ONE HUNDRED YEARS OF AMMONIA REFRIGERATION

By: M. W. Garland
Senior Consultant, Frick Company

HISTORY

Information presented originates from publications written prior to 1920, conversations with consultants, engineers and operators, with experience dating back to 1890, and observations from 1920 to the present time including patent studies and publication information.

The author has been associated with the Frick Company since July of 1920 in capacities of Trainee, Superintendent of Field Installations, Assistant Chief Engineer, Chief Engineer and Vice President of Engineering, retiring in 1967 but continuing as a consultant. Some of the following, therefore, will relate to Frick Company operations and application developments.

The Frick Company was established in 1853 by George Frick who in 1850 made his first steam engine for his own use and in 1853 built the first Frick engine of his own design, for sale to others. In 1861 the family and plant were moved from a nearby location into Waynesboro, Pennsylvania, producing engines from 2 to 150 horsepower.

The first refrigeration compressor design was made in 1882 and the first compressor for commercial use shipped in 1883. The engineer employed for the refrigeration design work was Edgar Penny and all designs were for ammonia as the refrigerant.

There exist numerous stories about the addition of refrigeration to a successful steam engine business. The following seems logical because of the advertising by George Frick—a part of which read: "I am prepared to do repairing of all kinds, in a workman like manner, and on reasonable terms."

Many of the repair items were rebuilding of De La Vergne ammonia compressor cylinders; such cylinders being damaged by excess oil used in the so-called "wet compression" system. The explanation of this system was given in an early publication issued by the De La Vergne Company, a reproduction and explanation of which is given in Figure 1.

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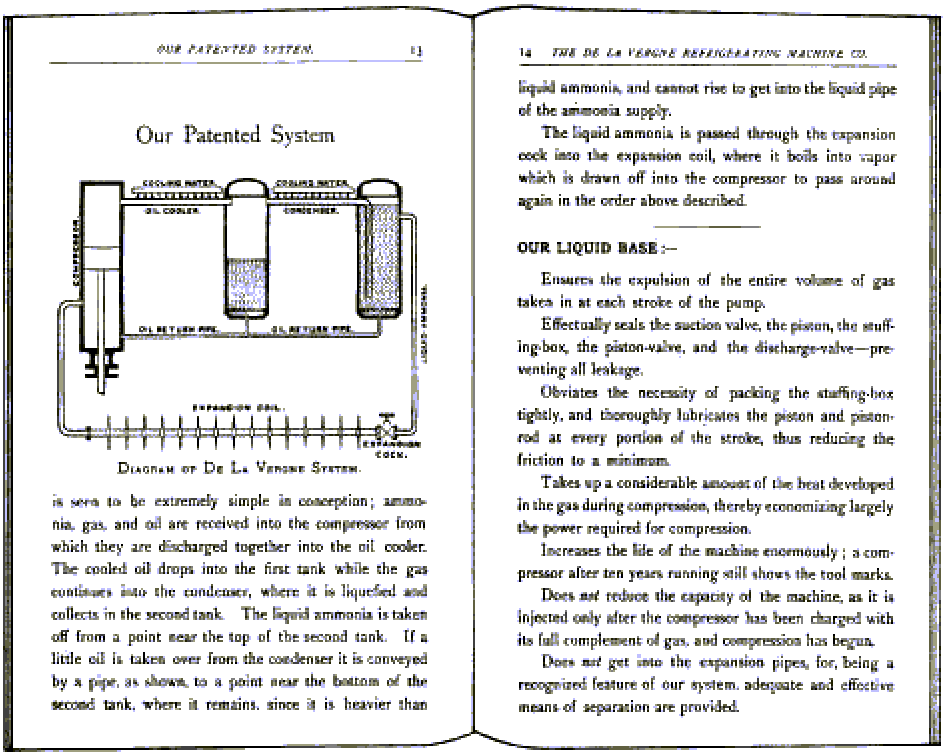
OUR PATENTED SYSTEM. 13

Our Patented System

DIAGRAM OF DE LA VERGNE SYSTEM.

is seen to be extremely simple in conception; ammonia, gas, and oil are received into the compressor from which they are discharged together into the oil cooler. The cooled oil drops into the first tank while the gas continues into the condenser, where it is liquefied and collects in the second tank. The liquid ammonia is taken off from a point near the top of the second tank. If a little oil is taken over from the condenser it is conveyed by a pipe, as shown, to a point near the bottom of the second tank, where it remains, since it is heavier than

14 THE DE LA VERGNE REFRIGERATING MACHINE CO.

liquid ammonia, and cannot rise to get into the liquid pipe of the ammonia supply.

