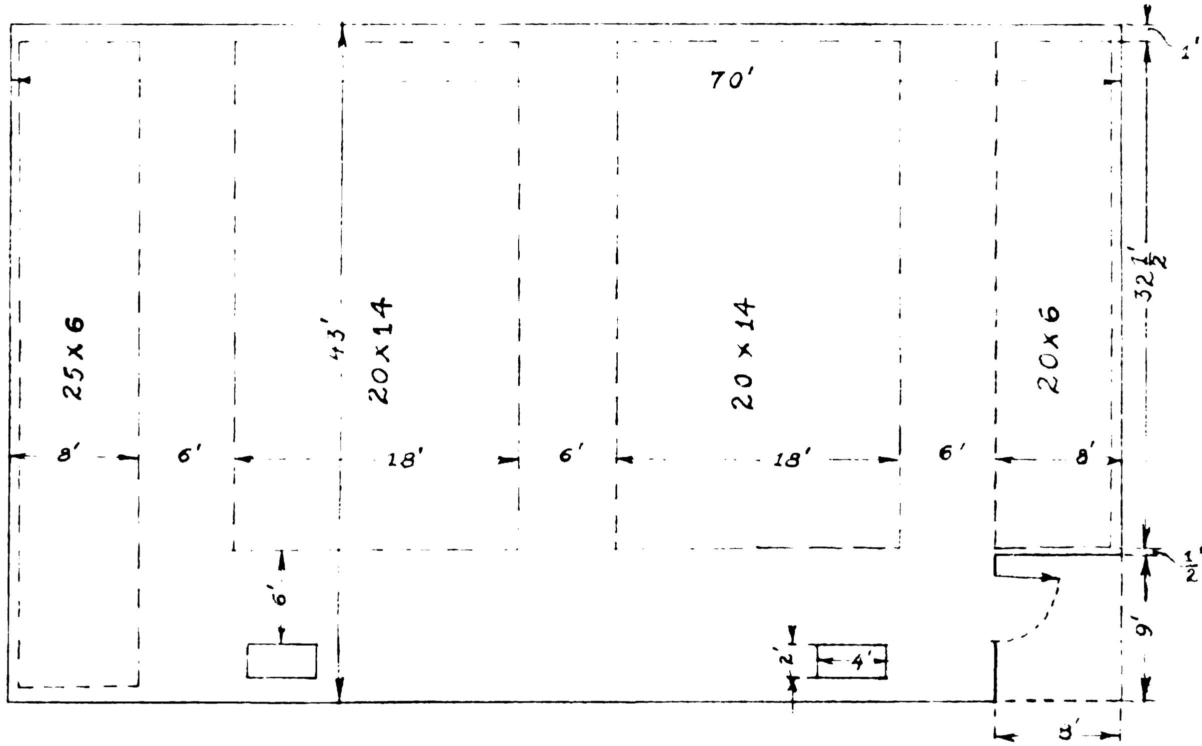


Fig. 11. Storage Room Layout showing location of Unit Coolers and Stacking Arrangement.

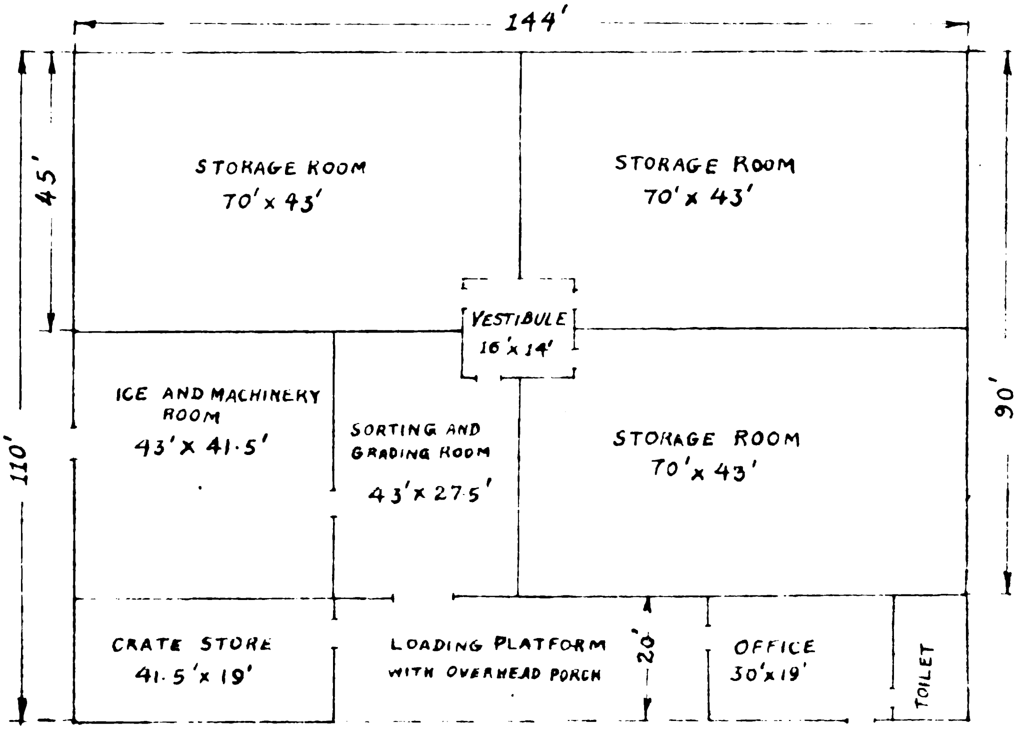


Fig. 1[illegible]. Tentative single line Building Layout.

bottom corner of the Fig. forms one-fourth part of a common vestibule or antiroom 16 ft. x 14 ft. inside dimensions with 4 refrigerator doors 4 ft. x 6-1/2 ft. opening outwards shown in tentative single line building layout in Fig. 14. The building layout is prepared to give surface area of walls, ceilings and floor for finding refrigeration load due to heat leakage. Single story building is preferred due to ease of handling and because ground space is not a big consideration in average sized town.

B. Design Data for Heat Gain Estimate

<u>Design Room Conditions</u>

The design temperature and humidity and the maximum storage period has already been decided upon in the previous discussion on physiological changes and optimum conditions of storage. They are:

Temperature for first 3 to 5 months - 38°F.
" pulled down for longer storage to 36°F
Range of Humidity - 85 to 90%
Period of Storage - 5 to 7 months

Room humidity for seed potato storage is quite important and must be maintained within the permissible range. Further, maximum air motion in the rooms may be 150 fpm. This is the recommended velocity of air in the product zone and is based on experience and judgment.

This may be exceeded during the chilling process to promote rapid temperature reduction.

Outside Conditions

During loading period--middle of April to middle of May--average monthly temperatures prevailing in northern India as collected from the Report on Metrology of India (See table in Section III C. of the manuscript) is about 85°F. Dry-Bulb and about 68°F. wet-bulb. Wind velocity will not affect the storage conditions as doors are well within the building with a vestibule and the walls are properly insulated against leaks. The soil temperature in tropics during summer may be taken 75°F. The mean maximum temperature during the month may occur at afternoons and is about 98°F. and minimum mean occurring in early mornings is about 72°F.

Heat Transfer of Walls, Ceilings, etc.

Compositions of walls, ceilings, etc., has already been discussed in Section III C. The values of overall coefficients of heat transfer found are:

Outside Walls:	U = 0.065 Btu/hr./sq.ft./deg.F.
Roof:	U = 0.056 "
Floor:	U = 0.066 "

Combined Chilling and Holding

This storage problem is a case of combined chilling and holding product. That is, after product is chilled,it is held in the same room for storage for extended periods. Such a procedure makes it necessary to use the same diffuser for chilling and storage simultaneously.

Daily fluctuations in room temperature through a relatively wide range must be extected during loading and accepted. But experience has shown that fruits and vegetables such as apples, pears, potatoes and like are not sensitive to these fluctuations and can be handled in combined chilling and storage rooms without harmful effects to the product. This is so because the loading-in period is of relatively short duration, and the quantity of product in storage becomes an ever increasing amount, producing a flywheel effect that permits little or no disturbance to room and product temperature. Large sized rooms also have favorable effect on combined chilling and storage.

Product Loading

Loading-in will take place during the main harvesting of seed potatoes, from the middle of April until the middle of May or 30 days. The average loading rate for the product for all the rooms would be:

$$\frac{1000 \times 2000}{80 \times 30} = 833 \text{ crates per day}$$

Maximum loading rate may occur on certain days. But in order to have uniform refrigeration load during chilling, the manager of the plant should see that loading fluctuations are not too great which may necessitate installing a larger refrigerating system in proportion to the load factor. Any day to day fluctuations will even out by themselves and will not tax the plant which may be designed for average loading requirements.

Initial Product Temperature

The actual initial product temperature should be determined by survey. Considering, however, that the product will be brought in during morning hours of April and May, when the temperatures are not so high, the average initial product and crate temperatures may both be taken as 75°F., which is a little over mean minimum temperature of the month.

In case loading time is irregular and very frequently the product is brought in at much higher temperatures of, say, 90°F., precooling must be resorted to in a separate room, where there is sufficient spreading space and suitable racks for precooling every day's stock for 24 hours before storing the product in the cold stores the next day.

Chilling Time

The product heat must be removed as quickly as possible before next loading so as to minimize loss of moisture and heat of respiration and perspiration. For potatoes, chilling time for daily load may be taken as twenty hours. Any slight load fluctuations would not affect the daily refrigeration capacity and the chilling can be accomplished within the 24 hours of that day.

Room Temperature during Loading

The room temperature should be maintained at the recommended design value before the hot goods are loaded. Since the product is loaded during morning hours when insulation heat gain is low, there is not much likelihood of room peak temperatures going up very much during peak loads. If the fluctuations are too great the average room temperature during loading should be the temperature recommended. Further, temperature split of air temperature in different parts of room should not be too great, say within 8 deg. F. during loading.

Chilling Rate Factor

Chilling rate factor is generally introduced in the calculation for product cooling load to recognize the unequal distribution of the load during the entire chilling time of the day. Because of the initial high temperature difference and high vapor pressure difference between the product and the room air, the load tends to concentrate in the early part of the day.

The chilling rate factors are generally based on experience and on test and will vary with the ratio of loading time to total chilling time. If it is assumed that the cooling rate during the first half of the chilling period for potatoes is 25 percent greater than the average rate for the entire period of the day, the refrigeration load for the product cooling required during the first half of the chilling period would be 1.25 times the average load based on average rate of cooling. Chilling rate factor is generally taken as the reciprocal of 1.25, that is it will be 0.8 for potatoes.

Specific heat of Potatoes

Specific heat of the potatoes in the temperature range of 75° to 35° will be required for finding the field heat of the product to be removed. Different sources give different values of this coefficient. Following are some:

Carrier (6):	c = 0.860
ASRE Date Book (2):	c = 0.770
Veneman (42):	c = 0.792

Average value will be about 0.81.

If amount of water in fruits or vegetables is given as x percent and the rest is solids with specific heat of 0.2, the specific heat of commodity would be

$$\frac{x}{100} + 0.2\left(1 - \frac{x}{100}\right) = \frac{0.8x}{100} + 0.2$$

For potatoes, average amount of water has been found (2) to be 77.8 percent. Substituting, we get

$$c = 0.822.$$

Figures found from the sources are so divergent that the average of them, viz. 0.81 would only be satisfactory.

C. Heat Load During Loading

Sources of Heat Gains

For calculating the heat gains for refrigeration, all of the heat sources must be taken either fairly accurately or by some simplified method. The sources of heat in combined chilling and storage rooms will have to be considered separately for refrigeration load during loading and during storage period. The requirements of machinery and pipe sizes will depend on peak loads which will occur during loading season. Hence we shall first consider heat sources during loading. They are:

(1) Heat due to Insulation

(a) Sunlight gain, (b) Transmission gain.

(2) Infiltration through door.

(3) Internal Heat

(a) Product field heat, (b) Reaction heat or heat of respiration, (c) People, (d) Fans, (e) Lights

Sensible and Latent Heats

The above heat gains are again classified as sensible heat gain, which contributes to the rise of temperature of air from cooling coils, and the latent heat gain, which adds to the moisture content of the air due to evaporation of water from the product inside or by infiltration. Item (1) above is purely a source of sensible heat gain. Item (2) is partly sensible and partly latent heat gain depending on the temperature and moisture content of outside air. Items 3 (b) and (c) would consist of both sensible and latent heats of the product and persons working, and the rest is all sensible heat.

Heat due to Insulation

For estimating this refrigeration load for cold storage rooms, fundamentally the 24 hour average temperature should be used (9) instead of the normal design dry bulb temperature used in air conditioning jobs, which is about 10°F higher than this average. Daily average temperature for the months of April and May have been taken as 85°F dry bulb. This dry bulb temperature would be used for transmission gain through shaded walls. For open walls, 10°F higher would be assumed to allow for sun's radiation on masonry walls of fairly dark color. For flat roof without attic or peaked roof with attic unventilated concrete construction of dark color, 30°F will be added to the daily average for sunlight gain.

Hence, looking to the building layout drawn in the earlier part of this section, we have the following:

Sunlight gain:

Walls on East or West			
135 x 16 x (95 - 38) x 0.065	=	8000	Btu/hr.
Roof			
3/4 x 144 x 90 x (115 - 38) x 0.056	=	42000	"

Transmission gain:

Shaded walls, north and south			
2 x 144 x 16 x (85 - 38) x 0.065	=	14100	"
Shadded wall, West of Room C			
45 x 16 x (85 - 38) x 0.065	=	2200	"
Floor			
3/4 x 144 x 90 x (75 - 38) x 0.066	=	22000	"
Total insulation gain	=	88300	"

Door Infiltration

For long storage warehouse with rooms protected by a vestibule through each 3 ft. swinging door, and assuming outdoor temperatures 95°D.B. and 78°W.B., relative humidity in the room 85% and conditions in the vestibule as 40°D.B. and 75% relative humidity, infiltration figures are as follows (8):

Cfm of Air infiltration	12.
Sensible heat in Btu/hr.	850.
Latent heat " "	530.

For 4 ft. door width, these figures will be 16 cfm., 1133 Btu/hr. and 700 Btu/hr. respectively.

For all the three rooms:

Infiltration sensible heat = 3400 Btu/hr.

" latent " = 2100 "

Product Load

Product heat load consists of both (a) product field heat and (b) heat of respiration. It is difficult to determine this heat to be removed in cooling the product as it depends on many varying factors like weight of the product loaded or lying already in the storage, the rate of cooling, specific heat of the product, its initial and final temperatures t_1 and t_2, and also rate at which it produces heat by respiration. This total product load will be maximum on the last loading day of the season, as not only product of last day will be giving out its field heat and its heat of respiration at the prevailing temperature, but also the product already cooled down will be generating its heat of respiration at the rate corresponding to the temperature of storage. Hence heat load calculations will be done for the last day of the loading period.

Field Heat

The product field heat is given by the formula

$$\frac{W \times C \times (t_1 - t_2)}{T \times L.F.},$$

where, W is the average loading rate of product in lbs. per day and is 833 x 80 lbs;

C is the specific heat of commodity in the temperature range, = 0.81;

$(t_1 - t_2)$ is the temperature difference of the product before and after cooling and is (75 - 38) = 37 deg. F.;

T is the chilling time = 20 hrs.;

and Load Factor (L.F.) is found to be 0.8.

Hence, field heat $= \dfrac{833 \times 80 \times 0.81 \times 37}{20 \times .08}$

$= 124{,}700$ Btu/hr.

It is highly improbable that on the last day of the loading period the rate of loading product will be anything higher than the average, and hence the value of W taken is quite safe value.

Since wooden containers also cool down from 75° to 38°F. and their specific heat may be assumed to be 0.5, the heat load for containers would be

$$= \frac{833 \times 10 \times 0.5 \times 37}{20 \times 0.8} = 9600 \text{ Btu/hr.}$$

Reaction Heat

Reaction heat or heat due to respiration of product would consist of heat due to product which is already cooled down to a temperature of about 38°F and for the product stored on the last day for temperature range of 75° to 38° F. The rate of heat evolution due to respiration increases with the product temperature.

Hence, although it would be easy to find the heat of respiration for the product cooled down to a constant temperature 38°F., the same is not true for fresh product, as the cooling requires time during which heat by respiration varies. Hence, average value of the rate should be taken.

The figures have been obtained on heat of respiration for Irish Cobbler variety by R. C. Wright and T. M. Whiteman (34) which are as follows:

Temperature, Deg. F.	Rate of Heat of Respiration (Btu. per ton per 24 hrs.) Range	Average
32	440 - 880	600
40	1100 - 1760	1300
70	2200 - 3520	2800

Assumption may be made, although it is known to be approximately correct, that the rate of temperature drop at any time during cooling is proportional to the difference between room temperature and product temperature at that time. As a result it will be found that heat produced by respiration during cooling is directly proportional to the length of the cooling period.

Taking average values of rate of heat h for the temperatures t, the equation of the graph could be found to be

$$h = -80.75t + 4.28t^2 - 0.0364t^3.$$

From this equation, the rate of heat generation at $38^{\circ}F.$ would be found to be 1128 Btu. per Ton per day, or 33.8 Btu. per bushel per day.

Also, average rate for temperature range 75° to $38^{\circ}F.$ would be

$$= \frac{1}{75 - 38} \int_{38}^{75} h\, dt, \text{ where h as above value;}$$

$$= 2180 \text{ Btu. per Ton per day.}$$

$$= 65 \text{ Btu. per bushel per day.}$$

Since 33 Tons are loaded daily, the total reaction heat on the last day

$$= \frac{(1000 - 33) \times 1128 + 33 \times 2180}{24}$$

$$= 48500 \text{ Btu./hr.}$$

Heat due to People

If 2 persons are working in each room at a time during day and each gives off 610 Btu/hr. sensible heat and 110 Btu/hr. latent heat per hr., total for all the rooms would be

sensible heat = 3700 Btu/hr.

latent " = 700 "

Fans

Total fan H.P. of the room coolers working continuously for 24 hours may be taken as a "guess" on the basis of about 30 Tons refrigeration load, as the sizes of the units are not known at this stage. This figure can be rechecked and corrected, if necessary, upon equipment selection. Taking total fan motors H.P. to be 10 and assuming 88 percent motor efficiency for finding out input, the sensible heat load due to Fans would be

$$\frac{10 \times 2545}{0.88} = 29000 \text{ Btu/hr.}$$

Lights

Electric lights in cold storage are usually placed at the rate of one watt per sq. ft. of floor area. A watt would generate 3.4 Btu/hr. If the storage rooms are lighted for 10 hrs. during the day, the sensible heat load due to lights for all the three rooms would be

$$= 3 \times 70 \times 43 \times 3.4 \times \frac{10}{24} = 12,500 \text{ Btu/hr.}$$

Sub-Totals

From the computations done so far, we find that total sensible heat becomes:

Insulation heat	88300	Btu/hr.
Door infiltration	3400	"
Field Heat (Product)	124700	"
Field Heat (Containers)	9600	"
Reaction Heat	48500	"
People	3700	"
Fans	29000	"
Lights	12500	"
Sensible heat - Sub-total	319700	"

A safety factor, depending on the accuracy of the various items making up the load and the accuracy of the information on building layout, but never exceeding 10 percent, is usually added to this sensible heat subtotal, as it is a big item. Adding 5 percent in this case, sensible heat total will be = 335700 Btu/hr.

Latent heat due to air infiltration and persons has been found and is = 2100 + 700 = 2800 Btu/hr.

Product Latent Heat or Heat of Perspiration

Now, the total product load given under sensible heat total includes the heat required for evaporation of moisture given out by the product. During product chilling, moisture is released from all fruits and vegetables, and this release is highest at the beginning of the chilling duration due to large vapor pressure difference between the fresh product and the room air, when the product internal temperature is high; hence chilling rate factor will apply to the freshly loaded product. However, moisture release continues even after the product is chilled. While value of latent heat load for cooled product at 36^{o}F. is given as 0.5 Btu. per lb. per 24 hrs. (6), the data for fresh potato stock during chilling time is not available. Comparing the commodity with others of similar behavior, such as, beets, carrots, for which such data is available, it would be reasonable to assume the product latent heat of 15 Btu. per lb. of potatoes for entire chilling time, i.e. 20 hours.

Since on the last day of the loading period, 33.3 tons of fresh product undergo cooling and give out moisture with absorption of latent heat at 15 Btu/lb/20 hrs. the rest (1000 - 33.3) tons have already cooled down and give out moisture only slightly, i.e. with latent heat load of 0.5 Btu/lb/24 hrs. Therefore total latent heat load required

$$= \frac{2000 \times 33.3 \times 15}{20 \times 0.8} + \frac{2000 \times (1000 - 33.3) \times 0.5}{24}$$

$$= 102500 \text{ Btu/hr.}$$

Now, this moisture gain from product is credited to the total product load to find the net room sensible heat load. The reason is as under:

The calculation of the product heat load indicates the total heat removed in chilling the product from its initial to the final state. However, since the chilling process for exposed product is accompanied by moisture release, the evaporation of this moisture from the product surface chills the product also. The chilling effect is exactly equivalent to the latent heat gain to the room, which therefore becomes a credit to obtain the net sensible product cooling load on the refrigeration system.

<u>Grand Total Heat and Sensible Heat Factor</u>

Deducting this latent heat load from the room sensible heat sub-total, we get

Net Room Sensible Heat = 3197000 - 1025.00

= <u>233200</u> Btu/hr.

And adding this latent heat to that already found for other sources, we get

Room latent Heat = 2800 + 102500

= 105300 Btu/hr.

Therefore Grand Total Heat (G.T.H.)

= 233200 + 105300

= <u>338500</u> Btu/hr.

= 28.2 Tons.

$$\text{Sensible Heat Factor (S.H.F).} = \frac{\text{Net Room Sensible Heat}}{\text{Grand Total Heat}}$$

$$= \frac{233200}{338500}$$

$$= \underline{0.688}$$

D. Heat Load during Storage

<u>Sources of Heat Gain</u>

Let us consider the same sources of heat gain during storage from May - Oct, viz:

(1) Heat due to Insulation

(2) Infiltration

(3) Internal Heat

Heat due to Insulation will remain, but may vary slightly due to variation in outside air temperature during the summer months. We may, take it to be the same as during loading, as temperature variations are not too great.

Infiltration loss may remain but will be considerably reduced due to less service factor due to lesser use of the storage rooms. We may take it to be half the previous value, i.e. sensible heat 1700 Btu/hr. and latent heat 1000 Btu/hr.

Among the various sources of internal heat, product and containers' field heat will completely vanish. Reaction heat will be for the product already cooled down to $38^{o}F$. and hence will be

$$\frac{1000 \times 1128}{24} = 47000 \text{ Btu/hr.}$$

Fans H.P. will be the same as they will continue to run for 24 hours, hence will give the same sensible heat load. But persons working may not remain so long in the rooms during storage and lights will not be on for all the hours of the day. These heat values may also be taken half the previous ones, i.e. sensible heat due to persons to be 1800 Btu/hr. and latent heat to be 300 Btu/hr, and sensible heat due to lights to be 6300 Btu/hr.