The liquid ammonia is passed through the expansion cock into the expansion coil, where it boils into vapor which is drawn off into the compressor to pass around again in the order above described.

OUR LIQUID BASE:—

Ensures the expulsion of the entire volume of gas taken in at each stroke of the pump.

Effectually seals the suction valve, the piston, the stuffing-box, the piston-valve, and the discharge-valve—preventing all leakage.

Obviates the necessity of packing the stuffing-box tightly, and thoroughly lubricates the piston and piston-rod at every portion of the stroke, thus reducing the friction to a minimum.

Takes up a considerable amount of the heat developed in the gas during compression, thereby economizing largely the power required for compression.

Increases the life of the machine enormously; a compressor after ten years running still shows the tool marks.

Does *not* reduce the capacity of the machine, as it is injected only after the compressor has been charged with its full complement of gas, and compression has begun.

Does *not* get into the expansion pipes, for, being a recognized feature of our system, adequate and effective means of separation are provided.

Figure 1

The extent of such compressor repairs as made by George Frick would seem to be the logical reasoning for adding refrigeration compressors and systems to an already prosperous steam engine building business.

The De La Vergne vertical compressor was double acting; it used spring loaded inlet and discharge valves and relied upon oil injecting to offset losses created by excessive clearance areas. (Figure 2)

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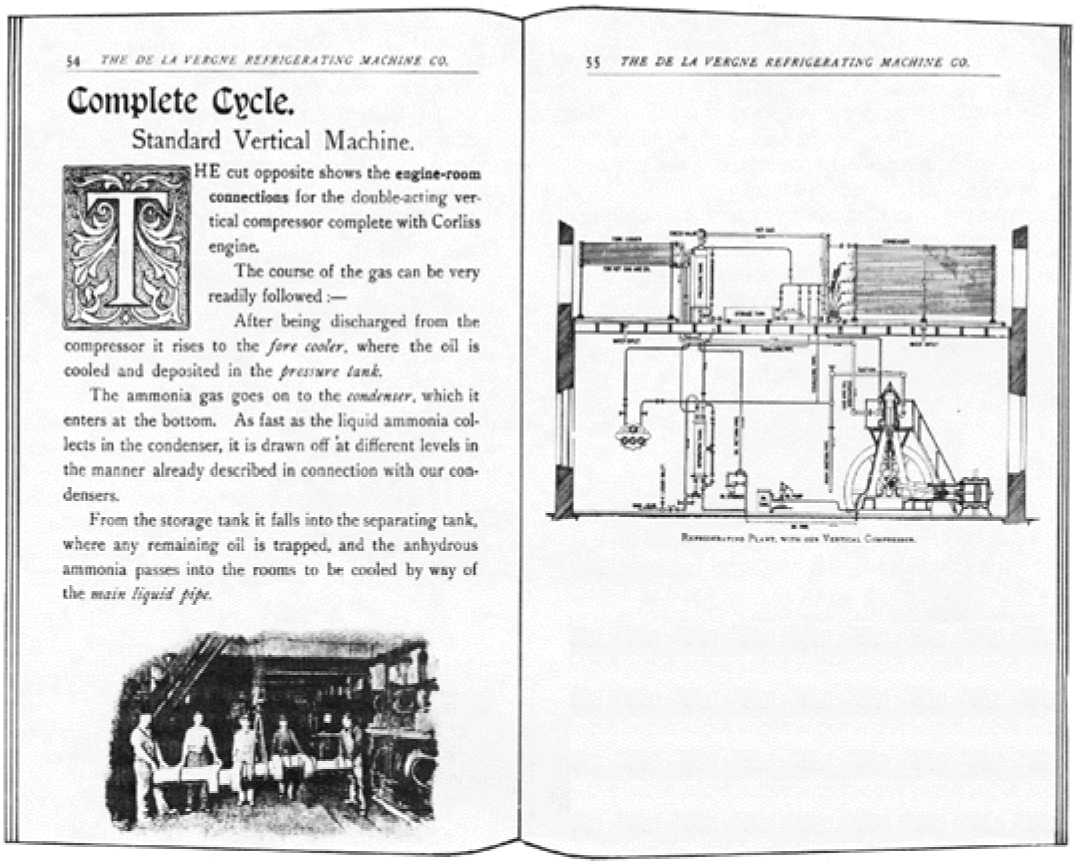