Sub-Totals

The above values for sensible heat gain may be added as follows:

Insulation Heat	88300	Btu/hr.
Door infiltration	1700	"
Reaction Heat	47000	"
People	1800	"
Fans	29000	"
Lights	6300	"
Sensible Heat Sub-total	174100	"

Adding 5 percent as before as a safety factor, we get sensible heat total as 182800 Btu/hr.

Product Latent Heat

During storage, all the 1000 tons of seed potatoes are at temperature of 38°F. and give out only slight moisture, which takes 0.5 Btu/lb./24 hrs. of latent heat for evaporation. Hence product latent heat will be

$$\frac{1000 \times 2000 \times 0.5}{24} = 41700 \text{ Btu/hr.}$$

G.T.H. and S.H.F.

Deducting this latent heat from room sensible heat total, as before, we get

Net Room Sensible Heat = 182800 - 41700

= 141100 Btu/hr.

And adding this latent heat to that for other sources, Room Latent Heat = 1000 + 300 + 41700

= <u>43000</u> Btu/hr.

There, Grand Total Heat (G.T.H.)

= 141100 + 43000

= 184100 Btu/hr.

= <u>15.34</u> Tons.

Sensible Heat Factor (S.H.F.)

$$= \frac{141100}{184100}$$

= <u>0.767</u> .

V. REFRIGERANTS AND REFRIGERATING SYSTEM

A. Primary Refrigerant

Preliminary

We have seen that the total maximum room refrigeration load for loading season is 28.2 Tons. The refrigeration system will be installed to do this maximum job at certain time during its entire period of operation and hence should be of this capacity.

But the cold-storage installation is combined chilling and storage process. During storage the refrigeration load is considerably reduced; and as found in the preceding section is only 15.33 Tons, a little over one half the original value. Hence, we shall have to run the system at reduced capacity. For capacity control, multi-compressors' system may be employed with three compressors, so that during storage, two of the compressors may work at reduced capacity required by rooms load and the third compressor may lie as a standby. But, as already suggested, the economical way is to run an ice plant of sufficient capacity simultaneously during holding season, i.e. during May to October, when there is heavy demand for the ice in India. As the complete installation is to run only during the summer until the next planting of seed potatoes (unless cold-storage is utilized for other commodities and run all the year round), the major repairs or overhauling of the plant can be done during the winter months, as is usually the case in all seasonal factories. Any temporary trouble in one of the compressors during the running of the plant in summer may only halt the manufacture of ice for some days. The cold storage rooms can still very safely be maintained at desirable temperature conditions throughout the storage period, as it is highly improbable that more than one machine may go out of order at a time. Being a small plant it is not necessary to have a compressor always as a standby.

General Properties of Refrigerants

The choice of primary refrigerant for the system depends on its physical and thermodynamical properties, so that the system may give efficient and trouble-free service. These properties involving numerical values are tabulated in the accompanying table for more common refrigerants. In a ddition to the above physical properties, the refrigerant should be inert and stable during the working range, non-corrosive, non-toxic and non-irritant. For good heat transfer in refrigeration equipment, it should have high thermal conductivity and high film coefficient of heat transfer. It should not mix considerably with oil. It should be easily detected in case of leaks and lastly, it should not have properties injurious to goods, foods, etc., stored.

PROPERTIES OF COMMON REFRIGERANTS

(Standard Ton Conditions)

	Ammonia	Freon-12	Methyl-Chloride	Sulpher Dioxide	Carbon Dioxide
Physical:					
(1) Boiling Point (Sea Level) deg.F.	-28.0	-21.7	-10.7	14.0	-108.4
(2) Freezing Point, deg.F.	-107.86	-252.4	-144	-98.9	-69.9
(3) Critical Pressure, psi.abs.	1651	601	640	1141.5	1069.9
(4) Critical Temperature, deg.F.	271.2	233.7	421.0	314.8	87.8
(5) Sp.Gr.of Liquid at B.P.	0.684	1.48	1.00	1.357	1.56
(6) Density of Vapor at 5°F., lbs/cu.ft.	0.1227	0.6735	0.22	0.1557	3.74
(7) Sp.Heat of Liquid, 5° - 86°F.Aver.	1.12	0.23	0.38	0.34	0.77
Thermodynamical:					
(8) Suction Pressure at 5°F., psi.abs.	34.27	26.51	20.9	11.81	331.9
(9) Discharge " at 86°F., psi.abs.	169.2	107.9	96.0	66.5	1043.0
(10) Temp.of Compression, °F.	222.7	100.0	165.5	202.9	176.1
(11) Refrg.Effect, Btu/lb.	474.45	51.07	148.7	141.37	56.69
(12) Pounds Ref.per min.per ton	0.4216	3.916	1.345	1.414	3.528
(13) Theast.1 H.P. per ton	0.988	0.997	0.97	0.966	1.98
(14) Coeff. of Performance	4.77	4.72	4.9	4.87	2.4
(15) Cu.in.of liq.ref.per min. per ton	19.6	83.9	41.7	28.9	162.8
(16) Cfm Piston Displacement-per ton	3.436	5.815	6.091	9.084	0.943

Source: ASRE Refrigerating Data Book; Kinetics Chemicals, Inc., and other publications.

Choice of Refrigerant

In cold storage application, physical properties of refrigerant like non-toxity, non-irritant properties, etc., are of minor importance, as the space refrigerated is not normally inhabited by persons. Only physical factor of considerable importance is that refrigerant should not have injurious effect on product stored, if by chance any leak occurs. From that point of view ammonia is not permitted in direct expansion system for storage of fruits and vegetables such as potatoes. Thermodynamical properties are of primary consideration in cold-storage, as they determine the thermal efficiency of the system for economical running.

It will be observed in the table for properties of refrigerants that, with the exception of carbon dioxide, all the refrigerants have almost the same coefficient of performance or refrigeration capacity per theoretical indicated H.P. Hence it is immaterial which of those refrigerants is employed, provided its use is consistent with other factors involved in a particular installation. One of the most important of these factors to be considered, particularly in larger installations, is the piston displacement per ton of refrigeration capacity. This should be appreciably low so that small compressing units should do a given job. For that reason, Freon refrigerating units are employed to a maximum of about 25 ton refrigerating systems. Beyond that Ammonia systems are invariably used.

The second most important factor is the working pressures range of compressor units for different refrigerants. From this point of view, carbon dioxide is easily rejected which operates on very high pressures and would require very heavy and cumbersome machinery. Freon-12, Methylchloride and Sulpher Dioxide have very low operating pressures and could afford to have very light machinery. If we choose Freon-12 as the possible refrigerant for average jobs from these three refrigerants (being also non-toxic, non-irritant and non-explosive), we can run the refrigerating equipment at low pressure and with possible high piston speeds to the extent of say 500 ft./minute or main shaft speed of 1500 rpm. with forced feed lubrication. This would considerably reduce the size of the compressors required. But, although higher speed may give higher refrigeration capacity in almost the same proportion, it is cause for compressor life to reduce in the inverse proportion. Hence, low operating pressures do not have very much advantage except from the point of view of initial cost and frictional losses which will reduce the running cost. Hence, in larger installations, as explained, piston displacement per ton rather than low operating pressures is given prior consideration. Ammonia works at higher operating pressures, but these pressures are quite tolerable to run the plant without much losses. However, the running speed usually does not exceed 500 rpm.

Now, ammonia, as already stated, has strong objection to its use in cold storages with direct expansion systems. Ammonia is not safe from contaminating the product and causing damage in case of a leak. Hence, if ammonia must be used for refrigerating system, the latter should run with indirect brine system. The cold storage plant for seed potatoes under consideration is of about 30 tons capacity. It is therefore a border case between whether ammonia or Freon-12 may be used as a refrigerant.

Many food storage installations and Farm storages of similar capacity in the U.S.A. actually work on Freon-12 because of the latter being adopted to more convenient and efficient direct expansion system and further because Freon-12 system can easily be operated with controls automatically. But in India, there are other considerations which require ammonia to be more suitable and desirable refrigerant in the installation under discussion. These considerations are numberated below:

(1) As already decided, ice plant will be operated during storage or holding period for which indirect brine system for freezing ice is essentially needed. Ice plant will work during the hot summer months when demand for ice is very pressing. Though brine system of refrigeration is costly both from the point of view of initial extra expenditure and running efficiency (as two heat transfers are involved--one from refrigerant to brine, the other from brine to refrigerated space with resulting heat loss and low refrigerant temperature), but the running of the ice plant in combination with cold storage job will make the whole installation profitable. Hence, ammonia refrigeration with indirect Brine system both for cold storage space and ice plant will be desirable.

(2) Freon-12 is available in India through imports from U.S.A. Hence, its supply is not always guaranteed and cannot always be relied upon. During the last war, Freon used to be imported in India under lend-lease arrangement and distributed only through recognized dealers to essential end-users classified in certain categories depending upon a system of essentiality and priority. It is highly improbable that the manufacture of ice would be classified an essential service, hence, during storage, limited supply of Freon-12 would be available for cold storage

only and the ice plant may remain stopped, if such exigencies occur. Temporary switching over to Methyl Chloride for systems designed for F-12 is permitted but not recommended. To run such a plant on Methyl Chloride for several years is certainly not desirable.

(3) Ammonia is easily available in India at much cheaper rates. Ammonia is also well adapted for ice plants.

(4) With ammonia as primary refrigerant and indirect brine system, the temperature of the refrigerating space is generally more easily controlled. It may also even out slight load fluctuations between the refrigerated space and ice tank.

Hence, ammonia is chosen for the present application. It will be self-contained in one room by using indirect brine system and with proper isolation of the compressor room from the refrigerated space.

Properties of Ammonia

Ammonia (NH_3) is colorless, has sharp pungent odor and an alkaline taste. It has a specific gravity of 0.6382 at 32°F., boils at -28°F. at atmospheric pressure and solidifies to a white crystalline mass at -107.8°F. Gas is commercially available in steel cylinders containing 25, 50, 100 or 150 lbs. net, into which it has been compressed and liquified. Some physical and thermodynamic properties of anhydrous ammonia are already given, and further data are given in National Bureau of Standards circular 142.

Ammonia is explosive in proportions 13.1 to 26.77 percent in air. At ordinary temperatures and pressures ammonia is stable; however, exposure to temperatures of 400° to 500° F., such as can result from lack of cooling water in the condenser and the compressor heads, may cause some breaking down or decomposition.

Moist Ammonia, being a strong alkali, will attack copper, brass, zinc and aluminum. Iron, steel and lead are the metals normally used in contact with it. Ammonia is very soluble in water. This property is utilized in the event of bad ammonia leak; it is possible to absorb the gas and prevent severe damage, if a stream of water can be trained on the leak.

Ammonia is not poisonous; it is, however, a powerful irritant upon the mucous membranes and any part of the skin which happens to be moist. The chart below indicates the physiological response to inhaling various concentrations of ammonia in air (18):

	Parts per Million by volume
Least detectabhe odor	53
Least amount causing immediate irritation to eye	698
" " " " irritation to throat	408

(Continued)

	Parts per Million by volume
Least amount causing coughing	1720
Maximum concentration allowable for prolonged exposure	100
Maximum concentration allowable for short exposure (1/2 to 1 hr.)	300 - 500
Dangerous for even short exposures (1/2 hr.)	2500 - 4500
Rapidly fatal for short exposure	5000 - 10000

It is a fortunate thing that ammonia has a pungent odor. For this reason there is no possibility of a worker unknowingly exposing himself to dangerously high concentrations. Ammonia in small concentration acts as a powerful heart stimulant.

The cost of ammonia is low. Its dangers, which are only confined to compressor romm in an indirect system, can be minimized by careful attention to maintainance and repair and by safeguarding against accidents.

Ammonia leaks are easily detected by the odor. Practical tests to locate the exact source of leaks may be conducted by the use of ammonia sensitive (phenol phthalein impregnated) paper, or sulphur-impregnated stick or cotton cord. White smoke is formed when leaks are present.

Ammonia Cylinders

Anhydorous ammonia is commercially available and

generally shipped in steel cylinders containing normally 50, 100 or 150 lbs. net, and tested to withstand a working pressure of at least 300 psi. Cylinders may be filled to only 54% of the water capacity of the cylinder. Fig. 15 indicates the construction of the valve end of an anhydrous ammonia cylinder. When the dipper pipe is pointing down, as shown, the cylinder will discharge liquid ammonia. If cylinder position is reversed, the ammonia will be discharged as gas.

Ammonia cylinders should never be connected to an ammonia refrigerating system except when the system is being charged or drained.

If it becomes necessary to withdraw ammonia from a refrigerating system into cylinders, great care should be taken to avoid overcharging such cylinders. The safest plan to pursue is to limit the amount of liquid which is withdrawn into the cylinders to:

Size of Cylinders	Capacity in lbs.
7 ft. long x 12 in. dia.	150
7 " " x 10 " "	100
46 in. " x 10 " "	50
42 " " x 10 " "	40
28 " " x 10 " "	25

Weigh the cylinders before and after filling. If accidentally overfilled, allow excess to excape immediately into water.

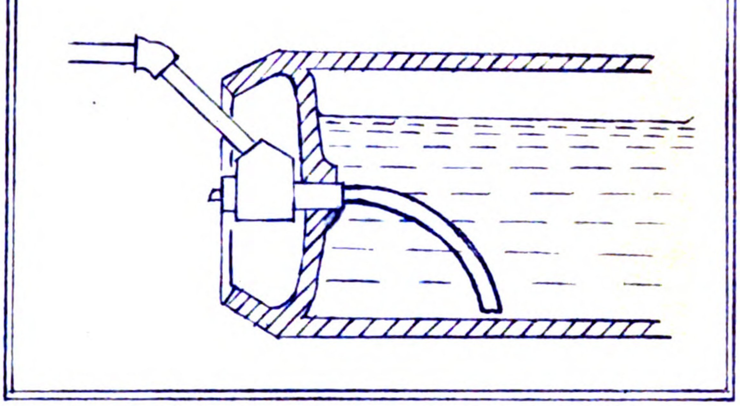


Fig. 15. Valve End of Ammonia Cylinder showing Dipper Pipe.

Where caps are provided on ammonia cylinders, such caps should be carefully attached to the cylinders when they are being returned. Ammonia cylinders should be stored on their sides in a cool dry place.

B. Secondary Refrigerant

Brine

Brine, which serves as a secondary refrigerant in indirect systems, is usually a solution of salt or calcium chloride in water. A good brine for general use should be non-corrosive, have a low freezing point, be inexpensive, have high specific heat and be readily available. No one brine is perfect in all the above respects.

Sodium chloride or common salt is widely used for brine for medium temperatures. It is relatively inexpensive and is easily available in India. It is not unduly corrosive if the solution is kept free from excessive air and if the strength is kept up.

The heat absorbing properties and freezing points of a brine depend largely on the concentration or strength of solution. The properties of the sodium chloride brine within usual operating range are given in the table below. It will be noted that a 20 percent solution by weight has a specific gravity of 1.150, has 1.92 lbs. of salt per gallon of brine, a specific heat of 0.813, and a freezing point of 1.8°F.

% Pure NaCl by Weight	Salometer Deg. 59°F.	Sp.Gr. 59°F.	Sp. Heat 59°F.	Weight of one gallon in lbs.	Pounds per gal.of Na.cl.	Freezing point deg. F.
5	18.2	1.035	0.938	8.65	0.432	27.0
10	37.2	1.072	0.888	8.95	0.895	20.4
15	56.8	1.111	0.847	9.28	1.392	12.0
20	75.2	1.150	0.813	9.64	1.920	1.8
23	86.8	1.175	0.796	9.81	2.256	-6.0 (eutectic)

Source: York Tables and Data. Section 250-T.

Care of Brine

Care of brine systems consists, first in keeping them at the proper strength so that they will not freeze. Brine must be kept non-corrosive. Corrosion in brine systems is usually caused by acidity which in turn is caused by air absorption in the brine. Air may enter through leaky stuffing box or seal of brine pump or it may enter readily if the brine is allowed to splash in air due to agitation in ice tank, and when a brine return pipe does not lead below the level of the brine in the tank. Acid brine should be neutralized by adding limewater or caustic soda solution until the proper pH is obtained--usually between pH 7 and 8.

Addition of certain materials to refrigerating brine has been found to help materially in preventing or minimizing corrosion. It has long been known that chromates inhibit corrosion by brines. Sodium chromate is the best of the known corrosion retarders. Sodium bichromate also has a retarding effect and is usually more conveniently obtainable. It is readily converted to the chromate by dissolving it in water together with suitable quantity of caustic soda.

Refrigerating brine occasionally is contaminated with insoluble particles, either suspended in the brine or settled in the bottom. Most of this sediment, particularly in old brines, is composed of products of corrosion such as iron rust and zinc salts. A smaller portion may have its origin in the use of mineralized water for making up the brine and in the small quantity of insoluble matter originally present in the brine medium. Various products have been recommended as clarifying agents, but their use is questionable for, insoluble matter does no harm so long as it stays suspended, circulates with brine and does not retard heat transfer. Clarification, which consists of adding certain compound, precipitates in the brine, settles readily carrying suspended particles down with it. The net result is more insoluble matter than was originally present. Hence, it is better to keep the insoluble matter in circulation and whatever settles down in brine tank can be removed by withdrawing the brine and taking out cans. Sediments in brine cooler can be removed by emptying the cooler and overhauling the whole equipment. It is usually within the first year or two after a new plant is started in operation that sediment builds up to troublesome proportions. After this initial deposit of sludge is washed out, there is no further appreciable accumulation.

C. Brine Circulating System

A brine system may be of the brine-circulating type, the brine-storage type or the congealing tank type. The first system carries a given amount of brine in circulation under all loads, whereas in the latter two types brine is either stored or congealed in a tank to serve as a reserve for peak loads during the day, as in dairy plants. In our refrigerating system, there are no considerable peak loads in day to day running of the plant, we do not need the latter two brine systems and shall use the brine-circulation system.

With this system, brine in tanks or coolers is reduced to the proper temperature by contact with direct expansion coils, is then circulated by a pump through coils in cold storage rooms or the ice tank, where it picks up heat, and returns through continuous circuit back to the tanks or coolers, which is again cooled by direct expansion coils and recirculated.

In addition to the equipment required with a direct expansion system, the brine circulating outfit requires a brine tank or cooler, a brine pump and brine coils or other final heat exchanger, namely, brine cold diffusers and ice tank. Therefore, as already pointed out, there is an extra transfer of heat involved, resulting in the necessity for ammonia compressors to operate at a lower back pressure than in the case of direct expansion. The heat equivalent of the energy used in circulating the brine, as also insulation loss of brine coolers, must also be added to the net refrigerating effect needed. This also makes the system less efficient.

This system stores no excess brine in the tank or cooler, and when the circulating pump stops, the cooling will cease except for that small amount stored in the pipe coils, etc. The temperature of brine usually rises 4° to 8°F. in passing through a cold room and about $1/2^{\circ}$F. in ice tank (as ice tank contains brine almost at the same average temperature as the fast circulating brine), and the final or leaving temperature in case of cold storage rooms is 10° to 20°F. colder than that of the room. The compressors and complete equipment for cold storage rooms and ice plant will be selected in the next chapter and the temperatures of refrigerants (both ammonia and brine) and room prevailing will be discussed therein.

VI. COLD STORAGE PLANT AND EQUIPMENT

A. Unit Coolers or Cold Air Diffusers

Choice of Units

The application of room coolers requires an intimate knowledge of the principles underlying product conditioning, the characteristics of perishable product and relation of air conditions to the processing and storage of that product. The selection requires consideration of three major conditions to insure a successful well engineered installation.

1. Physical adaptability of the equipment in regard to installation and servicing comes first. This involves a consideration of the type of cold diffuser, floor mounted or suspended, the outlet velocity air distribution characteristics with respect to the size and shape of the room and sensitivity of the product to air motion. These considerations will allow the choice of the general type of unit before actual capacity figures are completed.

2. Thermal adaptability of the unit or units and refrigeration equipment to the loads and conditions to be maintained. This involves the selection of the actual unit size and refrigerant (brine) temperature to meet the specified sensible heat and latent heat loads and room conditions.

3. Total installation cost with accessories. The wide variety of sizes and types of cold diffusers makes it desirable to compare the installed cost of several small units of one type against a fewer number or a single large unit of another type. Either system may prove itself satisfactory and economical.

In modern cold storage rooms of sufficient size, floor mounted cold diffusers or unit space coolers--blower type are invariably employed. They are comparatively compact and self contained; and where more than one room is cooled, greater control is possible of temperature, etc. It can be brought and installed easily. The first cost is also less as the factory built units are of standard sizes produced in mass scale. They are easily accessible for inspection and repair.

These coolers consist of cooling surfaces or Brine coils, a motor driven fan or several fans on the same shaft, and directional outlets of different types. The air is directed horizontally at velocity of about 1000 rpm, near the ceiling and above the product and is well diffused throughout the space. The suction return for the air is placed at the floor level. For medium temperatures experienced in potato storage, the surfaces which are generally below 32°F. are defrosted by automatic shutdown in case the apparatus is required to operate for less than 24 hours. For such duty, brine spray is not necessary as it adds to the operating cost and frequent check on brine concentration of circulated brine in the tank, though brine spray cold diffuser would require less amount of cooling surface due to high rate of heat transfer per unit area.

Considering both the size and shape of potato storage rooms, too units per room will be desirable for flexibility of operation and even distribution of air.

Apparatus Dew Point

The rate of moisture deposited on cooling coils as frost depends on the difference of vapor pressures of room air and coils and should balance the rate of moisture given out by product and infiltered air--indicated by room sensible heat factor, at any time. For a given room temperature and a given sensible heat factor there is a corresponding supply air temperature and supply air moisture content which will satisfy the room cooling and dehumidifying requirements simultaneously with a fixed air quantity. This air temperature and moisture content are indicated by apparatus dew point (ADP), which is the temperature on cent percent saturation line of the air psychrometric chart intersected by the line drawn from point representing optimum room conditions and with a slope representing the S.H.F. Taking the potato storage rooms to be at 38°F., with 85 percent humidity and having sensible heat factors of

0.688 and 0.767 respectively during loading and storage, the corresponding apparatus dew points will be 26° and 31°F. This means that during loading period, the brine cooling coils should have surface temperature of 26°F., so that a given amount of air from the room when cooled down to this temperature will accomplish the required refrigeration job with required dehumidification to take care of the room latent heat. The cooling and dehumidification lines for both chilling and storage periods are shown in the accompanied pyschrometric chart (Fig. 16).

Only one air quantity will satisfy this dual requirement and any other air quantity will fail to absorb both heat and moisture at the required rate. Heat and moisture are originating within the air space or are being transmitted into the space (see method of computation of total sensible and latent heat loads) at a certain rate at the time for which estimate is made, and it is the function of conditioned air introduced into the space exactly to offset or balance these gains at the room conditions to be maintained.