54 THE DE LA VERGNE REFRIGERATING MACHINE CO.

Complete Cycle.

Standard Vertical Machine.

THE cut opposite shows the **engine-room connections** for the double-acting vertical compressor complete with Corliss engine.

The course of the gas can be very readily followed :—

After being discharged from the compressor it rises to the *fore cooler*, where the oil is cooled and deposited in the *pressure tank*.

The ammonia gas goes on to the *condenser*, which it enters at the bottom. As fast as the liquid ammonia collects in the condenser, it is drawn off at different levels in the manner already described in connection with our condensers.

From the storage tank it falls into the separating tank, where any remaining oil is trapped, and the anhydrous ammonia passes into the rooms to be cooled by way of the *main liquid pipe*.

55 THE DE LA VERGNE REFRIGERATING MACHINE CO.

Refrigerating Plant, with our Vertical Compressor.

Figure 2

System shown included Atmospheric Bleeder type condenser.
Figures 1 & 2 from MECHANICAL REFRIGERATION AND ICE MAKING.
By The De La Vergne Refrigerating Machine Co., New York, 1898

In the first Frick design, oil was limited to that amount necessary for lubrication purposes. Excessive clearance volume was eliminated by the use of a single compression at the top end of the vertical cylinder and with the discharge valve in the head and inlet valve in the piston. With both presenting a flat surface flush with the cylinder head and the piston head. Clearance volume was limited by the distance between the two heads. Opening of the inlet valve was assisted by a spring and by inertia; closing of the inlet valve was also assisted by inertia. Spring loading was provided for retention of discharge head assembly in place on the top of the cylinder wall for safety purposes, in the event of an accidental liquid refrigerant return to the compressor. The inlet vapor entered the bottom cylinder area below the water-jacketed area. Teachings at that time described the water jacketing as being used to reduce heat conductivity from the hot discharge area into the incoming vapor area. (Figure 3)

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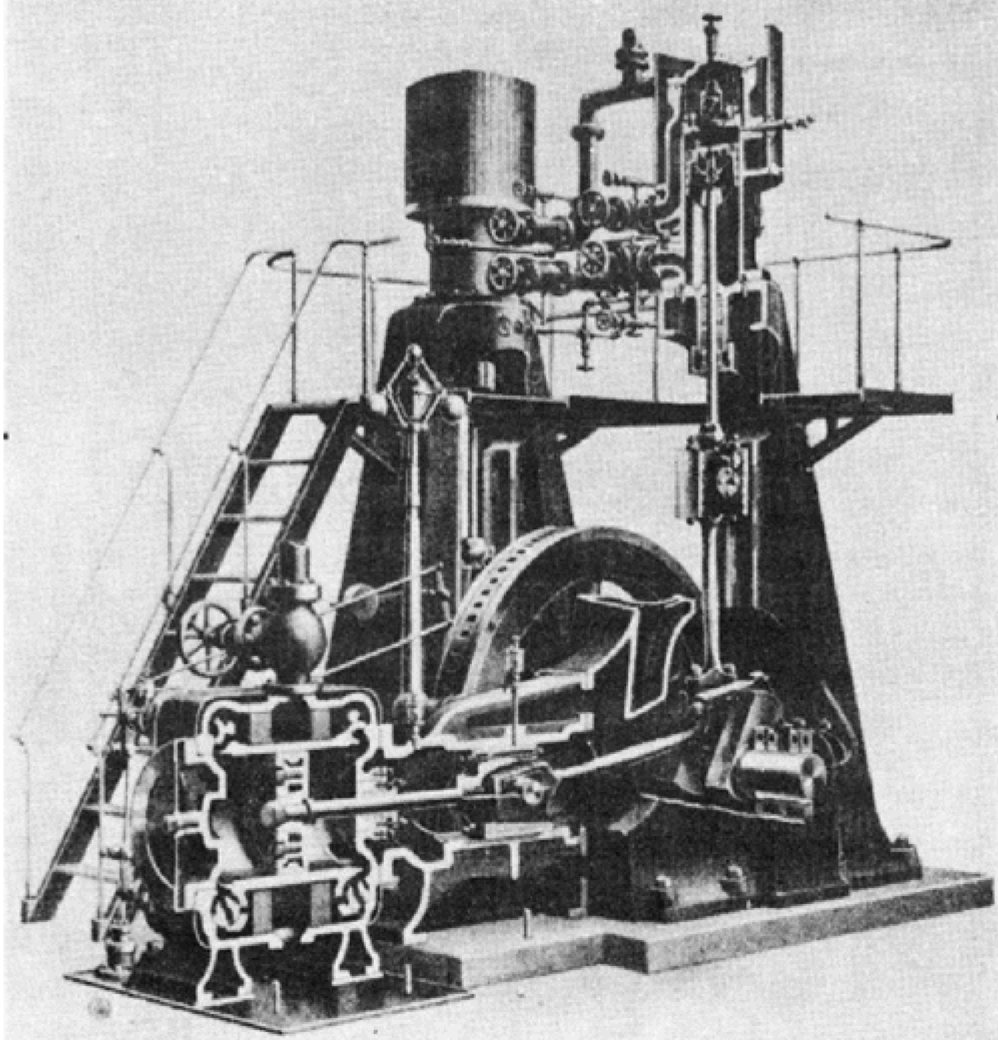

Illustration D82—Full Shaded Perspective Sectional View of Vertical Single-Acting Refrigerating Machine with Simple Corliss Engine

Figure 3

Open "A" Frame Type

All of the above features were incorporated in that first single acting, inlet valve-in-piston design which produced a most efficient reciprocating compressor. The concerns of minimum clearance volume, least

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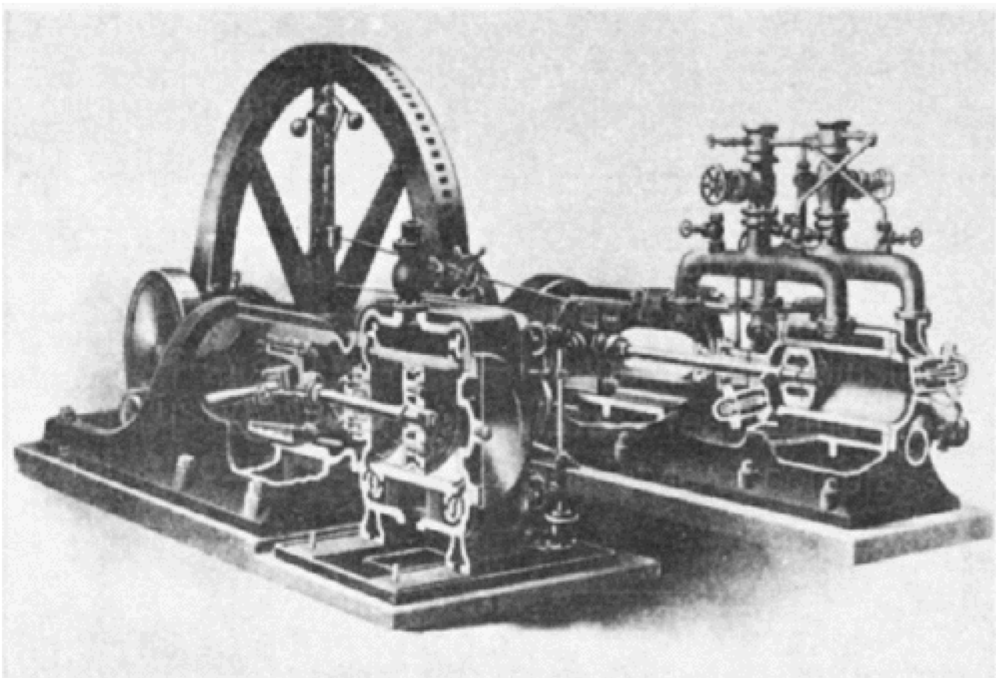
resistance to inlet valve opening, and least heat conductivity from discharge to suction are today the design aims in compressor construction.