By-Pass Factor

Now, for more uniform temperature control in the room, the difference of temperature of incoming and outgoing air should be as low as possible. This narrow temperature range or split during periods of maximum

PSYCHROMETRIC CHART

LOW TEMPERATURES

Barometric Pressure 29.92 In. Hg

ADDITIVE CORRECTIONS FOR W, h, AND v WHEN BAROMETRIC PRESSURE DIFFERS FROM STANDARD BAROMETER

Wet Bulb Temp. t'	Sat. Vapor Press. In. Hg	APPROXIMATE ALTITUDE IN FEET													
		−900		900		1800		2700		3700		4800		5900	
		$\Delta p = +1$		$\Delta p = -1$		$\Delta p = -2$		$\Delta p = -3$		$\Delta p = -4$		$\Delta p = -5$		$\Delta p = -6$	
		$\Delta W'_s$	Δh	$\Delta W'_s$	Δh	$\Delta W'_s$	Δh	$\Delta W'_s$	Δh	$\Delta W'_s$	Δh	$\Delta W'_s$	Δh	$\Delta W'_s$	Δh
−20	0.013	−0.06	−0.01	0.06	0.01	0.13	0.02	0.20	0.03	0.28	0.04	0.36	0.05	0.47	0.07
−18	0.014	−0.07	−0.01	0.07	0.01	0.14	0.02	0.23	0.03	0.32	0.05	0.41	0.06	0.52	0.08
−16	0.016	−0.07	−0.01	0.08	0.01	0.16	0.02	0.26	0.04	0.36	0.05	0.46	0.07	0.58	0.09
−14	0.018	−0.08	−0.01	0.09	0.01	0.18	0.03	0.29	0.04	0.40	0.06	0.52	0.08	0.65	0.10
−12	0.020	−0.09	−0.01	0.10	0.01	0.21	0.03	0.32	0.05	0.44	0.07	0.58	0.09	0.72	0.11
−10	0.022	−0.10	−0.02	0.11	0.02	0.23	0.03	0.36	0.05	0.50	0.07	0.64	0.10	0.81	0.12
−8	0.025	−0.12	−0.02	0.12	0.02	0.26	0.04	0.40	0.06	0.55	0.08	0.72	0.11	0.90	0.13
−6	0.027	−0.13	−0.02	0.14	0.02	0.29	0.04	0.44	0.07	0.62	0.09	0.80	0.12	1.00	0.15
−4	0.030	−0.14	−0.02	0.15	0.02	0.32	0.05	0.50	0.07	0.69	0.10	0.89	0.13	1.12	0.17
−2	0.034	−0.16	−0.02	0.17	0.02	0.35	0.05	0.55	0.08	0.76	0.11	0.99	0.15	1.24	0.19
0	0.038	−0.18	−0.03	0.19	0.03	0.39	0.06	0.61	0.09	0.85	0.13	1.10	0.17	1.38	0.21
2	0.042	−0.20	−0.03	0.21	0.03	0.44	0.07	0.68	0.10	0.94	0.14	1.22	0.19	1.53	0.23
4	0.046	−0.22	−0.03	0.23	0.03	0.48	0.07	0.75	0.11	1.05	0.16	1.36	0.21	1.70	0.26
6	0.051	−0.24	−0.04	0.26	0.04	0.54	0.08	0.83	0.13	1.16	0.18	1.51	0.23	1.89	0.29
8	0.057	−0.27	−0.04	0.29	0.04	0.59	0.09	0.93	0.14	1.28	0.19	1.67	0.25	2.09	0.32
10	0.063	−0.30	−0.04	0.32	0.05	0.66	0.10	1.03	0.16	1.42	0.22	1.85	0.28	2.31	0.35
12	0.069	−0.33	−0.05	0.35	0.05	0.73	0.11	1.13	0.17	1.57	0.24	2.04	0.31	2.56	0.39
14	0.077	−0.36	−0.05	0.39	0.06	0.81	0.12	1.25	0.19	1.74	0.26	2.26	0.34	2.82	0.43
16	0.085	−0.40	−0.06	0.43	0.06	0.89	0.14	1.38	0.21	1.92	0.29	2.49	0.38	3.12	0.48
18	0.093	−0.44	−0.07	0.47	0.07	0.98	0.15	1.53	0.23	2.12	0.32	2.75	0.42	3.44	0.53
20	0.103	−0.49	−0.08	0.52	0.08	1.08	0.17	1.68	0.26	2.33	0.36	3.03	0.46	3.79	0.58
22	0.113	−0.5	−0.08	0.6	0.09	1.2	0.18	1.9	0.29	2.6	0.40	3.4	0.52	4.2	0.64
24	0.124	−0.6	−0.09	0.6	0.10	1.3	0.20	2.1	0.32	2.8	0.43	3.7	0.57	4.6	0.71
26	0.137	−0.7	−0.10	0.7	0.11	1.4	0.22	2.3	0.35	3.1	0.48	4.1	0.63	5.1	0.78
28	0.150	−0.7	−0.11	0.8	0.12	1.6	0.24	2.5	0.38	3.4	0.52	4.5	0.69	5.6	0.86
30	0.165	−0.8	−0.12	0.8	0.13	1.7	0.27	2.7	0.42	3.8	0.58	4.9	0.75	6.1	0.92
32	0.180	−0.9	−0.13	0.9	0.14	1.9	0.29	3.0	0.45	4.1	0.63	5.3	0.82	6.6	1.01
34	0.197	−0.9	−0.14	1.0	0.15	2.1	0.32	3.2	0.49	4.4	0.68	5.7	0.88	7.2	1.11
36	0.212	−1.0	−0.15	1.1	0.17	2.2	0.35	3.5	0.53	4.8	0.74	6.2	0.96	7.8	1.20
38	0.229	−1.1	−0.17	1.2	0.18	2.4	0.37	3.8	0.58	5.2	0.80	6.8	1.05	8.4	1.30
40	0.248	−1.2	−0.18	1.3	0.20	2.6	0.41	4.1	0.63	5.7	0.88	7.4	1.14	9.2	1.42
42	0.268	−1.3	−0.20	1.4	0.21	2.8	0.44	4.4	0.69	6.1	0.94	8.0	1.23	10.0	1.54
44	0.289	−1.4	−0.22	1.5	0.23	3.1	0.47	4.8	0.74	6.7	1.04	8.7	1.34	10.8	1.67
46	0.312	−1.5	−0.23	1.6	0.25	3.3	0.51	5.2	0.80	7.2	1.11	9.4	1.45	11.7	1.81
48	0.336	−1.6	−0.25	1.8	0.27	3.6	0.56	5.6	0.87	7.8	1.21	10.2	1.58	12.6	1.95

t = Dry bulb temperature (F).
t' = Wet bulb temperature (F).
p = Barometric pressure (in. of Hg).
Δp = Pressure difference from standard barometer (in. of Hg).
W = Moisture content of air (gr per lb of dry air).
W'_s = Moisture content of air saturated at wet bulb temperature t' (gr per lb of dry air).
ΔW = Moisture content correction of air when barometric pressure differs from standard barometer (gr per lb of dry air).
$\Delta W'_s$ = Moisture content correction of air saturated at wet bulb temperature when barometric pressure differs from standard barometer (gr per lb of dry air).

NOTE: To obtain ΔW reduce value of $\Delta W'_s$ by 1% where $t - t' = 24$ F and correct proportionally when $t - t'$ is not 24 F.

h = Enthalpy of moist air (Btu per lb of dry air).
Δh = Enthalpy correction when barometer pressure differs from standard barometer, for saturated or unsaturated air (Btu per lb of dry air).
v = Volume of moist air (cu ft per lb of dry air).

$$v = \frac{.754\,(t + 459.7)}{p}\left[1 + \frac{W}{4360}\right]$$

PROPERTIES OF SATURATED AIR
−20 to −100 F

Temp. F	Vapor Press. In. Hg	Moisture Content Grains*	Volume cu ft*	Enthalpy Btu*
−20	.0126	1.83	11.08	−4.52
−22	.0112	1.63	11.03	−5.03
−24	.0100	1.45	10.99	−5.54
−26	.0089	1.29	10.93	−6.04
−28	.0079	1.15	10.88	−6.55
−30	.0070	1.02	10.83	−7.05
−32	.0062	0.90	10.78	−7.54
−34	.0055	0.80	10.73	−8.04
−36	.0049	0.71	10.68	−8.53
−38	.0043	0.62	10.63	−9.03
−40	.0038	0.55	10.58	−9.52
−42	.0033	0.49	10.53	−10.01
−44	.0029	0.43	10.48	−10.50
−46	.0026	0.38	10.43	−10.98
−48	.0023	0.33	10.38	−11.47
−50	.0020	0.29	10.33	−11.96
−52	.0017	0.25	10.28	−12.44
−54	.0015	0.22	10.23	−12.93
−56	.0013	0.19	10.18	−13.41
−58	.0012	0.17	10.13	−13.89
−60	.0010	0.15	10.08	−14.38
−62	.0009	0.13	10.03	−14.86
−64	.0008	0.11	9.98	−15.34
−66	.0007	0.10	9.93	−15.83
−68	.0006	0.08	9.87	−16.31
−70	.0005	0.07	9.82	−16.79
−72	.0004	0.06	9.77	−17.27
−74	.0004	0.05	9.72	−17.75
−76	.0003	0.05	9.67	−18.23
−78	.0003	0.04	9.62	−18.71
−80	.0002	0.03	9.57	−19.19
−82	.0002	0.03	9.52	−19.68
−84	.0002	0.03	9.47	−20.16
−86	.0001	0.02	9.42	−20.64
−88	.0001	0.02	9.37	−21.12
−90	.0001	0.02	9.32	−21.60
−92	.0001	0.01	9.27	−22.80
−94	.0001	0.01	9.22	−22.56
−96	.0001	0.01	9.17	−23.04
−98	.0001	0.01	9.12	−23.52
−100	.0000	0.01	9.07	−24.00

*Per lb of dry air

Example: At a barometric pressure of 25.92 with 30 F DB and 28 F WB, determine W, h, and v. $\Delta p = -4$ and from table $\Delta W'_s = 3.4$. From note above,

$$\Delta W = \Delta W'_s - \left(\frac{2}{24} \times .01 \times 3.4\right) = 3.4 - .003 = 3.4.$$

Therefore $W = 19.2$ (from chart) $+ 3.4 = 22.6$ gr per lb of dry air. From table $\Delta h = 0.52$. Therefore h = saturation enthalpy from chart + deviation + 0.52 = 10.1 + .06 + 0.52 = 10.68 Btu per lb of dry air. From equation above

$$v = \frac{.754\,(30 + 459.7)}{25.92}\left[1 + \frac{22.6}{4360}\right] = 14.32 \text{ cu ft per lb of dry air.}$$

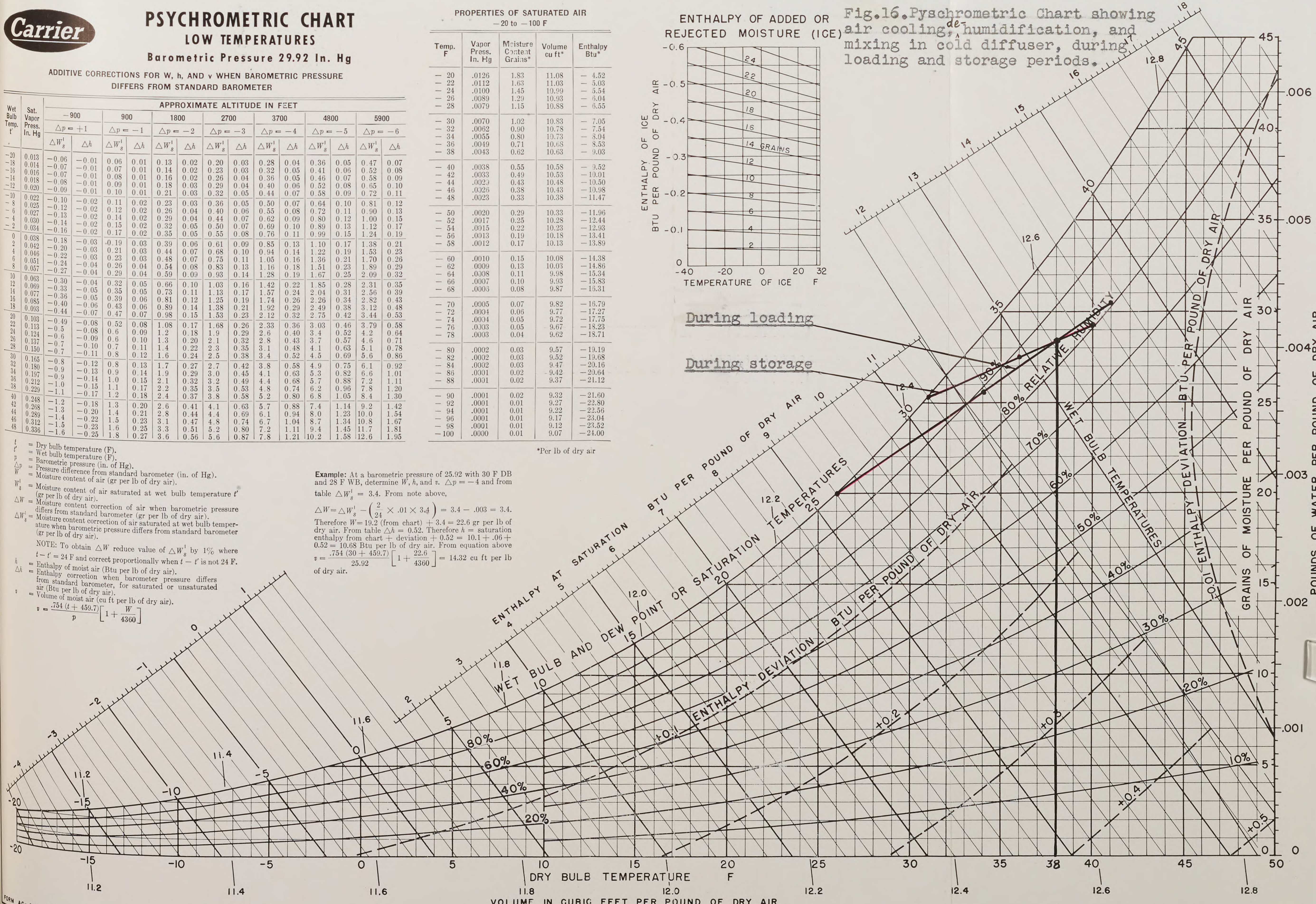


Fig.16. Pyschrometric Chart showing air cooling, dehumidification, and mixing in cold diffuser, during loading and storage periods.

load, i.e. during loading may be 6 to 10 degrees and not exceeding 4 degrees during storage.

If for proper cooling and dehumidification of room air apparatus dewpoint during loading is 26°F. and if all the air through the cooling unit or units is cooled to this temperature, the temperature range of incoming and outgoing air will be 26° to about 40°F. i.e. 14° split which is too great. The only course to avoid this split and bring it to about 8°F. is to by-pass a portion of hot air from room in the cold diffuser and mix it with the remaining air cooled and dehumidified to 26°F. saturated in such proportion that resultant air mixture gives the desired temperature split. For air mixture temperature to be about 34°F. and outgoing air temperature of say 41°, the split may be only 7°F. and ratio of by pass air to cooled air as

$$\frac{34 - 26}{41 - 34} = \frac{8}{7} \qquad \text{(See Fig. 16)}$$

and, by-pass factor (B.P.F.) which is the ratio of by-pass air to total air circulated through the cooling unit, will be

$$\frac{8}{8 \quad 7} = \frac{8}{15} \doteq 0.55 \text{ approximately.}$$

It will be noted that average air temperature in the room will be $\frac{1}{2}(41 \quad 34) = 37.5^{\circ}$F. or very near to the room temperature desired.

Running Time

There are several considerations for determining the number of hours per day that the coils may remain in service circulating brine refrigerant. This may depend on plant layout, adaptability of automatic control system, size of the plant, defrosting, etc. Since automatic control is more accurate and convenient in a small plant like this, a very well trained engineer is not needed for frequent hand adjustments. Further, intermittent running of the brine will automatically defrost the coils when brine in pipes rises to room temperature. For the capacity of the plant in question, a 20 hours running time seems to be the most practical approach. Hence, the apparatus for cooling air and all the condensing and auxiliary equipment will be designed for higher capacity in inverse proportion to the running time. Since total rooms load is 28.2 tons druing storage, the capacity of each cooling unit or cold diffuser would be

$$\frac{28.2 \times 1.2}{6} = 5.64 \text{ Tons} = 67760 \text{ Btu/hr};$$

as there are 6 units in all the 3 potato storage rooms.

Cfm Requirements of Air

Amount of air cooled in direct contact with brine coils in lbs. can be found by dividing the total refrigerating job in heat units accomplished by each unit by the difference in enthalpy of the cooled air. Since amount of cooling and dehumidications are both balanced with the room conditions, this amount can be found by merely taking the sensible heat per unit and dividing it by degree of cooling accomplished in cooling coils. To find out the total amount of air circulated through the unit (which will include the by-pass air), by-pass factor will be taken into account. Expressed in cfm, the air quantity per unit will be

$$\text{Cfm} = \frac{\text{Sensible heat per unit in Btu/hr.}}{1.2 \times (1 - \text{BPF}) \times (t_a - t_{ADP})}$$

(Where t_a is the temp. of air entering and t_{ADP} is apparatus dew point)

$$= \frac{67750 \times 0.688}{1.2 \times (1 - 0.55) \times (41 - 26)} = 5770, \text{ say } 6000$$

This would necessitate two fans each 3000 cfm capacity operated by a common shaft direct coupled motor of about 1-1/2 H.P. running at 1450 rpm.

Quality and Amount of Brine Circulated

The average temperature of brine circulated will depend upon temperature difference of brine and the cooled air for required amount of heat transfer Q, the surface area A of the brine coils, the overall coefficient of heat transfer "U" of the coils and factor for running time N, as given by formula:

$$Q = AU \times \log \text{mtd} \times N$$

In designing the unit cooler, temperature difference and approximate value of "U" are assumed and area of surface found.

Brine is also needed during storage season for circulation through ice tank. We may safely assume the lowest brine temperature at any time during storage and loading periods to be 15°F. Hence, brine concentration should be such that its freezing point is at least 10°F. below this lowest brine temperature to minimize the risk of freezing of brine in the coils.

Taking 20 per cent concentration by weight of sodium chloride (salt), 75.2 Salometer degrees at 59°F., the freezing point will be 1.8°F. with 9.64 lbs. per gallon of solution and specific heat of 0.813 Btu per lb. per deg. F.

Temperature rise of brine in passing through cold rooms is usually 4° to 8°F. Assuming allowable temperature rise of 5°F., the weight of brine circulated in gpm W_b can be found from the formula:

$$Q = W_b \times 9.64 \times 60 \times (\text{Sp.heat}) \times (\text{temp.diff.}) \times N.$$

Since temp. difference is assumed to be 5°F. and $N = \frac{1}{1.2}$,

$$W_b = \frac{56500 \times 1.2}{60 \times 9.64 \times 0.813 \times 5} = 28.75 \text{ gpm.}$$

The exact range of brine temperature will be computed later with reference to the selection of brine coolers, and depends on both unit cooler and brine cooler design. These brine temperatures would also give the log mean temp. difference (dt) of the air in contact with coils and brine, which will give the surface area of the coils for design purposes.

Performance during Storage Season

During storage period, the refrigeration load on the cold diffusers is a little more than half the maximum load, i.e. total of 15.34 tons or 30700 Btu/hr. per each unit. The apparatus dew point is 31^{o}F. If the cfm of air discharged and B.P.F. per unit is the same (i.e. fan is running at constant speed) the temperature of room air t_{a_1} going in the cooler can be found:

$$t_{a_1} - t_{ADP} = \frac{\text{Sensible heat load of unit}}{1.2 \times (1 - BPF) \times cfm}$$

$$= \frac{141000 \times 1.2 \text{ (Factor for running time)}}{1.2 \times (1.055) \times 5770 \times 6} = 9^{o}F.$$

Temperature of leaving room air = 31 + 9 = 40^{o}F.

with same B.P.F. = 0.55 Temp.of entering room air t_{a2} is given by:

$$\frac{t_{a2} - t_{ADP}}{t_{a1} - t_{ADP}} = \frac{T_{a2} - 31}{Ho - 31} = 0.55$$

or $t_{a_2} = 36^{o}F.$

Hence, average room air temperature = $\frac{1}{2}(t_{a_1} + t_{a_2})$

$= \frac{1}{2}(36 + 40) = 38\ ^{o}F.$, as it should be.

Allowing for the same brine temperature rise as before i.e., 5^{o}F., the brine circulation pump capacity would be reduced. Hence, a suitable two-speed gear should be applied to the pump to give the reduced gpm capacity which will be

$$= \frac{30700 \times 1.2}{60 \times 9.64 \times 0.813 \times 5} = 15.5 \text{ gpm.}$$

Air Motion in Product Zone

The importance of uniform air distribution to every part of the product zone and permissible velocity of air over the product must be recognized when selecting floor mounted units and their outlets. The maximum allowable air velocity over the product surface to prevent excessive drying for potatoes, as recommended by various authorities on potato storage, is 150 feet per minute. It is important that the calculated average velocities in the product zone do not exceed this specified maximum. If the calculated or assumed air velocity falls below the maximum allowable velocity it is of no consequence so long as the air distribution throughout the storage space is uniform and every portion of the room is being reached by diffuser air. Such air distribution is of vital importance in all cooler applications. Beamed ceilings, columns and methods of storing product will often complicate the distribution problem. These and other variables make it impossible to lay down exact rules as to allowable variation in air velocities. Considering all of these factors, a theoretical value of 100 fpm may be taken for the product zone velocity, though actual results must depend on judgment being used in the layout of outlets, adjustment of directional loures or dampers, and obstructions in the way. This velocity can be safely maintained both during loading and storage period, without much drying effect on the product.

The total quantity of air movement in the space can be found by multiplying this average product zone velocity with the net effective room cross section. The net effective cross sectional area of the room is the area of room taken at location of outlets of the cold diffuser and served by that unit only and perpendicular to the supply air stream multiplied by service factor depending on the stacking arrangement and service conditions of the storage. For long carrying warehouse with rooms having sufficient aisle ways for truck loading, this service factor may be assumed to be 0.5. Hence, net effective cross sectional area for each unit

$$= 0.5 \times \tfrac{1}{2} \times 70 \times 15 = 263 \text{ sq. ft.}$$

Hence, total air motion in part of the room served by each cold diffuser = product zone velocity x net effective area = 100 x 263 = 26300 cfm.