Having participated in the performance tests of many of those early compressors makes it possible for the author to state that the indicator cards taken from those tests show a performance never exceeded by any later design. The later designs provided for higher RPM ability and lighter parts, but their structure introduced a greater percentage clearance volume and other losses.

Horizontal compressors with double acting cylinders were competitive from the viewpoint of ground-level accessibility; however, because of excessive clearance volume they never matched the volumetric efficiency of the vertical single acting compressor. (Figure 4)



Full Shaded Perspective Sectional View of Horizontal Double-Acting Refrigerating Machine

Figure 4

Until about 1900, the vertical single acting and horizontal double acting compressors were predominant in all ammonia refrigeration applications, all of these compressors being of the cross head piston rod type. (Figure 5)

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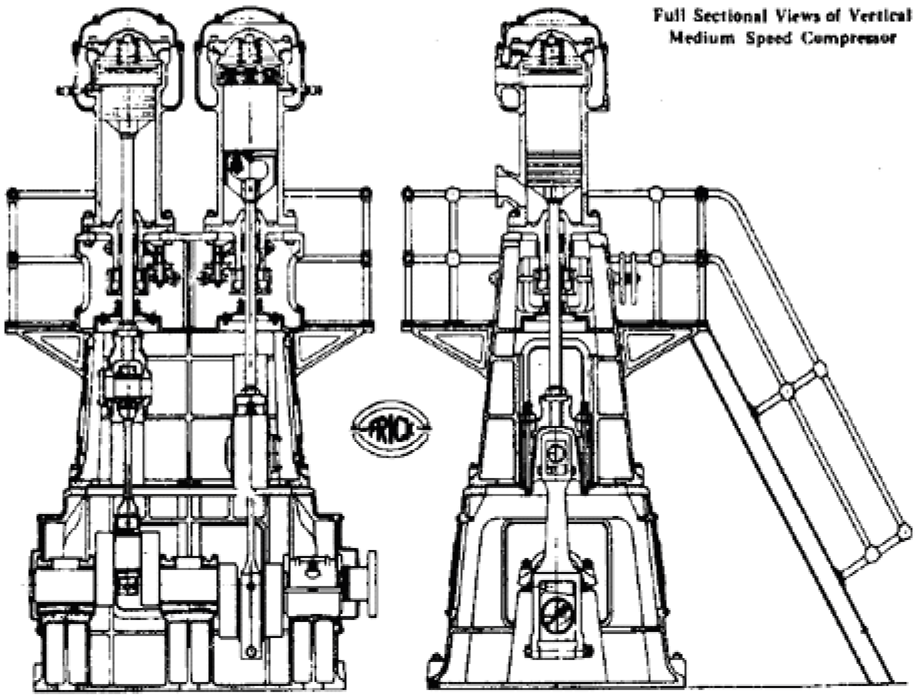

Figure 5

Full Sectional Views of Vertical Medium Speed Compressor, often referred to as a Semi-Enclosed Compressor.

Proof of the efficiency of the single-acting structure, with inlet valve in the piston, was demonstrated at the beginning of the 20th century. That structure was adopted by all of the major compressor manufacturers into the enclosed single-acting compressor with the piston serving the dual function of accepting the connecting rod thrust and compression of the ammonia vapors. The result was a tremendous saving in structural weight and cost of manufacture per ton of refrigeration. This enclosed type structure eliminated individual piston rods and required refrigerant sealing at the crankshaft projection from the crankcase only. (Figure 6)