The quantity of air in room is not that passing through the cooling coils only, for additionally, there is always set in motion a quantity of air many times in excess of air handled by cooling unit. Since the cfm discharge of each unit is about 6000, the ratio of total air set in motion to the cfm discharge, which is called velocity factor, is $\frac{26300}{6000} = 4.4$.

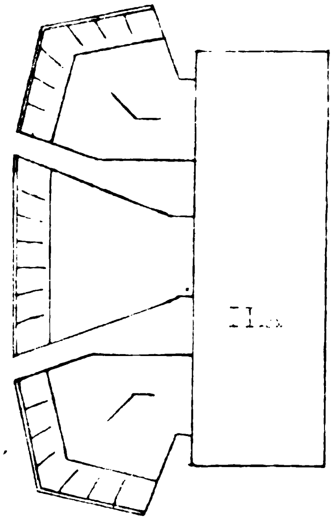
This velocity factor depends on the outlet velocity of the unit cooler, as higher outlet velocity will set up large air currents. A fairly good relation between outlet velocity and the velocity factor has been found by experiments (8) and is given by the following table:

Outlet velocity f.p.m.	vel.factor	Outlet velocity f.p.m.	vel.factor
700	3.4	1300	6.8
800	3.7	1400	7.6
900	4.0	1500	8.4
1000	4.5	1600	9.2
1100	5.2	1700	10.2
1200	6.0	1800	11.2

From this table, it can be found that when velocity factor is 4.4, the outlet air velocity from cold diffuser will be 1000 fpm. Hence, standards low velocity outlets, such as shown in Fig. 17 can be designed to give this air velocity.

Length of Throw

From the view point of air distribution, wall location of diffusers, as decided upon by us, is only permissible where the required "throw" of the air, which is the distance across the units to which the cooled air can be satisfactorily distributed, is within the maximum obtainable with the given conditions. The length of throw required depends upon the type and size of the room construction and in case of present room layout will be a



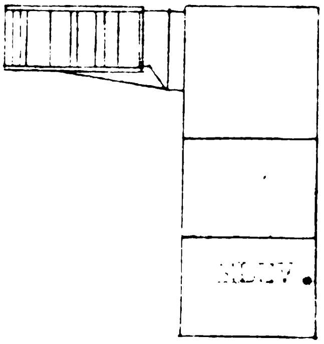


Fig. 17. Standard Low Velocity
Outlets for Blower type Unit Coolers.

little over 50 feet from the unit outlets. Maximum horizontal clear throw obtainable for standard low velocity outlets with straight vane setting and various outlet air velocities for rooms with flat ceilings and no obstructions are given in the following table (10):

Outlet vel. in fpm.	Approximate throw in ft.
500	55
1000	110
1500	160

Hence, since outlet velocity of cold diffusers has been found out to be 1000 fpm, the obtainable air throw of 110 feet is more than sufficient for the prevailing room conditions of the potato cold storage. If the discharge air strikes obstacles, such as beams, columns, etc., poor distribution will follow. The unit should accordingly be located, and the air distributed, so as to avoid beams and columns wherever possible. Since air discharged should not directly hit the product but be distributed from the open space at the top, the air discharge outlets should be at sufficient height above the floor.

B. Brine Coolers and Race System for Brine Tank

General Outline

While discussing incoming and outgoing air temperatures and temperature rise of brine through the cold diffusers, it was noted that during loading time the total refrigeration load was 28.2 tons and apparatus dewpoint 26°F. Average brine temperature in the coils may be about 10°F. below the apparatus dew point, i.e. about 15°F. and refrigerant temperature of say 5°F. for a given area of heat transfer.

During storage period the load is decreased to 15.34 tons and apparatus dew point raised to 31°F. due to proportionately less latent heat load. If the ammonia refrigerant temperature in a brine cooler, designed to do the maximum job during loading, is kept the same, viz. 5°F. (which is the same as that for ice tank for maximum compressoring efficiency) this brine cooler will be too large to do the job of nearly half the refrigeration load during storage, particularly when temperature difference of brine and air and brine and ammonia is going to be higher under these conditions due to higher apparatus dew point. This brine cooler for maximum job cannot be used during storage period both for cooling rooms and for the ice tank, as the brine temperatures needed for these two jobs are very different, as will be found later.

To overcome this difficulty, the best way is to employ either (a) two brine coolers of suitable sizes-- one insulated and solely for the rooms and the other submerged single-pass type in a race way of ice tank, or (b) one insulated brine cooler as before and one vertiflow unit evaporator in a race way of ice tank. The capacities and the surface areas should be such that during loading period, both brine coolers or brine cooler and vertiflow unit should supply brine at same temperature to the rooms for maximum load conditions; but when storage season begins and ice plant begins to operate, the brine coolers supply brine at suitable temperatures to cool rooms and ice tank entirely independently. The submerged brine cooler or vertiflow unit coils may be located in an adjacent trunk or tunnel, so that brine velocity in the trunk may be boosted up around the coils or in the brine cooler to achieve high rate of transfer. Further, it will be convenient to circulate the brine from this end of the ice tank to the room coils during loading season without disturbing the mass of brine in the whole ice tank. During storage season, this brine supply will be entirely shut off to the rooms and brine will be made to circulate by means of agitator within the tank at lower temperature for the manufacture of ice alone.

During storage conditions, though brine temperatures in the brine cooler and ice tank are different, it is convenient to always maintain the same brine concentration, i.e. giving freezing point of $1.8^{o}F.$, so that immediate switch over from loading conditions to storage conditions is possible.

Capacities and Efficiencies

Before we find out the various brine temperatures and ammonia evaporating temperature to suit the above conditions, let us find the refrigerating tonnage of the two brine coolers or brine cooler and vertiflow race way coil for these conditions.

Brine cooler for room coils may be horizontally placed on the skids with insulation all round or buried in the ground to save insulation. But some insulation losses are bound to accur. A 95 percent thermal efficiency may be taken to account for these losses, including those in brine lines. This loss also occurs in the ice tank which will be taken care of later while determining its capacity in tons of ice. As running time of room coolers is 20 hours, the running time of brine cooler and brine tank coils will also be the same during the loading period. This is because brine will circulate through them only when it circulates through rooms. Hence, total evaporating capacity of both brine coolers or a brine cooler and vertiflow unit would be:

$$= \frac{28.2 \times 24}{0.95 \times 20} = 35.6 \text{ tons.}$$

Evaporating capacity of the insulated brine cooler for rooms during storage only would be:

$$= \frac{15.34 \times 1.2}{0.95} = 19.37 \text{ tons}$$

Hence, evaporating capacity of submerged brine cooler or vertiflow unit for ice making would be

= 35.6 - 19.37 = 16.23 tons.

Coefficients of Heat Transfer

Shell and Tube brine coolers have high overall coefficients of heat transfer or "U" value. The Air Conditioning and Refrigerating Machinery Association have adopted "U" values for horizontal multipass shell and tube (flooded) coolers using ammonia. The values differ with brine velocities through coils, log mtd and average brine temperature. For temperature difference of 10° to 15°F. and brine temperature varying between 15° to 30°F. the "U" values vary between 80 and 100 in a region giving brine velocity range of 200 to 300 fpm. An average value of 80 Btu per hr. per sq. ft. of external surface per degree F. may be taken for this purpose during loading and 100 during storage.

For submerged coils of modern design or submerged brine cooler, both with flooded system, the same average value of 80 Btu per hr. per sq. ft. per deg. F. may be taken for average brine velocities of 100 fpm in the ice tank during loading and 100 for higher velocity during ice manufacture.

In ammonia flooded systems, the static pressure of liquid ammonia in the cylinder or submerged coils will cause the upper layer of ammonia to boil at lower pressure and hence, lower temperature than for lower portion. Brine inlet and outlet temperatures are also different. For accuracy log mtd should be used in the formula $Q_e = UA\ (dt)$, but since variation is not very great an arithmetic mean temperature difference may be used for design purposes.

Refrigerant Temperature

Ammonia evaporating temperature should be a few degrees above the freezing point of brine so that sudden fall in the room load or in ice tank may not cause brine congealing in the pipes or in shell coolers. To avoid installing a back pressure valve at the cost of thermodynamic efficiency on the ammonia line leading to insulated brine cooler operating at higher brine temperature, let ammonia evaporating temperature be the same throughout. Temperature of brine for ice manufacture is generally in the neighborhood of 15°F. Ammonia should be 5 to 8° below this temperature. Hence, we choose ammonia temperature of 7.5°F. for design purposes.

Brine Temperatures

Temperature of brine t_b in the ice tank for the manufacture of ice depends on freezing time of ice required in hours T and thickness of the ice block w in the following (29) relationship, based on data taken from modern ice tanks:

$$32 - t_b = \frac{6.2 \times w^2}{T}$$

Temperature of brine varies between 12° to 18°F. depending on different conditions. If it is desired to freeze ice within 48 hours and ice cans have maximum standard width of 11-1/2" at the top, the brine temperature of 14°F. will be the most desirable as will be clear from the freezing time, which will be

$$= \frac{6.2 \times (11.5)^2}{32 - 14} = 45.6 \text{ hrs.}$$

Now, we have only to find the average brine temperature during loading period t_1 common to both brine cooler and ice tank and during storage season t_s for the brine cooler only supplying brine to rooms. Surface area of brine cooler or submerged coils in the ice tank is given by expression $A = \frac{Qe}{U \times dt}$, where Qe is the refrigerating effect or capacity of evaporators' coils and dt the arithmetic mean temperature difference between ammonia and brine. "U" has been assumed to be 80 Btu per hr. per sq. ft. per deg. F. during loading and 100 during storage for both evaporators. Equating area required during loading period with total area of both brine cooler and submerged coils or cooler during storage period, we get

$$\frac{35.6}{80 \times (t_1 - 7.5)} = \frac{19.37}{100\ (t_s - 7.5)} + \frac{16.23}{100\ (14 - 7.5)}$$

or, $$\frac{44.6}{t_1 - 7.5} = \frac{19.37}{t_s - 7.5} + 2.5 \qquad \text{(i)}$$

Again, for same room coolers operating with same cfm of air discharge, the "U" values for air cooling will be the same. Hence, log mtd of brine in the room coils and room air will be proportional to the room loads for loading and storage periods. (See Fig. 18 for finding out the log mtd under loading and storage conditions).

i.e., $$\frac{10}{\log \frac{38.5 - t_1}{28.5 - t_1}} : \frac{4}{\log \frac{37.5 - t_1}{33.5 - t_s}} = 28.2 : 15.34$$

or, $$\log \frac{37.5 - t_s}{33.5 - t_s} = 0.736 \times \log \frac{38.5 - t_1}{28.5 - t_1} \text{ . } \_\_\_\_\_\_(ii)$$

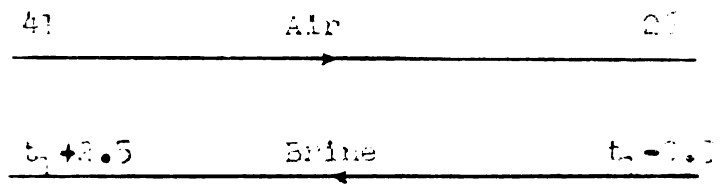
To solve the simultaneous equations (i) and (ii) for values of t_1 and t_s is not so easy. The best way is to find the values when arithmetic rather than logarithmic mtd is used for equation (ii), which will become

$$\frac{33.5 - t_1}{28.2} = \frac{35.5 - t_s}{15.34} \quad \_\_\_\_\_\_(ii),$$

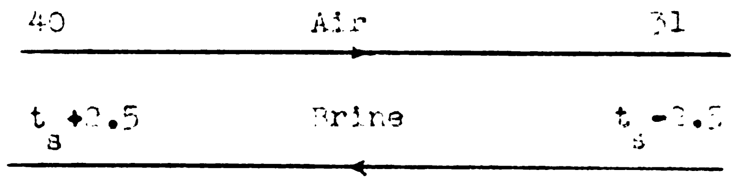
and then applying trial and error method to solve the original equations, this will give

$t_1 = 20.5^{\circ}$F. and $t_s = 28.5^{\circ}$F.

Hence, during loading, temperature of brine entering 23°F., and temperature of brine leaving 18°F.; and, during storage, temperature of brine entering 31°F., and temperature of brine leaving 26°F.



(a) During loading



(b) During storage

Fig. 18. Counterflow lines showing air and brine temperatures in cold air diffuser, during loading and storage.

Surface Areas

Having found the average brine temperatures, it is easy to find the effective surface area of the brine coolers or vertiflow unit from the formula:

$$A = \frac{Qe}{U \times (dt)}$$

Surface area of insulated brine cooler =

$$\frac{19.37 \times 12000}{100 \times (28.5 - 7.5)} = 111 \text{ sq. ft. or say, } 115 \text{ sq.ft.}$$

and, surface area of submerged cooler or vertiflow unit =

$$\frac{16.23 \times 12000}{100 \times (14 - 7.5)} = 300 \text{ sq. ft.}$$

Total surface area = 115 300 = 415 sq.ft.

To verify this, find the surface area needed during loading period, which will be

$$= \frac{35.6 \times 12000}{80 \times (20.5 - 7.5)} = 412 \text{ or say } 415 \text{ sq.ft.}$$

same as above

ASRE Standard Ratings

ASRE ratings are given under the conditions specified in at least one of the following groups for brine coolers.

Group No.	Temp.of entering brine, ^{o}F.	Temp.of leaving brine, ^{o}F.	Sat.Temp. of leaving refr.nt.^{o}F.	Ambaient air Temp. ^{o}F.
V	27	12	5	90
VI	24	14	5	90
VII	21	16	5	90
VIII	14	11	5	90

Multipass brine cooler has capacity of

$$19.37 \times \frac{19 - 5}{28.5 - 7.5} = 13 \text{ tons under group VI}$$

Submerged brine cooler or race coil has

$$16.23 \times \frac{12.5 - 5}{14 - 7.5} = 18.5 \text{ tons under group VIII}$$

$$\text{or,} \quad 16.23 \times \frac{19 - 5}{14 - 7.5} = 35 \text{ tons under group VI}$$

Hence, from surface areas or standard tons capacity, it is evident that submerged brine cooler or race vertiflow unit has about two and one half times the capacity than that of multipass insulated cooler.

Horizontal multipass and single-pass shell and tube brine coolers are shown in Figs. 19 and 20.

Brine Flow Rates

During loading period both multipass brine cooler and submerged coils or cooler supply the circulated brine to all the 6 room cold diffusers at the rate of 6 x 28.75 or 172 gpm. Since temperature drop of brine in both the brine coolers is the same, the rate of brine flow in the two units will be in proportion to their standard tonnage ratings or the surface areas. Hence, rate of brine flow in submerged cooler or coils will be about 122 gpm and that in the multipass brine cooler will be 50 gpm during loading period.

During storage, multipass brine cooler alone will supply the brine to the rooms and is 6 times rate per

Fig. 19. Horizontal Multi-pass Shell and Tube Brine Cooler.

Fig. 20. Submerged Single-pass Shell and Tube Brine Cooler.

each cold diffuser, i.e. 6 x 15.5 or 93 gpm. In designing the brine coolers, i.e. finding number and size of tubes and number of by-passes, it should be borne in mind that brine velocity should vary between 200 fpm to 375 fpm during loading and storage respectively, as "U" values do not differ greatly in this range, and are 80 and 100 Btu respectively.

Similarly for submerged brine cooler single-pass type or vertiflow unit, the brine flow will be speeded up during ice manufacture so that fast circulation keeps very low temperature gradient in the ice tank- not more than $\frac{1}{2}$°F. Brine agitator or circulator may be speeded up (by a variable speed drive) to give the desired temperature gradient. Velocity of brine in the trunk or tunnel may be high, up to 200 fpm, but that round the ice cans should be between 18 to 30 fpm.

Design of Cold Diffusers

Having found out the probable entering and leaving brine temperatures for cold diffusers, it is easy to find out the surface area of the coils from the formula

$$Q = U F A (dt) \times N$$

Where, U = Coefficient of heat transfer for air cooling, in forced air circulation through the brine coils.

Motz (29) gives the following values of "U" for brine coils in moving air:

Vel.of air,fpm.	200	300	400	500	600	700	800
U (external surface), Btu/hr/sq.ft/deg.F.	2.9	4.0	5.0	5.9	6.8	7.7	8.6

Usually air velocity through coil bunkers of blower type unit coolers runs very high, say 1000 to 1200 fpm. So, plotting the graph (Fig. 21) for the above table and extending it to give value for 1100 fpm, we get U = 11.3.

F = Frost Factor = 1, for automatic defrosting on off cycle.

A = External area of brine coils in sq. ft.

dt = log mtd,

$$= \frac{10}{\log_e \frac{18}{8}} = 12.3^{\circ}F.$$

N = Running time factor $= \frac{20}{24} = 0.833$,

Q = Room load in Btu/hr. for each unit, during loading.

Hence, 56500 = 11.3 x A x 12.3 x 0.833

A = 486 sq. ft.

Hence, about 1750 feet of 3/4" nominal size prime surface pipes will be required.

If the bunker has 10 rows, each bank will have 175 ft. length of coils or 48 coils each 3.75 ft. effective length. The coils should be so staggered in rows and banks that net effective area for air flow gives necessary high air velocity through the coils, i.e. 1100 fpm with given cfm discharge. A modern finned surface cold diffuser is shown in Fig. 22.

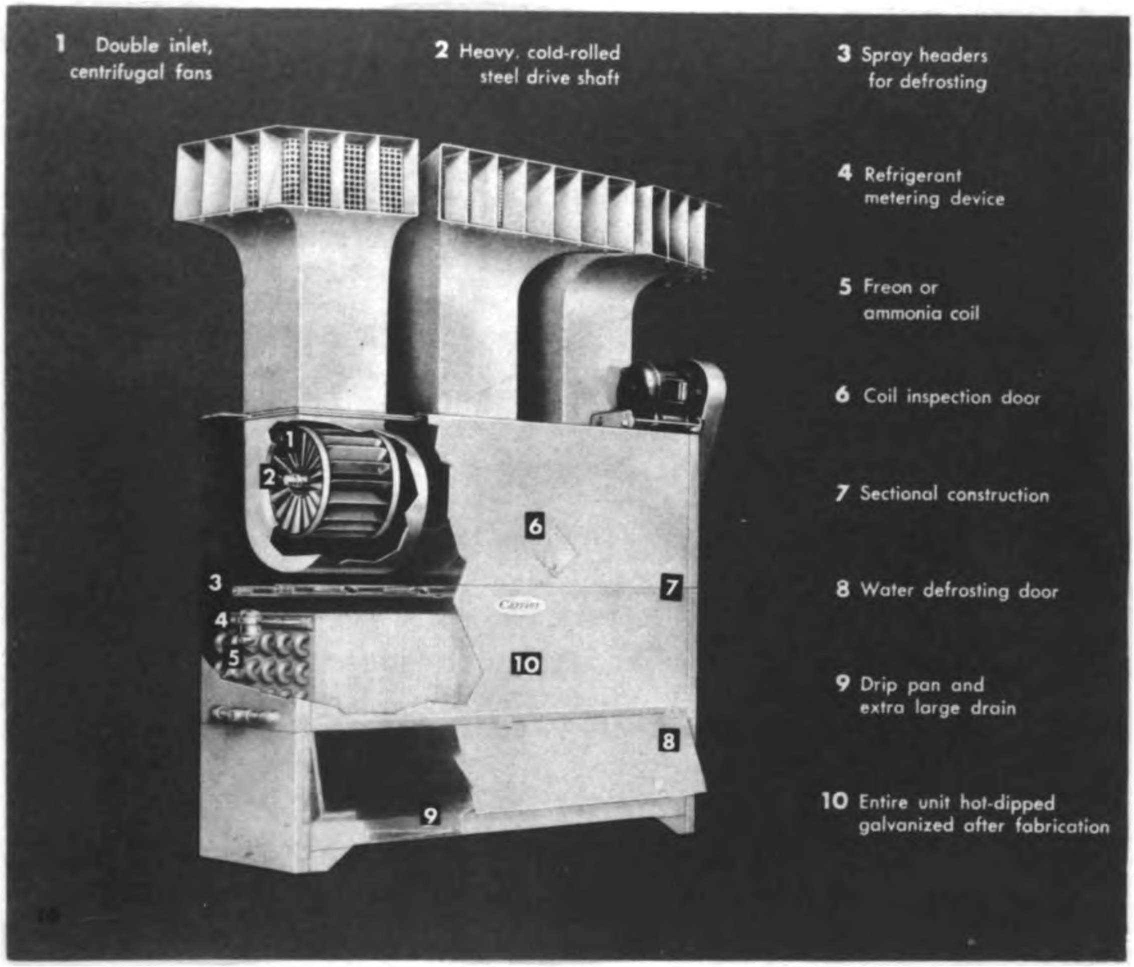


Fig. 22. Cold Air Diffuser with Finned Coils.
(Spray Headers shown at 3 will not be needed.)

C. Brine Circulation and Control

Brine Pressure Drop

Loss of head for a length of pipe of given size for certain gpm rate of flow is available in many data books for water, brine and certain other liquids. Having designed the brine cooler and the cold diffusers, we can find the total loss of head in feet orpressure drop in psi for the brine circulation system, taking into account the various lengths and sizes of pipes, tees, bends and fittings. The circulating pump should be designed to give the required discharge by working against this total head. As gpm and total head are different for loading and storage periods, the pump should be designed for maximum duty.

Brine Mixing

Brine concentration should be kept to required degree, Sodium chloride (salt) dissolves readily in agitated or circulated water or brine. It may be dissolved in a small separate tank about 4' x 4' x 4-1/2' preparatory to introducing it into a brine system. Natural circulation may be set up by placing the flake or solid material in a wire basket or other perforated container suspended just below the water or solution level. As the sodium chloride dissolves, the heavier solution settles toward the bottom of the tank and is replaced by lighter solution, this action continuing until the dissolving is completed.

Thus, no mechanical agitation is necessary. The solution may then be pumped in the circulation system.

Brine Pipes

Standard weight galvanized steel pipes screwed fittings are generally used in brine circulation system. Screwed joints are put together with graphite and oil pipe compound. Sleeves of moulded cork covering are provided for all pipes passing through refrigerator walls. They should be large enough to permit covering on covered pipes to pass through freely and should have ends flush with the sides of the wall. All horizontal piping should be properly supported by adjustable hangers spaced about 8 ft. apart adjusted to the drop and fastening the line outside the covering.

To permit smooth flow of brine without undue loss of head, pipes should be of ample size. A skeleton single line diagram of brine pipes leading to various room coolers giving suitable sizes is shown in Fig. 23.

The circulating system should operate in such a manner that during loading period, both brine cooler and brine tank for ice should supply circulating brine through the pump. When storage period starts, supply of brine from ice tank should be stopped and brine should be made to circulate in lesser rate of flow through the brine cooler only. This can be arranged by brine pipe lines

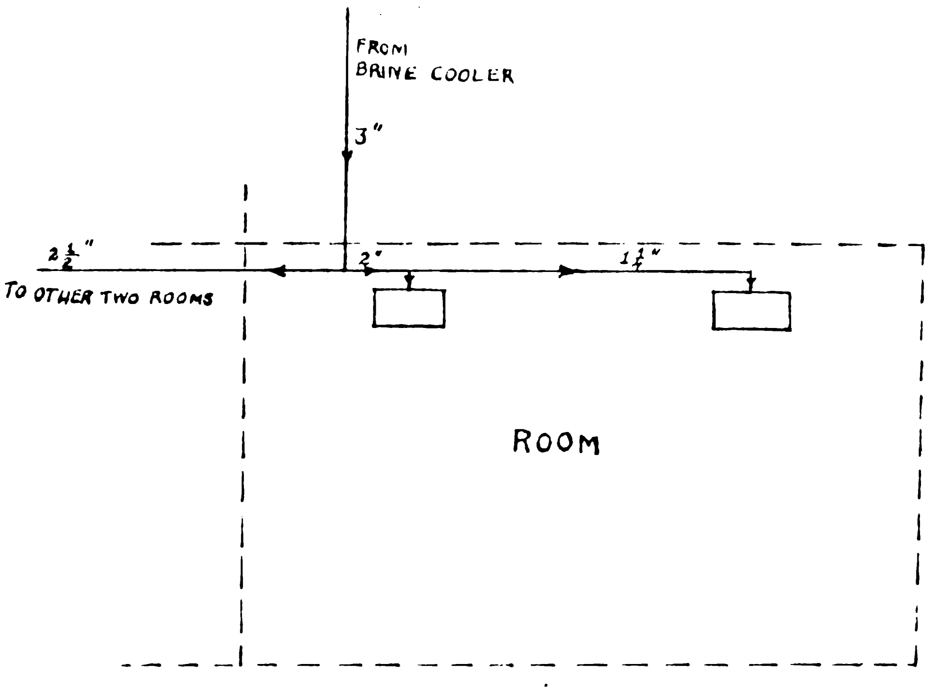


Fig. 23. Skeleton Diagram of Brine Pipes to Room Coolers, showing Pipe Sizes.

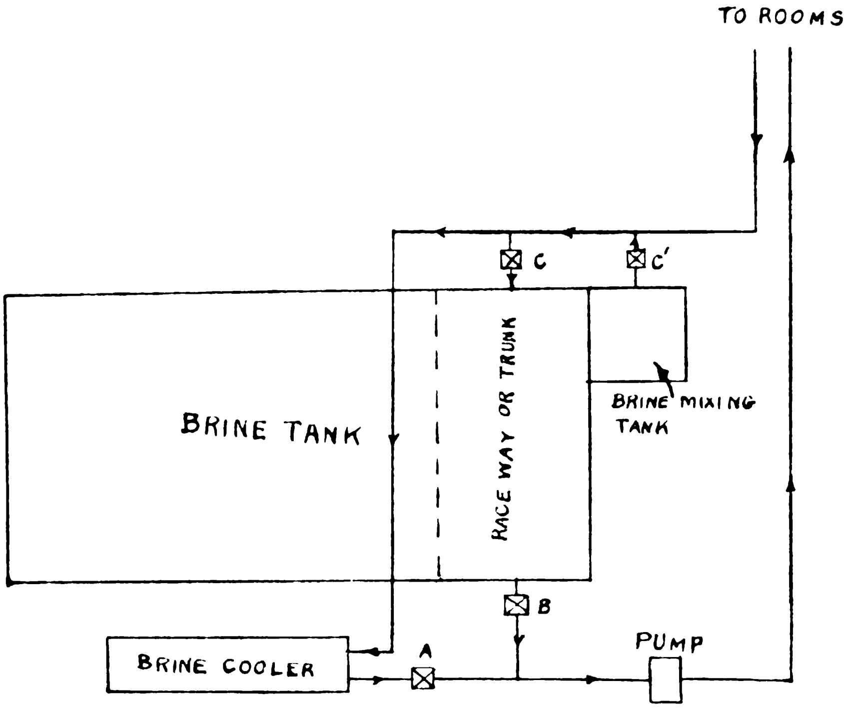


Fig. 24. Brine Circulation System with Sluice Valves.

shown in diagram (Fig. 24) with regulating (Sluice) valves A and B and a check valve C. Any adjustment in the amount of brine circulated through the two heat exchangers during loading may be made by means of sluice valves so that proper brine temperature in the rooms may be maintained, with resulting room temperatures at desirable level.

Brine Pump

The brine pump should be of rotary gear type with positive displacement and constant capacity, so that under varying heads (depending on number of cold diffusers in operation at a time) a constant supply of cold brine is circulated. A relief valve can be located at a suitable point in brine supply mains so as to bypass the flow of refrigerant, in excess of that which can be taken through the circuit that remains open, back into the brine cooler.

The pump will be connected to the motor through a reduction gear giving two speeds of the pumps, so that during loading season the capacity of the pump is $6 \times 28.2 = 170$ gpm and during storage it reduces to $6 \times 15.3 = 92$ gpm.

Generally two identical pumping sets are installed side by side--only one in operation at a time and the other as a standby, so that there is no break in the supply of brine to the cold stores.

Temperature Control

The success of any cold diffuser installation can be affected by the type of control system used. In the brine circulation system, automatic control is quite simple and affords better control of room conditions due to flywheel effect of brine.

It is necessary to place a magnetic stop valve in the feed line to each diffuser ahead of the hand operated control valve. Magnetic valve will be operated by a room thermostat which opens and closes the valve in accordance with the refrigeration demands of the enclosure. Strap-on thermostat on the return line near the brine coils will offer additional control of brine temperature leaving the rooms, such that

(a) on fall in room temperature, roomstat alone would close the solenoid valve and stop the flow of the liquid; but

(b) on rise of room temperature, room stat closes the circuit RW in relay (See Fig. 25), but the solenoid remains closed until strap-on-stat closes circuit WB in relay which and only then can open the solenoid valve to refrigerant to the coil. Strap-on-stat setting should be made high enough to indicate defrosted coil.

With the plant having full automatic operation, it would be advisable to shut down the brine circulating pump

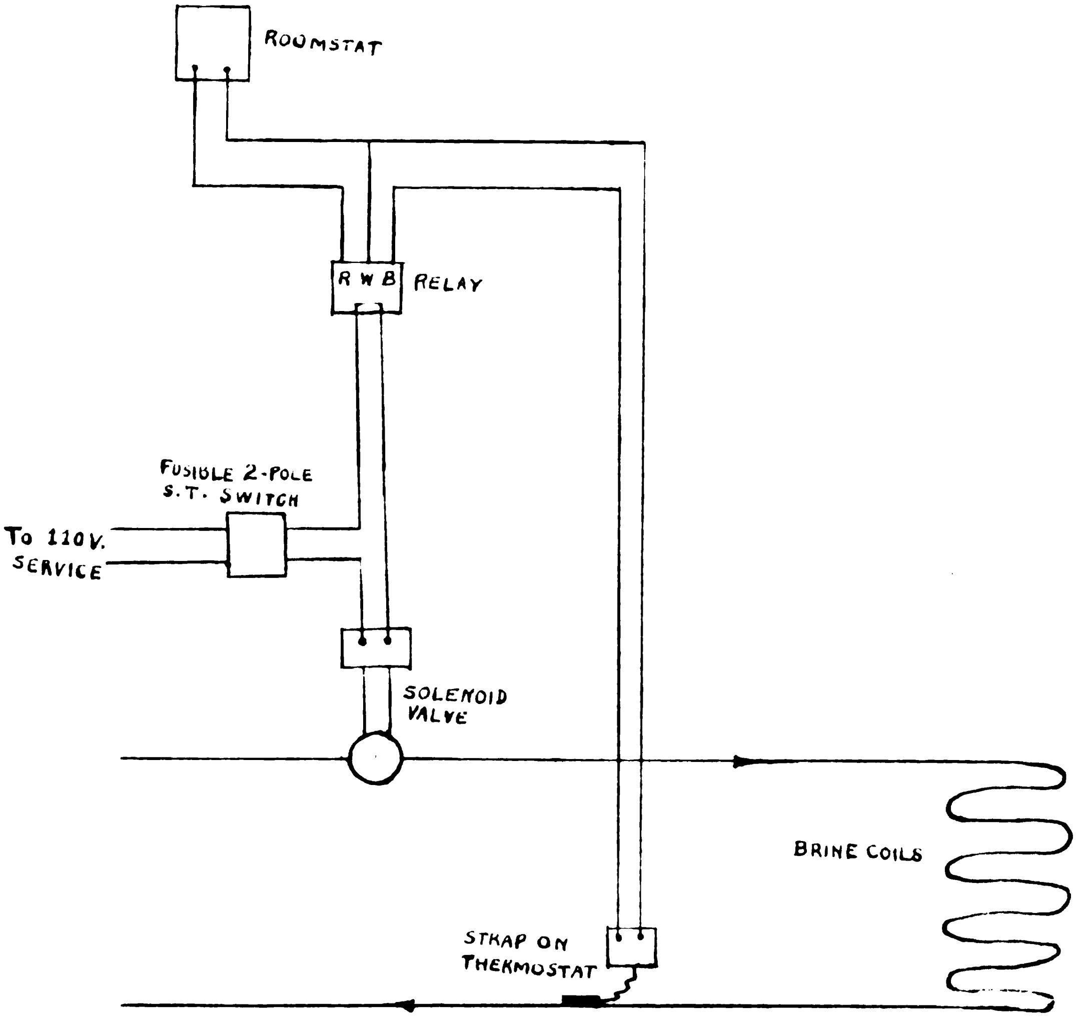


Fig. 25. Automatic Thermostatic Control for Cold Storage Room.

when all the room thermostats are in a satisfactory position. This feature can be incorporated by adding an additional contact on each room thermostat, these contacts being connected in parallel into the control circuit on the magnetic starter of the brine pump. This will allow the pump to be held in operation as long as any one room requires refrigeration.

D. Ice Manufacture

Use of ice in refrigerated railway cars in the distribution of produce, in cool storage and in serving food and beverages in hot summer months in India is indispensable.

Most (solid) artificial ice is made by the can system, which will be discussed here. In this system, tapered cans of standard rectangular cross section are immersed in a moving bath of chilled brine until freezing is completed.

Capacity

Refrigeration required to produce ice depends on the heat necessary to cool water available for making ice to the freezing point, to freeze it and then to cool the ice to the temperature of the circulating brine in the tank. If initial temperature of cool water available for filling in of cans during summer months be about 75°F. and the temperature of brine assumed is 14°F., the total refrigeration per pound of ice would be equal to sensible heat before freezing plus latent heat plus sensible heat after freezing

$$= (75 - 32) + 144 + 0.5\,(32 - 14)$$

$$= 196 \text{ Btu.}$$

Adding 20 per cent insulation and other losses

Total refrigeration per lb. of ice = 235 Btu.

For this set of conditions, one ton of ice would require

$$\frac{235 \times 2000}{288000} = 1.635 \text{ tons of refrigeration}$$

Since brine cooling capacity of the submerged cooler or vertiflow unit evaporator in the ice tank is 16.23 tons, the total capacity of the ice plant in tons of ice would be $\frac{16.23}{1.635}$ = 10 tons.

Number and Size of Ice Cans

Various sizes of standard ice cans are now available. Most commonly used size is 300 lb. cans with 11" x 22" inside dimensions at the top with a 1" taper and 50 in. long, having No. 14 gauge of steel all welded and galvanized. Weight of this can would be about 90 lbs.

The number of ice cans required per ton of ice per day depends on the weight of ice block and the freezing time of ice. If freezing time of ice is exactly 24 hours, the number of ice cans per day per ton of ice

$$= \frac{2000}{\text{Weight of ice blocks}}$$

The longer the freezing time required, the more cans will be required in the same proportion, or

Number of cans/day/ton of ice

$$= \frac{2000 \times \text{Freezing time in hrs.}}{\text{Weight of ice blocks} \times 24}$$

Since it has already been found that with 14°F. brine temperature freezing time for the standard 11" x 22" tanks is 45.6 hrs., the number of cans per day for a 10 ton ice plant will be

$$= \frac{2000 \times 45.6 \times 10}{300 \times 24} = 126.5$$

or, $= 144$, allowing for safety.

Ice Tank

Ice freezing tank should be of 1/4 in. riveted steel or welded construction. It should be 53" deep for 50" long cans. The level of brine in the tank should be such that the top of the finished ice block is about 1-1/2 in. below the level. Tank dimensions will be about 15 ft. x 30 ft. if we put 12 cans in each row and column with a race way at an end. Insulation of brine tank has been discussed earlier.

Frame Work and Covers

In this plant, two cans can be harvested at a time and standard wooden frame work and covers can be used in place of steel can grid, as for big plants. This frame work, constructed of heavy selected oak, is slotted where necessary to receive the air laterals and to permit the passage of return air going to the blower (See low pressure air system). Suitable hold-down strips of spring steel keep the cans in a vertical position in the brine, also insuring proper submergence. The tank covers are made of two layers of 7/8 in. lumber with a layer of waterproof, fungus-resistant paper between them. They are dipped in hot linseed oil to make them moisture proof. (See Fig. 26).

Evaporator

As already mentioned, either single pass shell and tube submerged brine cooler or trunk and the brine race type evaporator known as vertiflow unit is economical of space and eliminates piping between the cans. The flooded system of evaporation is employed which permits high values of "U" for the evaporator surface, about 80 to 100 Btu/hr/sq.ft. Both these types of evaporators are shown in the accompanying diagrams (Fig. 20 and 27). Evaporators are located in the end and ice tank divided in compartments by partitions to achieve rapid circulation of brine caused by brine agitator. Proper brine velocities around the evaporator and the ice cans have already been discussed.

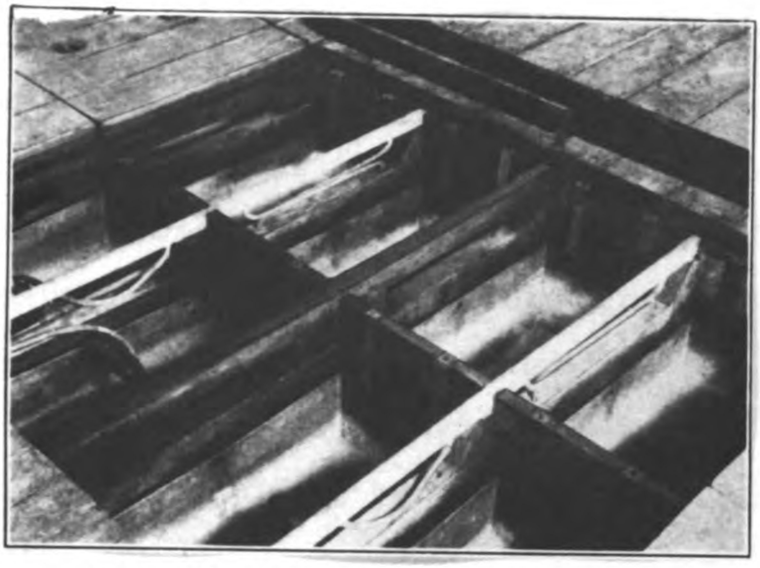


Fig. 26. Wooden Framework and Covers in Small Ice Plant, showing Air Laterals.

Fig. 27. A Vertiflow Unit.

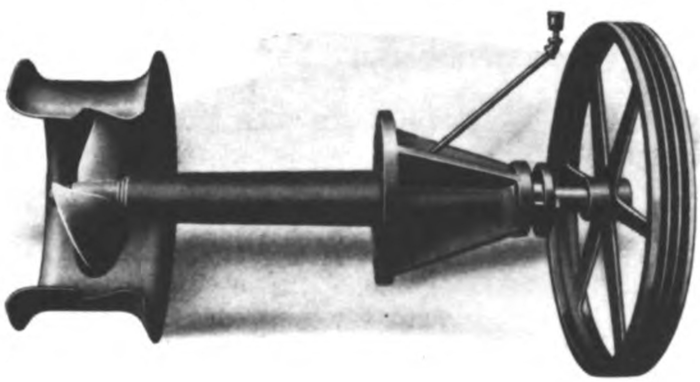


Fig. 28. Assembly of Horizontal Agitator.

Brine Circulator

V-belt driven horizontal brine circulator or agitator should be located at the entrance of the brine to the evaporator. It is equipped with propeller for moving correct quantity of brine, about 60 to 70 gallons per ton per minute when the ice is being manufactured. Brine passages should be carefully designed to give proper brine velocity and reduce loss of head. The brine circulator with stuffing box and driven pulley is shown in Fig. 28.

Low-Pressure Air System

Although water for ice making may be clean, it nevertheless contains dissolved gases and solids; and if frozen quietly, the ice would not be transparent. To make clear ice from ordinary drinking water (without going to the expense of distilling it), the water is agitated while it is being frozen in the cans. Air motion sweeps away dissolved gases and prevents them and solids from freezing into the cake. For this purpose a stream of air is passed through water. Better agitation system is using air at a very low pressure which gives clear ice blocks with cupped tops. Approximately ½ cfm of free air is supplied at about 2 lb. per sq. in. for each can.

In this system the air passes through a pendulum tube suspended in the middle of each can by means of pivot so that the tube is free to swing across the can from side to side but is restrained from motion in any other plane. The result is that the tube acquires a pendulum motion in the water, allowing the agitation to wash thoroughly each of the flat surfaces of the ice in turn and collect impurities of water at the central core. When the ice block is about to be frozen completely, the central core cold water with its entrained impurities is removed by a suction nozzle connected to the pump and core or cavity is refilled by fresh drinking water, which gets frozen. Before freezing of the core takes place the pendulum tube is removed. A small blower is employed to supply the air at low pressure. The air is drawn into the blower from under the tank top, where cool supply is always available. Air blower may be rotary type belt or motor driven.

Auxiliary Equipment

In addition to main parts of the ice plant, the following equipment is also needed:

A hand operated single can dump is furnished with drip pan and with or without sprinkler pipes for thawing the ice loose from can, and is kept on thawing plat-form adjacent to ice tank.

Single can filler is used in the small plant for filling the cans with fresh supply of water after the cans are replaced in the brine tank, It is connected to the water line with a hose. A float valve in the center of the filler acts to trip the water valve, cutting off the supply when the water has reached the desired height in the can.

Small hand or electric hoist mounted on hand pushed crane with a single I-section cross beam can be employed over the ice tank with rails for pushing the I-beam.

Storage Room

In India generally ice is disposed of the very day to wholesalers and retailers. If the storing of ice becomes necessary the storage room of suitable size should be constructed adjacent to thawing platform and refrigerated by direct expansion coils or brine coils led from the brine tank.

Harvesting Time

Since total freezing time is around 48 hours it will be a good practice to harvest ice twice daily--say at 8 A.M. and again at 8 P.M. (amount harvested each time would be half the total daily capacity). This will distribute the refrigeration load evenly throughout freezing process.

Automatic Ice Plant

Although ice plant is assumed to run continuously for 24 hours, it will usually give satisfactory control of brine and refrigerant temperatures consistent with load due to float type expansion valve. Compressor motors are already equipped with low pressure switch, excess pressure cut-out and other safety devices. The plant can be made fully automatic by adding thermostatic control from the brine to the compressor motors in parallel with that from brine cooler for room coils, and a cut out in case of decrease in brine flow for any reason whatever. Under such condition, the brine temperature will perhaps vary over a wider range from 12^{o} to 15^{o}F. and freezing time will have to be adjusted to suit the conditions. The coils may supply refrigerant for 20 hours a day, in which case capacity of ice manufacture would reduce to about 8 tons instead of 10 tons.

VII. COLD STORAGE PLANT AND EQUIPMENT (Contd.)

A. Ammonia Condensing Equipment

The application of ammonia refrigerating compressors and condensers includes the following important considerations:

(1) Selection of Compressors and condensers to match the refrigeration load.

(2) Balancing compressors and condensers' capacity with the evaporators.

(3) Selection of motor and drive.

(4) Refrigerant piping for the system and choice of accessories.

(5) Selection and application of safety devices, controls and interlocking of controls with fans, pumps, etc. in the same system.

(6) Estimate of power and water cost.

<u>Number of Compressors</u>

It was suggested in earlier section that three ammonia compressors of equal capacity may be employed for the cold storage plant. Three units will give flexibility of performance, multiple units capacity control, besides facilitating maintainance and repairs. Normally, all the three compressors would function, but if in certain seasons, ice plant is not operated or load in the cold storage rooms is less, two compressors may be sufficient, the third one serving as a standby. If during long period of storage breakdown occurs in any one unit, ice plant may be shut off and cold storage only may run on the other two compressors, which are of ample capacity to cope with the refrigeration load of rooms during storage season.

Number and Type of Condensers

Due to high cost and restricted supply of city water for condensor purposes and due to local municipal regulations prohibiting the discharge of large quantities of such water into the sewage systems, the evaporative condenser is well adapted to conditions in Northern India. Greatest saving of city water will result in summer months when the plant is working and when there is heavy drain on municipal water plants. Besides, in tropical climate city water supplied from open tanks in summer gets very hot during the afternoon and may reach 100°F. when the ambient air temperatures go beyond 105°F. City water normal temperature may be 90°F. Looking to the metreological chart prepared for various towns and cities in India where cold storage plant for potatoes would work, it will be found that the average wet bulb temperature during the months for places in northern

India plains varies between 70° to 75°F., even though dry bulb average goes beyond 85°F. Hence, evaporative condenser would offer additional advantage of cooling the city water to near the prevailing wet bulb temperature. If conventional type condenser is employed, the hot city water will have to be cooled separately in cooling tower. Hence, from the point of view of economy of operation, space occupied and perhaps the first cost, the evaporator condenser is possibly the best answer to our problem.

For total refrigeration tonnage needed by evaporators, one condenser would be selected as, throughout the working of the plant, the total refrigeration load of cold storage and ice tank is the same.

Compressors' Ratings

Refrigerating compressors are rated, according to ASRE Standards, for the suction pressure corresponding to saturation temperature of 5°F. and discharge pressure corresponding to saturation temperature of 86°F. There is a change in compressor refrigeration capacity for pressures other than the above, increasing from minimum at low suction temperature and high compression ratio to maximum at high suction temperature and low compression ratio. This is explained as follows:

(1) The cfm/ton of refrigerant gas required is

greatest at low suction temperature since the gas is in a rarified condition at the lower temperature. As the gas density increases at higher suction temperature, the cubic feet per pound or specific volume becomes relatively smaller, and therefore, the cfm per ton becomes less.

(2) The volumetric effficiency, i.e. $\frac{\text{cfm pumped}}{\text{cfm displaced}}$ of the compressor, which covers volumetric efficiency due to clearance increases as the compression ratio decreases. Therefore, since there is a relatively narrow range of economical condensing temperature for single stage compression, it follows that operation at the higher suction temperatures will usually result in lower compression ratio.

Condenser Ratings

Evaporative condenser is also rated (in tons refrigeration effect) for saturated condensing temperature which again depends on outside (air) wet bulb temperature, saturated suction temperature of compressor (as this would determine the temperature of discharge gas to be condensed) and the amount of subcooling, if obtainable.

Selection Data

Hence, before we select the compressors and condenser for required refrigeration load in the evaporators, the following further information is necessary:

(1) Suction temperature and pressure at the compressor.

(2) Discharge pressure at the compressor.

(3) Cooling water temperature or design wet bulb temperature of air in case of evaporative condenser.

(4) Degrees of liquid subcooling, if obtainable.

With this information in hand selection may be made of both compressors and condenser either from the published data on ratings of the two equipments or most practical p - H chart may be drawn and figures worked out.

Suction Temperature

The evaporating temperature of 7.5°F. average has been assumed for brine cooler and submerged coils or cooler in ice tank in our previous discussion. It was also pointed out that in flooded systems static pressure of the liquid refrigerant in the coolers or coils may cause the lower layer evaporating or boiling temperature to be higher, say 9°F., than that at the surface of the boiling liquid. Since compressors pump the gas from the upper suction header or surge tank of evaporators, the temperature corresponding to the suction pressure of the compressor may be 1° to 2° F. lower than that prevailing in the evaporator coils or cylinder. Hence, it is safe to assume this suction temperature to be 6°F.

Choice of Condensing Temperature

The capacity of the evaporative condenser should be selected for the mean maximum wet bulb for the most extreme month of the period for which cold storage is in operation, for the locality. There will be wet bulb temperatures greater than the mean maximum for the extreme month, but they ordinarily prevail for only a few hours at a time. If, during operation the wet bulb exceeds that for which condenser was selected, the condensing temperature may rise and either overload the compressor motor or cause the high pressure cut-out to stop refrigeration at a time when it is most needed. Thus, the choice of wet bulb is more important in the selection of an evaporative condenser than that of air temperature in refrigeration load estimate (given earlier), where a more extreme temperature reading causes only a temporary failure to maintain the estimated room conditions.

The mean maximum or maximum average monthly temperature for any locality is determined from Metreological chart compiled for the most extreme month, usually July or August in Monsoons in India. As previously noted the average wet bulb temperature for the period for places in Northern India plains varies between 70 to 75^{o}F. The maximum may be assumed to be 80^{o}F. which is the design wet bulb temperature.

Assuming water to be brought to within 10° of this wet bulb temperature in the evaporative condenser under normal operation, the cooling water temperature of the condenser may be taken as 90°F. as a safe value, a further 15° temperature difference between refrigerant condensing temperature of 105°F., with a possible sub-cooling of the refrigerant in the condenser to, say, 100°F. The condensing temperature on the compressor end may be assumed 2° higher, i.e. 107°F.

Note: The condensing temperature selected should result in a balance between investment in equipment and operating cost. Since the condensing temperature is dependent on water temperature or air wet bulb temperature, the latter values should be the average for the period taken, rather than the designed maximum as shown above, when used for the purpose of analyzing operating cost. These values, as stated, vary between 70° to 75°F. Actual analysis for various condensing temperatures for an economical balance between power and water cost can be made by making trial condenser selections for local rates and optimum temperature relationship found. The value of 105°F. assumed above is based on judgment only.

Refrigerant Temperatures

The actual temperature of the refrigerant vapor at the compressor suction valve may be about 15° higher than saturation suction temperature due to superheat in the accumulator, small insulated refrigerant line and compressor valves. Therefore, actual suction temperature may be assumed to be $20^{o}F$. If this gas is compressed in the cylinder from $P_1 = 35.09$ to $P_2 =$ 236 psi abs. according to law $PV^n = C$, where $n =$ 1.28 psi abs., for ammonia systems, the final temperature of gas leaving the compressor is obtained from the formula

$$t_2 + 460 = (t_1 + 460) \times \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

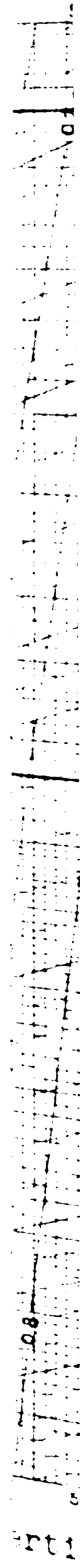
$$= 480 \times 1.515 = 728$$

giving $t_2 = 268^{o}F$.

Some cooling takes place in the discharge lines leading to the condenser and we may assume the temperature of ammonia entering the condenser to be $240^{o}F$.

Refrigeration Cycle

Having determined all the refrigerant temperatures with corresponding pressures, we are now in a position to draw complete refrigeration cycle on p - H (Molliers) chart for one lb. of ammonia, showing superheat at the compressor inlet, actual discharge temperature, and subcooling. For computation ideal adiabatic (isentropic) compression line is also shown which cuts the discharge pressure line at $280^{o}F$. The complete chart is shown in Fig. 29. The following important values on enthalpies,



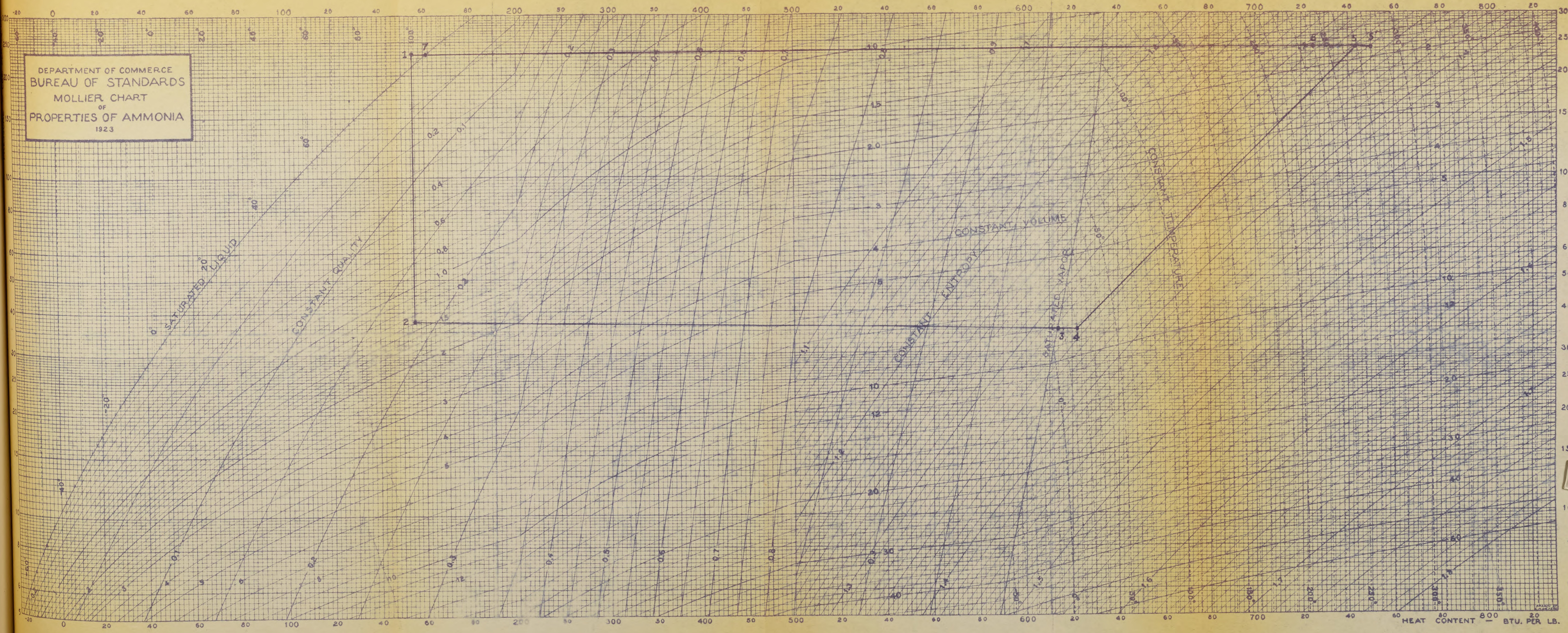


Fig. 29. Properties of Ammonia
Refrigeration Cycle

Supplement to
Refrigeration Theory and Applications
Published by Nickerson & Collins Co.
433 N. Waller Ave. Chicago 44, Ill.

specific volumes and entropies are obtained:

Point	Temp.	Pressure abs.	Sp. Vol.	Enthalpy h	Entropy S	State
1	100	228.9	--	155.0	--	Subcooled liquid
2	9	37.63	--	155.0	--	Mixture--18% quality
3	6	35.09	--	613.6	--	Dry saturated vapor
4	20	35.09	8.287	622.0	1.3413	Superheated vapor
5	268	236.0	--	745.0	--	Superheated gas
5^1	280	236.0	--	750.0	1.3413	" "
6	240	236.0	--	725.5	--	" "
7	105	228.9	--	161.1	--	Saturated liquid

From the table, the following important results are obtained:

(1) Refrigeration effect = $h_3 - h_1^* = 613.6 - 155 = 458.6$ Btu/lb.

Pounds per minute for ton of refrigeration

$$= \frac{200}{458.6} = 0.435$$

Now, the total refrigeration effect of evaporators is 35.6 tons.

Weight of refrigerant circulated per minute

$$w_a = 35.6 \times 0.435 = 15.5 \text{ lbs.}$$

(2) Condensing effect = $h_6 - h_1^* = 725.5 - 155 = 570.5$ Btu/lb.

Total heat rejection by condenser = $w_a (h_6 - h_1)$

$= 15.5 \times 570.5 \times 60 = 531{,}000$ Btu/hr.

---

* Although during the processes pressure varies slightly heat gained or rejected by the refrigerant may be taken as the change in enthalpy.

(3) Work of adiabatic compression $= h_{51} - h_4 =$

$750 - 622 = 128$ btu/lb.

Ideal cylinder H.P. $= \frac{128 \times 15.5 \times 60}{2545} = 46.8$

Ideal I.H.P. for each of three compressors

$= \frac{46.8}{3} = 15.6$

Assuming compression efficiency of 85 per cent, due to cylinder loss, Actual I.H.P. for each compressor $= 18.35$.

Also, assuming compressor mechanical efficiency of 85%, the B.H.P. of each compressor $= 21.6$

(4) Volume of suction vapor in the compressors

$= v_4 \times w_a = 8.287 \times 15.5 = 128.5$ cfm.

Actual suction volume of each compressor $= 42.8$ cfm.

Assuming overall volumetric efficiency of 70 per cent for the given suction and discharge pressures, the actual piston displacement of each compressor

$= 61.15$ cfm.

Compressors--Design and Construction

Vertical single acting enclosed type ammonia compressors are generally used in ice-making and cold storage application due to their many inherent advantages. These include balanced and high speed vertical operation, adaptability to any kind of drive and to automatic control, one way gas travel, efficient compression, quietness and ease of operation, continuous automatic lubrication, small floor space required, prompt installation and long useful life. Multiple cylinder units and more than one unit offer flexibility of operation and capacity control.

Since piston displacement of each compressor is 61.15 cfm, a 6" x 6" size two-cylinder single acting compressor running at 330 rpm will do the job. The compressors should be water cooled and fitted with manifold assembly including ammonia gauges, main stop valves, purge valves, by-pass valves, high pressure relief valve and suction scale trap. It should be fitted with main ball bearings and with outboard ring bearing to carry large multiple V-grooved pulley for being driven by a motor. Belt guards are provided for drive. Large sizes generally have full forced-feed lubrication to all internal wearing surfaces. The shaft packing is of seal ring type submerged in oil under pressure with automatic tension adjustment. All oil for charging the compressor initially is supplied by refrigerating contractor. A vertical forced-feed lubricated ammonia compressor is shown in section in Fig. 30.

Compressor Motor

In making the motor selection for compressors, the full load brake H.P. required for refrigeration should never exceed the recommended maximum loading for each

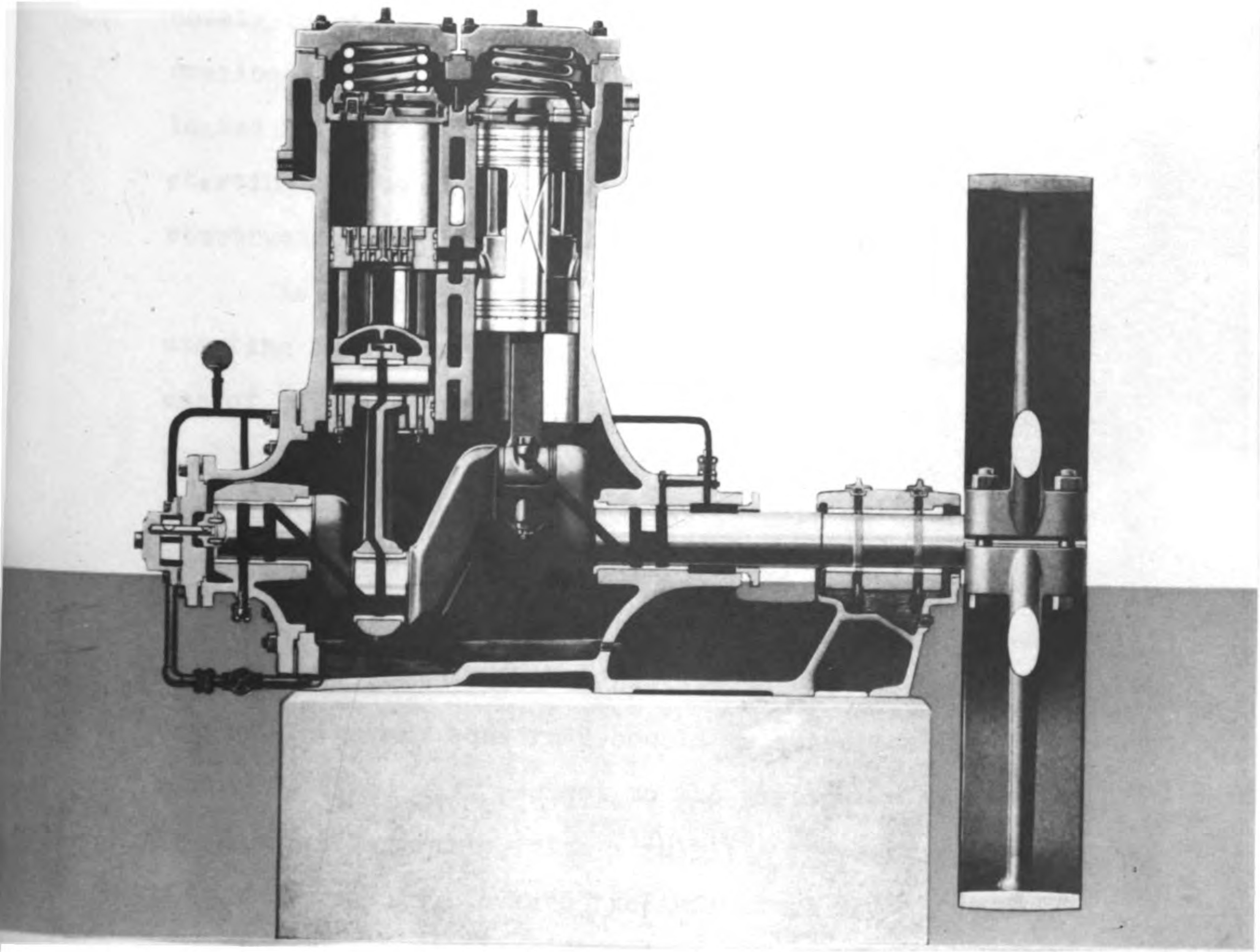


Fig. 30. Sectional View of Vertical Tow-Cylinder Enclosed Ammonia Compressor, showing Forced-Feed Lubrication.

motor size as indicated by their ratings. Since H.P. requirements are 21.6 per compressor, 25 H.P. continuously rated three phase induction motor with temporary overloads would be needed. Since the compressors start loaded for automatic operation, they require high starting torque motors with special double bar rotor construction.

As a general rule, across-the-line or "direct-on" starting is the most desirable for automatic system. The use of "direct" magnetic starter simplifies the control problem, is less costly and usually more trouble free than the more complicated reduced voltage or "star-delta" equipment. However, limitations of the distribution system capacity often require the use of reduced voltage starters for compressors in certain range. In such case, the power company concerned should be consulted and ruling obtained with respect to the particular application. It is often desirable, with a multiple compressor installation such as this, to have the individual units interlocked by means of time relay devices so that no more than one motor at a time will demand the starting current.

Motor Protection

It is essential that selection of the control equipment for motors for the refrigeration service include no-volt release and overload protection. Fuses and circuit breaker alone will not constitute adequate protection since these are sized to handle the required starting current. Another reason is that the blowing of one fuse in a three phase system will permit"single phasing"of the motor, which would then be destroyed. A no-volt coil and overload thermal relays are usually incorporated in the motor starter.

Isolation of Vibration

Heavy density isolation cork board for each compressor is used in the foundations to prevent noise due to vibration. Standard vibration isolation rubber package for smaller size compressors is available.

Evaporative Condenser Installations

Evaporative condensers are designed for either outdoor or indoor installations. For best results, evaporative condenser should be installed in a position where outside air can be drawn into the unit and discharged outdoors without the use of long ducts. It can be located in the machine room close to the compressors. The most economical system will result when the unit is located to allow short hot gas lines, liquid lines and short duct connections to the outside. The discharge duct outlet should direct the air away from the intake and into a space subject to the free moving wind current.

The evaporative condensers are rated on wet bulb temperature of entering air, saturation temperature of the refrigerant corresponding to a given condensing temperature and volume of air through the unit. Standard ratings for ammonia condensers are made for operation at 78°F. wet bulb temperature and leaving liquid refrigerant temperature of 105°F. corresponding to a gauge pressure of 214 lbs. per sq. in.

Water Requirements

The capacity of the condenser must be sufficient to remove the heat absorbed in the evaporator (refrigeration effect) and heat of compression added to the gas by compressor. This sum has been found to be 531,000 Btu/hr. Since the latent heat of vaporization of water to be evaporated under air conditions of about 100°F. dry bulb and 80°F. wet bulb is about 1040 Btu/lb., theoretically, $\frac{531000}{1040 \times 60} = 8.5$ lb. or 1.02 gallons of water are required per minute. But in practice, about 3 gpm are supplied with about 2 gpm going as waste in carry over along with air discharged and for dilution of the available supply in the base. The quantity of water pumped can easily be placed at any desired figure which keeps the tubes well wetted. In general 1 to 2 gpm of water per ton of refrigeration are circulated. So, about 50 gpm will be a good figure for this condenser.

Air Circulation

The circulated spray water leaving the bottom tubes will have a temperature which lies between the original wet bulb temperature of the air and the temperature at which the refrigerant condenses in the tubes. Increasing the quantity of air circulated brings the water temperature closer to the wet bulb temperature of entering air. This increase in the range from 150 to 300 cfm per ton shows justifiable gains, but above this point the rate of gain is so small that the fan power costs begin to offset gains from decreased water temperatures.

Assuming that the water leaving the condenser tubes is 10° higher than initial wet bulb temperature of the air, i.e. water temperature is 90°, the air discharged from fans will be nearly saturated at that temperature or at a temperature higher than this due to its contact with hot refrigerant in the upper section of the condenser pipes. The amount of moisture gained by air will be the difference of moisture contents in its original condition, i.e. 100° dry bulb and 80° wet bulb and final condition, i.e. 90° F. saturated. This difference is found to be 95 grains of moisture per lb. of dry air. Latent heat of vaporization of water vapor at normal atmospheric pressure and average air temperature of

$95^{\circ}F.$ is 1040 Btu/lb. Hence, to evaporate 95 grains of water from its initial temperature of 90° to saturated vapor condition at $90^{\circ}F.$, we require $\frac{95}{7000} \times 1040$ $= 14.1$ Btu of heat, which will be extracted from refrigerant and surroundings. For indoor installation, assuming condenser heat rejection effect to be 1.1 times the net condensing effect which is 531000 Btu/hr, the volume of air (taking 14.5 cu.ft./lb. of air) would be

$$\frac{14.5 \times 531000 \times 1.1}{60 \times 14.1} = 10000 \text{ cfm}$$

Hence, three 12" dia. fans running at about 1450 rpm by a direct coupled motor would each give about 3300 cfm discharge and consume total 3 H.P. approximately. This gives about 280 cfm per ton which is within the range suggested for economical running.

Design of Condenser Pipes

Coefficient of heat transfer for condenser pipes "U" is taken between 12 to 60 Btu.per hr. per sq.ft. external surface per deg. F., depending on the amount of water sprayed and extent of rust or scale formation on the pipes. For normal operation, assuming fairly clean pipes, an average figure of 50 may be taken. For design purposes the average refrigerant temperature may be taken to be $110^{\circ}F.$ as some coils at the top have superheated gas at high temperatures and lower one or two coils may have subcooled liquid at $100^{\circ}F.$ Water temperature is about 90°F. throughout. Hence, surface area of coils required for condensing effect (based on 24 hrs. running) will be

$$\frac{531000}{50 \times (110 - 90)} = 531 \text{ sq.ft. of external area}$$

giving about 1940 feet linear length of 3/4" full weight steel pipes. An evaporative condenser built in 3 sections is shown in the Fig. 31.

Ammonia Receiver

In a refrigeration system, the liquid refrigerant receiver serves a two-fold purpose:

(1) To provide a liquid seal, thereby preventing refrigerant gas from entering the liquid line to the evaporator.

(2) To provide a space for refrigerant storage to house the entire charge during a shut down period and to store the excess amount of refrigerant over and above the actual amount required for operation.

In sizing a liquid refrigerant receiver, the amount of refrigerant in the system will have to be ascertained. Receiver should be so sized so as to allow a 20 to 25 per cent gas expansion, as a safety factor. The ammonia level in the receiver should be carried between 1/3 to 3/4 full at all times, if possible.

Receiver is provided with inlet valve, liquid valve, charging valve and necessary ammonia fittings

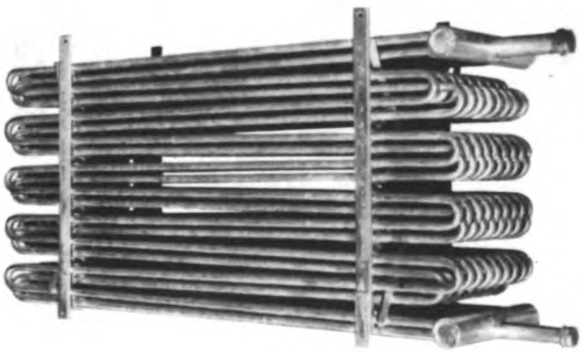


Fig. 31. Evaporative Condenser. (Left) Coil Assembly.

and connections. Sometimes receiver is equipped with steel automatic gauge cocks with glass and guards. A safety valve and a purge valve are provided on the top and a drain cock at the bottom for draining oil and sludge, whenever formed. Being a pressure vessel, receiver is tested with air at pressure of 300 lbs. per sq. in. under water before shipment.

Amount of Refrigerant

The quantity of anhydrous ammonia needed for the compression side of a refrigerating system is generally estimated by assuming the whole space of the condenser to be filled with ammonia vapor at standard pressure corresponding to condensing temperature of 86°F. and allowing 25 lb. of liquid ammonia for every cubic foot of liquid receiver capacity. The following table is based on experience:

Tons of Refrigeration	5	10	20	40	100	200
Ammonia, lbs.	110	150	230	300	440	620

Hence, in our system, 35.6 ton refrigeration, about 280 lbs. would be used. This should leave about 35 to 40% of liquid ammonia in the receiver when the plant is in full operation. To this figure must be added sufficient refrigerant contained in the two flooded evaporators filled entirely to the float level.

B. Accessories

Oil Separator or Trap

It is installed in line between the compressors and the condenser to remove oil from the discharge gas and return it to one of the compressor crank cases before it reaches the condenser and the evaporator side of the system to affect the heat transfer adversely. Oil laden gas enters a perforated steel pipe or mesh and is forced through an impingement type filter where oil and gas are thoroughly separated. Oil accumulated in the bottom of the separator shell will open a float valve and be returned to the crank case by discharge pressure.

Non Condensable Gas Remover

It is one of the most important accessories to improve efficiency and safety. It removes air and decomposed ammonia and oil products of a gaseous nature from the system and thereby maintains low head pressures, which will reduce power consumption and increase refrigerating capacity. Ammonia gas containing the foul gases is led from the top of the receiver into the gas purger, through which cold suction main passes. Contact with cold main causes the ammonia in the mixture to condense and is fed back into the system by a float control. The foul gases do not condense and are drawn off through a purge line to a water bottle which absorbs any slight odor of ammonia and allows the foul gases to escape to atmosphere. Valves are provided to regulate the various pressures and flows.

Ammonia Purifier

It is very useful device for removing oil, water and other foreign matter from the system. This centrifugal separator is placed in the discharge line between the compressor and condenser after the oil trap mentioned earlier, and removes the water, oil and other impurities very efficiently.

Thermometers

Thermometers for brine inlet and outlet, for brine tank (hanging type), for ammonia lines and for storage rooms (recording or dial type) are needed for satisfactory running of the whole plant.

Humidifying Equipment

It is assumed that brine temperatures are such that they would give proper room temperature and humidity both during storage and loading and no artificial methods of increasing humidity would be needed. But, if the loading conditions vary considerably and operating conditions are also such that it is no longer possible to maintain the required room conditions with existing plant, it would be necessary to purchase auxiliary humidifying equipment consisting of a set of turbo-humidifiers with an air compressor with suitable receiver and after cooler.

C. Refrigeration Piping and Control

Ammonia Pipes

Theoretically, for temperature conditions of refrigerant in the cold storage plant, about 3.5 cfm of ammonia gas is to be pumped in the system per 1 ton of refrigeration. Probable volume for actual refrigeration cycle may be taken about 30 per cent higher. Hence, actual volume of ammonia pumped in the system would be

$$1.3 \times 3.5 \times 35.6 = 161 \text{ cfm.}$$

Allowing permissible velocity of gas through suction pipe to be 4000 fpm, $\frac{4000}{161} \approx 24.8$ feet of a pipe line should occupy one cubic foot. Referring to tables giving data of extra heavy ammonia pipes, we find that 3" nominal size pipe would be 21.8 ft. per one cu. ft. of gas and hence, is of quite suitable size.

Again density of ammonia liquid at 5°F. is 42.5 lb/cu.ft. We have found ammonia circulated to be 15.5 lb/minute, which is equivalent to 0.365 cfm of liquid. Allowing 120 fpm as the velocity of refrigerant in the liquid line, cross sectional area of pipe will be

$\frac{0.365}{120} \times 144 = 0.437$ sq.in. Hence, 3/4" nominal size extra heavy steel pipe will be required for the liquid line.

Other pipe sizes may be selected arbitrarily in conformity with the above sizes found and whole refrigerating system piping will be as shown in the flow diagram (Fig. 32). Threaded pipes for ammonia require litharge and glycerine.

Equalizers

The three vertical enclosed compressors should have interconnecting piping arrangement by which refrigerant gas pressure in the crank case and suction pressure, as also oil levels are equalized, as shown in single-line equalizing arrangement. (Fig.33.).

Controls

Being electrically operated and intermittently run, the refrigerating system can be completely controlled automatically. Room temperature control by system of room thermostats and liquid solenoid valves in the brine lines leading to unit coolers has already been considered. A completely automatic refrigerating unit requires

(1) Regulation of the flow of the liquid into evaporators,

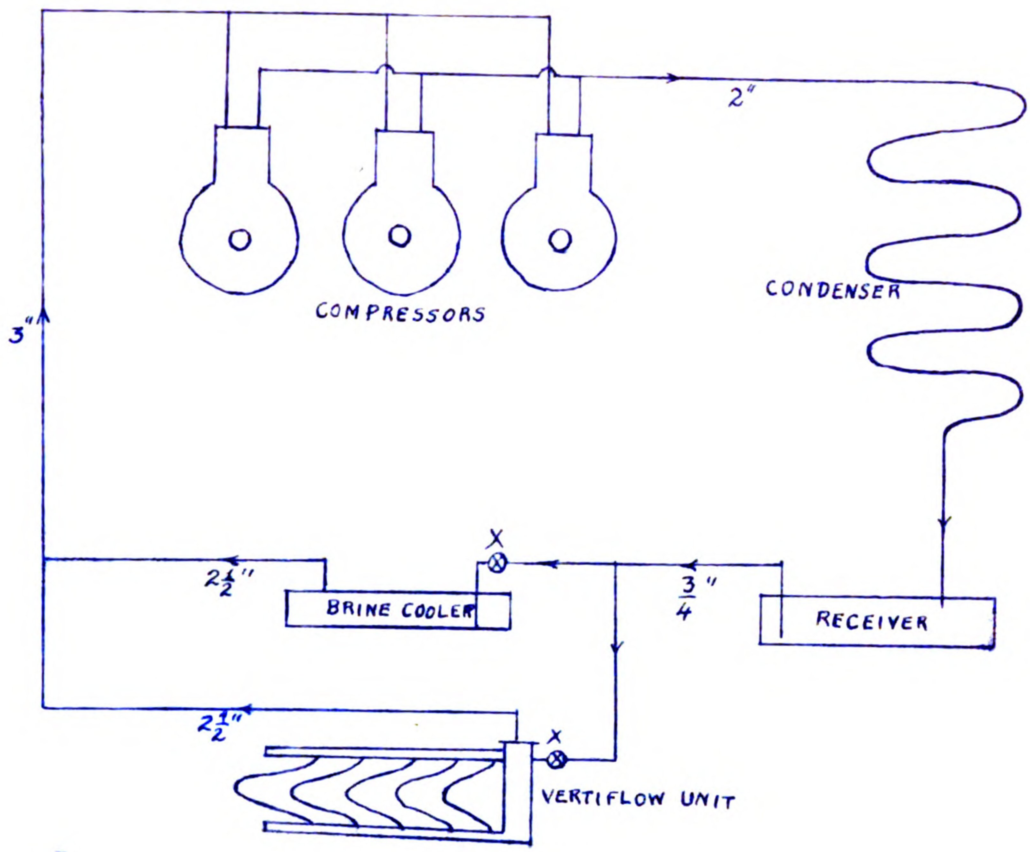


Fig. 32. Refrigerant Flow Diagram, showing Pipe Sizes.

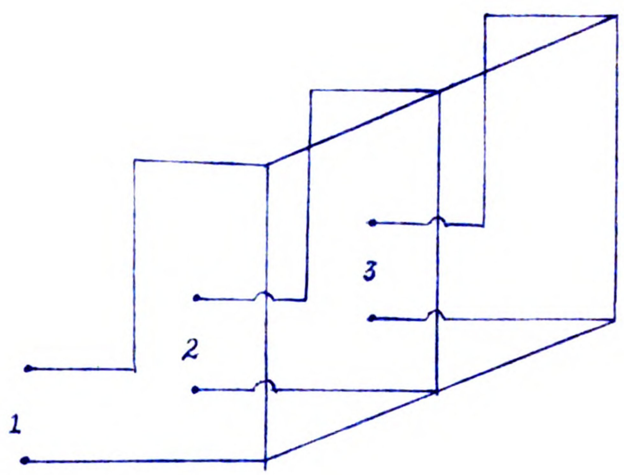


Fig. 33. Equalizer Pipes for Multi-Compressor System.

(2) a thermostatic control to start and stop the compressors

(3) A control on the water supplied to the condenser, and certain other motor-compressor controls.

Liquid Control

As already stated, both brine cooler and the vertiflow raceway coil in brine tank can work on the flooded system. This means that the liquid feed is heavy enough to constantly cover all the heat transfer surface. In order to protect the compressor from being flooded, a vapor separator called accumulator or surge drum is placed in the return header with a means of evaporating the trapped liquid in a suitable manner. To shorten the connection between the liquid header and the suction header in the vertiflow unit, coils are connected to the headers in parallel. The Figs. 34 and 35 show the float valve control to brine cooler and the ice tank vertiflow unit respectively, both equipped with accumulator.

Since both the evaporators operate on same refrigerant temperature viz 7.5°F., it is not necessary to put back pressure regulating valve in any suction line. But in the flooded system, it is recommended that a liquid line solenoid valve be also installed. This valve will close off the liquid line on shut-down of all the compressors and thereby prevent possible overflooding

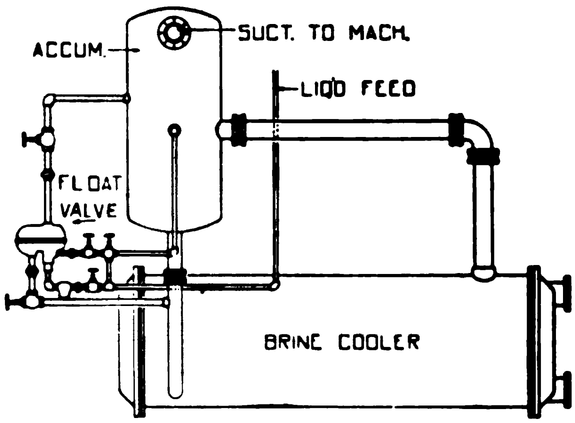


Fig. 34. Diagram of Connections for Brine Cooler when Float Valve Control and Accumulator are Employed.

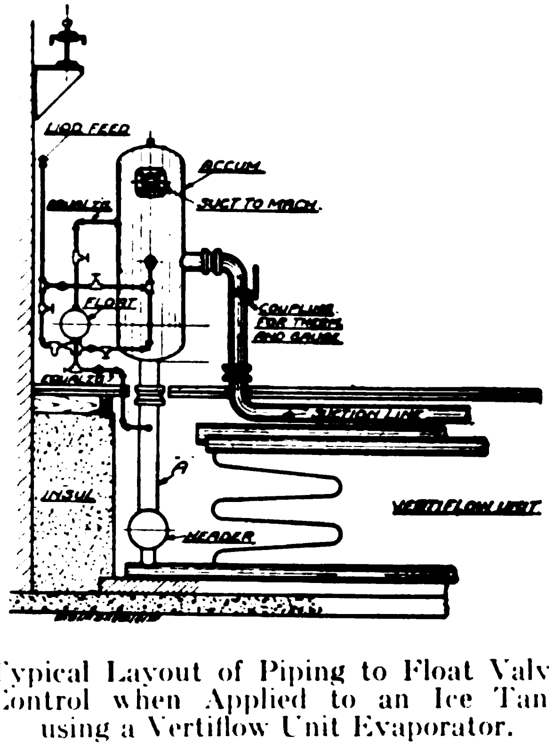


Fig. 35. Typical Layout of Piping to Float Valve Control when Applied to an Ice Tank using a Vertiflow Unit Evaporator.

of the coil due to any leak occurring in the float expansion valve. This will prevent the possibility of liquid slugging back on compressors' start-up.

Compressor Control

Combined low pressure and high pressure cut-out is used almost universally in refrigerating industry. The low (suction) pressure feature serves both as a control and safety function. A rise in suction pressure expands the bellows, which, in turn operates a linkage to tilt the mercury switch and start the compressor. The high pressure cut-out performs the safety function only. The controller or pressurestat has additional bellows for this high pressure cut-out which tilts the same mercury switch to break the motor control circuit in the event of excessive high side pressure. An arrangement with alarm bell for high pressure is shown in Fig. 36.

The controllers have direct reading main and differential scales and pressure and differential adjustment screws are generally at the top of the case. High pressure cut-out is adjustable internally.

It would appear that out of the three compressors, two would be running almost constantly, both during loading and storage periods. Hence, any two compressors could both be set for pressure corresponding to suction

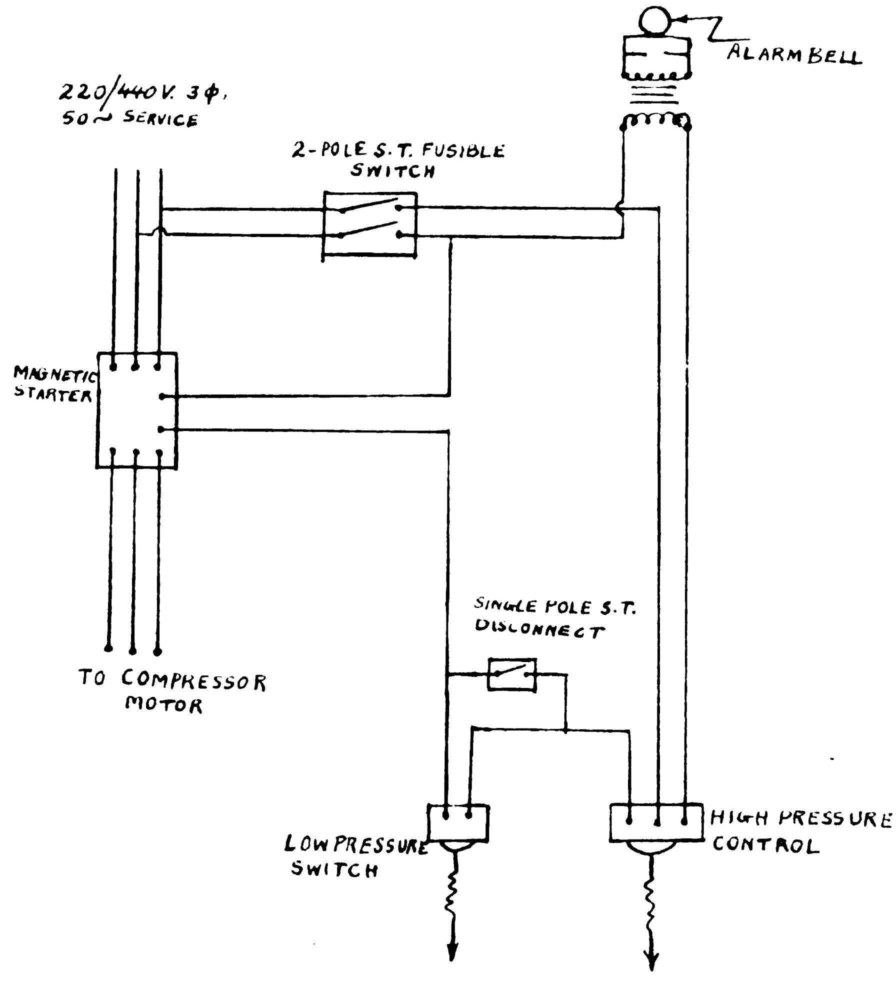


Fig. 36. Typical Compressor Control Diagram.

temperature of little below $5^{o}F.$, or say 18 psi gauge, with a differential of 3 psi and the third compressor may be set to operate at higher pressure, say 20 psi gauge with same pressure differential. Since equal running time of all the compressors is also advisable, there should be a regular schedule of changing low pressure control limits so that first one compressor or group and then another will carry the base load. High pressure cut out may be set for 220 psi. gauge with a pressure differential range of 30 psi so that compressors continue to work economically at low condensing temperatures up to $98^{o}F.$

Water Control

Evaporative condenser is equipped with a float control for regulating the water supply to the spray tank and maintaining an overflow while the unit is in operation. The water after passing through the float valve flows through a water treatment container filled with compound and drops into the spray tank.

In evaporative condenser, no water regulation is necessary, as water spray should constantly cover the condenser pipes as long as the latter remain in operation. But condenser fan motors can be wired with compressors' motors so that condenser and compressors will operate simultaneously. The pump motor of the condenser is wired in parallel with the fan motor. Thus the pump and fan motors will operate only when one or more compressor motors operate.

Solenoid valves can be used for water connection to each compressor so that each valve opens when the compressor cuts in and closes when the compressor stops.

Interlocking Controls

The control of compressors is intimately associated with the balance of the control of the system, that is, the control applied to evaporators, the evaporative condenser and other refrigerating equipment. Accordingly, it is desirable to provide a system of interlocking of the compressor controls with those operating the rest of the system, to assure compressor starting and stopping in proper sequence with reference to the remainder of the system.

In the multiple cold diffuser installation with the liquid (brine) control solenoid valves, wiring should be so arranged that evaporative condenser and compressors can run only when one of the diffusers is in operation. Relays are used to isolate diffuser not in operation.

The use of time relays for interlocking individual compressor units in multiple units installation has already been explained.

Instrument and Control Board

In the machine room, an instrument and control board is installed, on which such devices as gauges, thermometers, and automatic controls for various compressors and other equipment are mounted. All instruments and devices should be wired so as to interlock the system perfectly as explained above.

D. Auxiliary Equipment

Potato Graders

A complete grading equipment for potatoes consists of wood roller elevator, rubber spool grader and wood roller sorting table arranged in series. It sizes, sorts and bags in one continuous operation. Heavy lifting of boxes or crates is eliminated by the elevator which puts the feeding hopper down close to the floor and allows the potatoes to arrange evenly on the rollers before they are fed to the grader. Rubber spool grader eliminates cutting and bruising and sizes potatoes of different shapes accurately. Finally, the sorting rollers turn the potatoes many times as they travel over the table so that a bad potato cannot possibly escape observation.

Daily loading of potatoes in storage rooms is $\frac{1000 \times 2000}{60 \times 30}$ or about 1100 bushels. If we consider loading to continue for 4 hours in the morning and that all potatoes brought from farms need grading, the capacity of the grader should be about 300 bushels per hour. A warehouse model of this size would consume about 3/4 H.P. for the grader to which should be added about 1/2 H.P. for elevator and 1/4 H.P. for the sorter. Overall length of complete unit will be about 20 feet. Maximum width with bag holders on both sides would be about 6 feet. A complete grading and sorting unit is shown in Fig. 37.

This assembly gives reasonable cleaning on the grader itself, but where considerable cleaning is required, especially for muck grown potatoes, a two-way cleaner may be added between elevator and grader which will give extra length of 4 feet on the complete unit.

Handling Equipment

Telescopic high-lift trucks available in the market are designed with low mast that will operate through doorways. The collapsed height is only about 5 ft. overall. They are able to stack high in storage spaces by lifting the pallet up to about 7 ft. height. Overall height of the mast when fully extended is about 8 ft. so that with about 5 ft. height of the crates on the pallet, the stacking can be done up to 12 ft. height in the storage room.

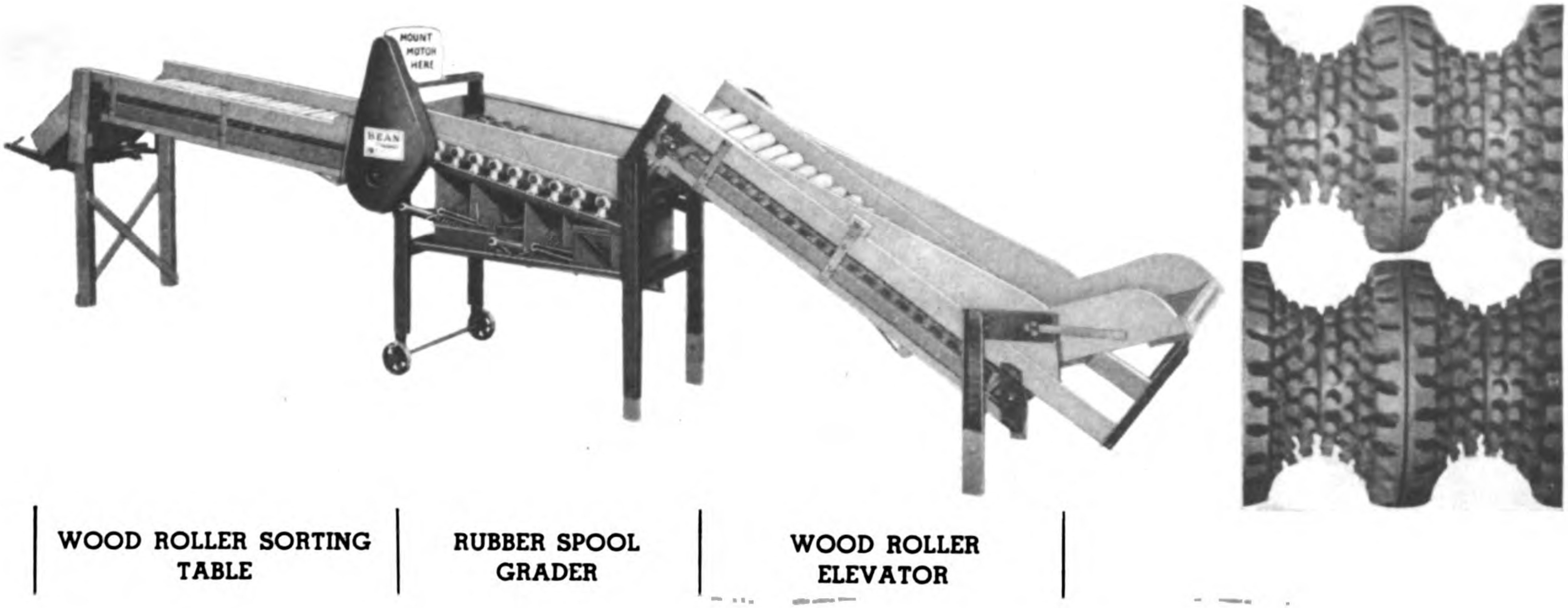
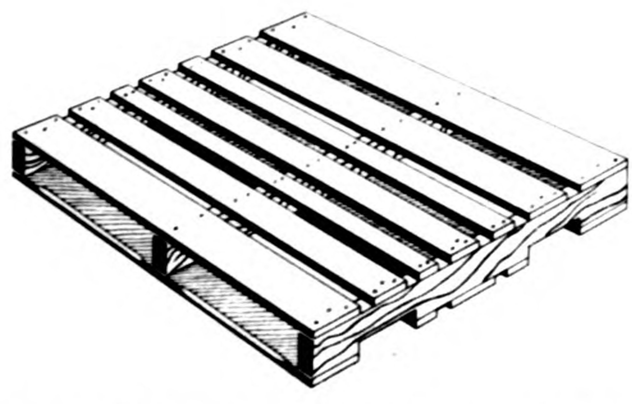


Fig. 37. Complete Grading and Sorting Unit.
(On the right are shown Rubber Spools.)

The trucks usually operate in 6 ft. aisles with a 36" x 36" pallet. The outriggers fit in between the wing type pallets for close stacking. The trucks have rated load capacity of about a ton. They can turn in their own length. The load carrying frame is supported on all four corners by sheets. Interchangeable battery unit is attached to the load frame by a hinge. A new truck model is shown in Fig. 38.

The selection of the proper pallet is an important single step in the development of a fork truck pallet system. The standard four-way pallet for use with fork lift is generally used and is shown in Fig. 39.

Fig. 38. Telescopic High-
Lift Pallet Truck.



For use with Fork Lift and Pallet Hand Lift Truck. Forks of the lift truck can enter any one side of pallet.

Fig. 39. Standard Four-Way Pallet.

VIII. COLD STORAGE PLANT LAYOUT

For computing refrigeration loads of cold rooms, a tentative building layout was given. Sufficient floor space was allotted for:

(1) Three cold storage rooms of equal size (70 ft. x 43 ft. inside) and similarly situated.

(2) Common anti-room in the center, about 16 ft. x 14 ft. inside dimensions.

(3) Ice and machinery room, size 43 ft. x 41.5 ft.

(4) Sorting and grading room, size 43 ft. x 27. ft.

(5) Empty crate storage, size 41.5 ft. x 19 ft.

(6) Main office and toilet, size 41.5ft. x 19 ft.

(7) Loading platform for potatoes, size 56 ft. x 20 ft.

(8) Loading platform for ice, size 25 ft. x 10 ft.

Machinery Layout

Before preparing a machinery layout, overall dimensions or floor space occupied by each machine or equipment must be known. The approximate dimensions as gathered from manufacturers' catalogs for suitable sizes of machinery are given as follows:

(1) Compressor and Motor base, each 7 ft. x 4 ft.

(2) Evaporative condenser, 10 ft. x 4 ft.

(3) Ammonia receiver, 6 ft. x 15 in. dia.

(4) Multipass brine cooler, 6 ft. x 14 in. dia. Outside dimensions with insulation, 8 ft. x 2 ft.

(5) Ice tank with submerged coils or cooler, 30 ft. x 15 ft. Outside dimensions including insulation, 32 ft. x 17 ft.

(6) Blower type floor mounted room coolers or cold diffuser, each taking 5 ft. x 2-1/4 ft.

(7) Sorting and grading unit, 20 ft. x 6 ft.

Two geared rotary pumps placed side by side, brine circulator motor drive on ice tank and air blower for ice cans are also required to be installed but they are unimportant from the layout point of view, as they do not occupy much space.

Having known the approximate overall dimensions, a machinery layout can be planned by template method, keeping in mind that for servicing and accessibility of inspection

(a) At least 2 ft. distance is left between any part of machine and walls.

(b) At least 2 ft. distance is left between any two machines.

Further, evaporative condenser should be nearest to the outside wall with short inlet and outlet ducts. Compressors and the control board should be near the operator's room. Refrigerator doors should open outward.

Piping Layout

Length of interconnecting ammonia and brine cold lines should be reduced to a minimum. Brine pipes leading to and coming from the cold rooms should pass through the rooms as far as possible, so that little refrigeration is lost to the outside. Location of water lines and drains should also be properly arranged.

Plant Layout

Taking into consideration all of the essential points, a typical plant layout drawing is shown in the accompanying sheet (Fig. 42, folded and placed in leather pocket inside of the back cover).

IX. PLANT OPERATION AND MANAGEMENT

Many cold storage plants are not utilized to the best advantage, either because of short-sightedness in management or failure to operate at maximum efficiency.

Compressors and other equipment need to be in good shape, condensers must be clean, evaporators should give desired cooling effect on brine and room unit coolers properly defrosted, etc. Good operation and management include such handling of the stored product as will utilize the plant to the best advantage and such control over the operation of the plant and care of the equipment as will keep both at top operating efficiency.

A. Plant Operation

The value of cold storage plant in maintaining the value of seed potatoes is largely determined by the way it is operated. Even the best designed plant with automatic control needs more or less continuous attention to insure the best results. Theoretically, all of the machinery may be installed and operated in proper manner and hence, no difficulties in actual operation should ensue. But this is not always true in practice and engineers are required to analyze the trouble and apply remedies.

In the operation of the mechanical equipment for the refrigerating and ice making plant the operating engineer should remember that it is his primary duty to get a dertain desired result and produce the result with maximum efficiency and minimum amount of expense.

Performance

For a given compressor with certain mechanical efficiency due to moving parts and compression efficiency due to cylinder loss under given load conditions, B.H.P. required to drive the machine depends on its theoretical performance or theoretical indicated H.P. The latter again depends on the weight of gas handled per unit time and the compression ratio of the machine. B.H.P. value in ratings of the compressors will also indicate that, for a constant discharge pressure, the B.H.P. per ton of refrigeration is greatest at the lowest suction temperature. This is true because the compression ratio is greatest at this condition. If the suction pressure rises, the total required B.H.P. is increased because a greater weight of gas is compressed; the B.H.P. per ton of refrigeration, however, drops because the work done in handling the greater weight of refrigerant is offset by the lesser lift from suction to condensing pressure. As the suction pressure is increased further, a point is finally reached at which the total B.H.P. is at a maximum, from which point it decreases on rising suction pressure, for a given condenser pressure. At this point the benefit derived as a result of the decreasing compression ratio overcomes the effect of the additional work done by handling a greater weight of gas.

Influence of head pressure on the compressor performance is still great. The B.H.P. increases rapidly with the discharge pressure and refrigeration capacity decreases slightly as the discharge pressure increased. The net result is that B.H.P./ton of refrigeration increases still more rapidly with increased pressure.

Both of the above statements can best be exemplified by a compressor performance chart prepared from compressor ratings; or expressed mathematically, H.P. required per ton of refrigeration, being inversely proportional to the coefficient of performance, will be

$$= \text{a constant} \times \frac{h_4 - h_3}{h_3 - h_2}$$

where h_4 is the enthalpy of discharge refrigerant gas

h_3 " " " " saturated gas entering compressor (assuming no superheating)

h_2 is the enthalpy of liquid at expansion valve (assuming no sub-cooling).

From the above expression it is evident that B.H.P. per ton of refrigeration decreases or efficiency of the compression system improves with either h_3 increasing or

h_2 decreasing. Now, h_3, which is enthalpy of saturated ammonia gas at suction pressure, will increase with suction pressure and h_2, which is enthalpy of saturated liquid at head pressure decreases with the discharge pressure. Hence, for greater thermodynamic efficiency, suction pressure of the system should be as high as possible and discharge pressure as low as possible, under given conditions and set up.

It is the duty of the operating engineer to constantly bear this important point in mind and do whatever is possible in the circumstances to improve the pressures and hence, prevailing saturation temperatures. Suction temperature though greatly limited by the temperature of the medium to be cooled can be brought up to some extent by improving the "U" value of the evaporator and the room coolers by keeping the coils clean, having good brine or air circulation and keeping them defrosted, as the case may be. Head pressure is also limited to some extent by temperature of cooling medium in the condenser which is related to wet bulb temperature of outside air in case of evaporative condenser. But here too, condenser tubes can be kept clean, air velocity and spray water adjusted to give better "U" value for condenser coils, consistent with water and power economy.

Operation

Proper operational sequence of ammonia plant is:

(1) Cleaning the system, which has been idle for an extended period, so as to loosen all scale and dirt. Scale traps examined and cleaned;

(2) Applying the various tests--low pressure air test, high pressure test, leak detection by soap film, repair of leaks, final air test;

(3) Vacuum testing and evacuating the system of air and moisture;

(4) Charging the system slowly in usual manner;

(5) Detecting leaks after charging by litmus paper in the condenser water, brine tank or cooler; and by sulphur stick for exposed piping and joints.

(6) Starting the system and checking and adjusting pressures.

Maintainance

Proper maintainance of the system includes:

(1) Keeping proper amount of ammonia refrigerant in system;

(2) Purging system of air and other noncondensible gases;

(3) Removing oil from system;

(4) Cleaning of condenser and other heat transfer equipment;

(5) Keeping compressor valves, stuffing box and other moving parts of compressors in good operating condition and repair;

(6) To follow right procedure in case of leak in high side, compressors or in low side and in case of other troubles.

The operator is supposed to know the right procedure to follow for the above maintainance jobs. Other points for maintainance, which have been discussed to a lesser or greater degree in previous pages, include, preventing freezing of brine cooler, removing ammonia impurities, stopping brine corrosion and removal of noncondensible gases.

Care of Cold Rooms

Proper and efficient management of cold storage rooms consists principally of:

(1) Preventing wide temperature fluctuations by regulating the flow of brine, proper air circulation and stacking goods properly;

(2) Keeping coils clean and properly defrosted;

(3) Preventing heat leakage into the rooms by keeping the doors shut tight and by reducing traffic in and out of the rooms;

(4) Keeping and checking good temperature record;

(5) Keeping the rooms clean and sanitary.

Care of Ice Making

This consists of:

(1) Maintaining proper amount (level) of brine of proper concentration;

(2) Maintaining proper temperature for slow freezing of ice;

(3) Frequent examining of ice cans for leaks, etc.

Lubrication Oil

Selection of proper lubricating oil in relation to the refrigerant used is an important factor in the efficient operation of the whole system.

Anhydrous ammonia has little effect on a well refined lubricating oil. Hence, it does not lower the viscosity of the oil. In the presence of moisture, ammonium hydroxide formed will emulsify with oil. For this reason the compressor oil should be dehydrated and water be kept out of the system to prevent freezing of the evaporator coils and formation of oil emulsions. Absorption of ammonia in oil (by volume of gas per unit volume of oil) is shown in the following table, which is taken out of ASRE Data Book, Vol. I:

Pressure lbs./sq.in.,abs.	Ammonia Temperature 32°	72°	150°
14.7	3.14	2.2	1.24
29.3	6.34	6.44	2.39
44.0	10.05	6.60	3.56
58.7	15.66	8.53	4.69
146.7	--	--	12.43

Compressor oils refined from naphthene-base crudes have naturally low pour point and have distinct advantage over paraffin-base oils. It is essential that a refrigerating compressor oil has a pour point lower than the lowest temperature encountered at any point in the system. Further, oil should have sufficiently high viscosity to maintain an effective lubricating film on the bearings. However, it should not be too heavy, as besides causing drag, loss of power and increased cost of operation, it will not drain sufficiently from evaporator coils or brine cooler. It should be refined thoroughly to prevent sludge forming reactions with the refrigerants. Further, oil should be chemically stable, not react with ammonia, do not carbonize at high temperatures.

The small vertical enclosed type ammonia compressors should be lubricated with an oil having a Saybolt viscosity ranging from 60 to 70 at 100°F., a pour point of-40° to-50°F., and a flash point of 300° to 325°F. (29).

Lubrication System

In the latest vertical ammonia compressors, oil is pumped from the crank case and circulated under pressure and returned through filters to the crank case; a mechanical lubricator forces engine oil to cylinder wall. This reliable positive lubrication system is shown completely in Fig. 30.

The oil cycle from the reservoir through the mechanical gear type pump, into the main bearings and up to the wrist pins provides a supply of oil to the principal moving parts and, in addition, supplies the stuffing box seal just forward of the center bearing. An additional lubricator (not seen in the figure) belt driven from the crankshaft, furnishes oil to the cylinders at a controlled rate. The built in gear pump supplying the bearings provides a positive pressure of about 40 lbs. a bove crankcase pressure. The geared pump supplying the cylinder walls feeds oil at a rate of about two drops per feeding period of 15 seconds. The compressors are fitted with oil filter in the steel oil pressure line and an oil screen in the crankcase.

The outboard bearing is lubricated independently by means of an oiling ring.

Machinery Records

Maintainance operations will be facilitated and general operation of the equipment would be improved if better records are kept on each piece of equipment. As a minimum, a card should be kept for each compressor, showing latter's serial number and size, the belt number and size, the refrigerant used and probably brief specifications of oil. Space should be provided on the card to make notation of the date, and brief details of service or maintainance operations done on the machine. With this information on hand, all maintainance and repair work is placed on a better and more accurate basis and better operation of the equipment results.

Operational Records

Operational records such as power and water costs, the room temperatures, loaded capacity of the rooms or amount of ice harvested should be kept at regular intervals. This will not only have the bookkeeping in up-to-date order but will provide a ready guide on how the whole plant is functioning. The method of keeping such records will vary with local conditions.

B. Economic Considerations

Many factors come into the picture in determining the true cost of storing the product or manufacturing the ice. The actual cost will depend on geographical location of the plant in relation to the markets, size of the equipment, cost of power, water, labor, machinery and supplies, design and efficiency of the refrigeration equipment, management and administration, etc. Some of these factors have already been considered in the introduction.

It would appear that the unit cost per ton of refrigeration or per ton of ice could be used as a basis of comparison for plants of different sizes located at different places and under different operating conditions. This unit of comparison is being used to form an overall picture of the functioning of the plant. But when one observes that so many factors enter into the final picture, it would be better to make an intelligent comparison between plants by observing the relative magnitude of the different factors involved.

Total Cost

Most common factors which combine to determine the overall cost of storing product or manufacturing ice are:

(1) Fixed charges or overheads - DIRT Law., i.e. Depreciation, Interest, Repairs and Taxes (including insurance and incidentals).

(2) Operating Charges - Power, Water, Ammonia, Oil, Supplies and Labor.

(3) Management - Procurement or Purchase, delivery or sales, office expenses, salaries, miscellaneous.

The practical engineer is primarily interested in the manufacturing or storing cost which comprise items (1) and (2), and is not particularly concerned with the expenses of the office, sales and delivery, which come under item (3). But in a small plant like this, it is generally found that one man acts both as an engineer and manager. Hence, he has to think of all the items.

Storing or Manufacturing Cost

The fixed charges or overhead are almost constant throughout the year irrespective of the relative loads on the plant. But operating engineer can very well reduce two items of these fixed cost--depreciation and repairs.

Depreciation is inversely proportional to life of the equipment and the engineer's job it is to keep the equipment and manager's job to keep the buildings insulation in good condition and shape so that their life is longer. Repairs can be considerably reduced by following a systematic method of repairing and check up and operating the plant with due care.

The fixed charges make about 10 to 15 per cent of initial investment on the complete plant per year or per season, but due to load factor for 7 months' working period being 7/12 or about 60 percent, the fixed cost per unit of production or job will be more than what it would be if plant works throughtout the year.

Operating Expenses

Overall operating costs are subject to many variables as shown before. Real control of costs is best obtained by breaking them down into various units and considering each of them on the basis of plant design, load factor and the other conditions existing in the plant. Here, operating engineer can offer considerable saving by economic and efficient running of the plant, and by saving in power and water bills, ammonia and oil consumption. He can operate the plant under favorable thermodynamical conditions, already explained, and may in addition, bring about an economic balance between power costs and water costs.

Labor forms considerable part of operating expenses. Labor saving devices like material handling equipment for potatoes, lifting equipment for ice and automatic running of the system have been provided. If we include both trained and unskilled workmen to form labor, it may consist of:

One Manager-Engineer for administration and supervision
one Refrigerating mechanic for operation and maintainance.
Four Sorters and graders
Two Product loaders
One Ice man for ice pulling and crane operation
One cleaner

The Engineer can give material suggestions in labor saving devices and as manager, he can think of ways to employ the labor fully and efficiently by a time schedule.

For instance, the product loaders may be directed to so arrange their schedule that while one is loading the truck by the side of grader, the other is unloading the truck in the store room.

Management and Costs

The cold storage plant may function as a proprietory concern purchasing the potato stock on outright cash basis or hiring the cold storage space. Some potato houses can be run on a co-operative system by groups of growers or merchants. Methods of procuring the stock for storage or outright purchase will vary with the local environment. Similarly, method of selling or delivering the hired stock will be different with different concerns. In all cases, it is the duty of the manager to arrange all of the transactions in the best possible manner for mutual benefit.

The cost of "assembling" of potatoes to the warehouse comprises all charges incurred in the movement of the produce from the growers' field to the storage premises. It includes handling and transportation charges, commission agents' commission and brokerage when paid, cost of crates and other incidental charges. When outright purchases are made for storage direct on the farm, these factors in the procurement cost are important, and must be brought to a minimum. The best and most economical method of transport of the farm produce must be found. Bullock carts are convenient for short distance transport of 20 to 40 miles. For 100 to 200 miles, motor lorries ply not only between farms and warehouse, but from latter place to railroad stations.

Other administrative expenses like office expenditures, salaries, advertising are almost fixed. It is the manager's job to see, however, how they could be kept at low value without loss of office and administrative efficiency.

Overall Costs

Having found the cost of procuring or purchasing the product and the storing of it, the concern is in fairly definite position to evaluate the total cost after certain period of storage. They could judge how profitable it would be to sell the product at a certain market rate or how much hiring charges could be taken to run the storage house on a good profitable basis, with reasonable profit to product owners. Again, from the growers or merchants point of view, who feel inclined to store the product, it is necessary that ultimate selling price should equal the selling price at the harvest time plus a premium to cover losses in storage, cost of hiring the storage space plus some additional profits which they may expect due to cold storage quality of their product.

Prices of Seed Potatoes

There is always a seasonal rise in prices of potatoes in India. Quoting the prewar (1940) figures for comparison: during February and March, table stock sell at Rs.1/8/- to Rs.2/8/- per mound or 35 to 60¢ per bushel; but from July to November, prices go up and are nearly double of what they are in the previous period.

The rise in the price of seed potatoes is even higher: at the time of harvest their prices range nearly the same as table stock; but after six months, prices go up in most of the markets to Rs.5/ to Rs.14/- per mound or $1.20 to $3.10 per bushel, due to the greater demand for seed at that time.

Under present methods of storage, more than 50 per cent of the potatoes are lost and the seed potatoes left for planting are greatly deteriorated in quality; and hence, in spite of such high prices, there could be not much profits. But if seed stock is graded and stored under good condition in refrigerated storages, it is clear that very good profits can accrue to the growers or merchants, as the cost of storing should come much less than difference in the price.

Storage Charges

As already explained, the total cost of operation of the cold storage plant consists of the sum of various unit costs. Cost of land, building, initial investment on machinery and other equipment and operating expenses on power, water, oil, etc. vary from time to time and from place to place. Hence, no attempt is made here to evaluate the total cost of operating the plant, and to work out cost of storing the product per unit weight or for manufacturing a ton of ice. But it is evident that the cold storage charges should be so maintained that it is profitable for the owner of the plant to maintain the warehouse as well as to the customers to make use of the storage facilities to the greatest advantage.

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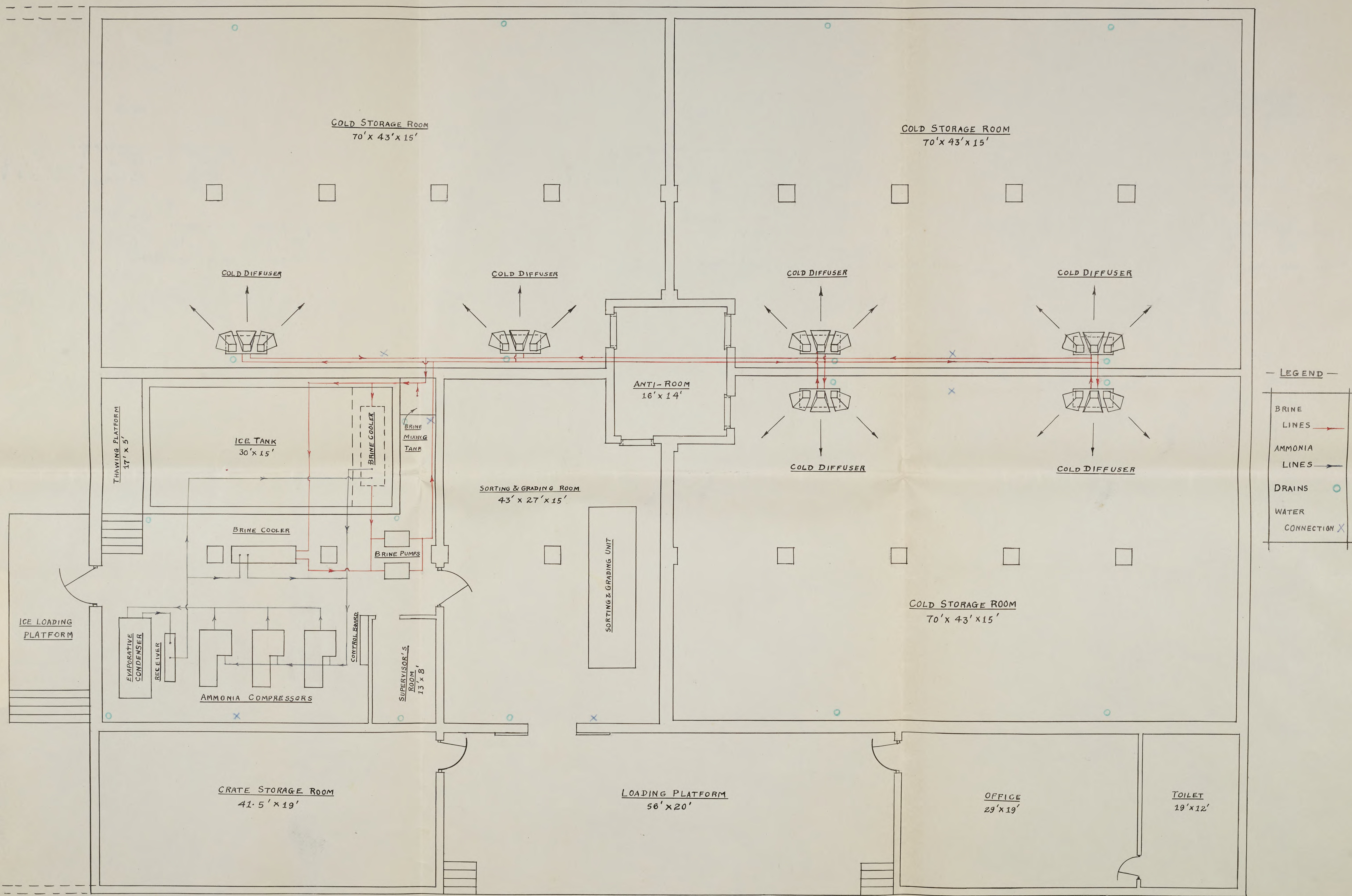

FIG. 40. COLD STORAGE & ICE PLANT LAYOUT SCALE - 1 IN. = 8 FT.

CAPACITY - 1000 TONS POTATOES & 10 TONS DAILY ICE