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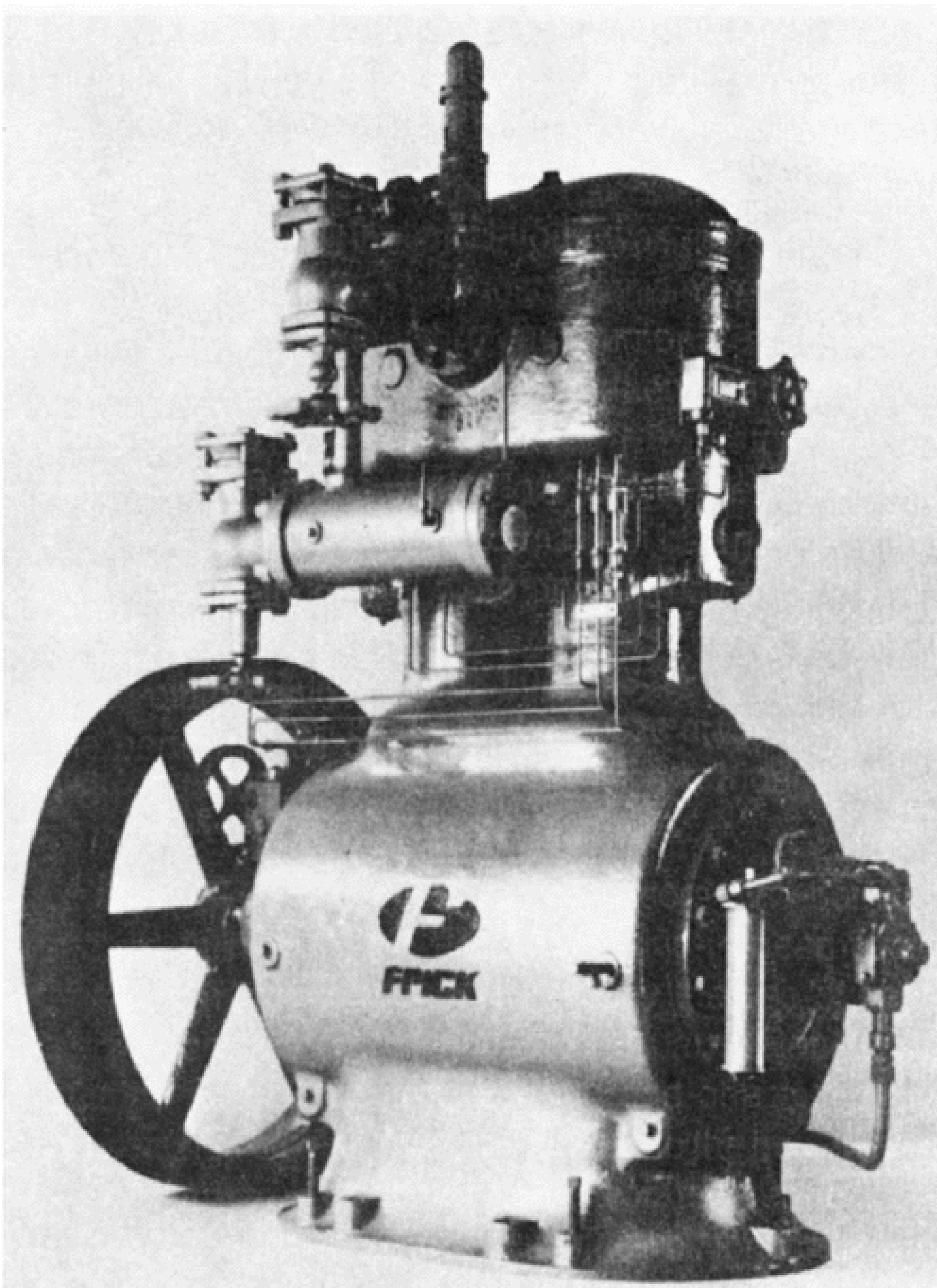

Figure 6

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Refrigeration in practically all applications, ice making, breweries, cold storages and freezers were, until the late 1920's, manually operated with remarkable efficiency.

For example, as a trainee in 1921, it was the author's nighttime chore to manually operate the hand expansion valves on two one-hundred-ton can-ice-producing tanks during the acceptance tests. Then, and now, the prevailing question: What is the optimum superheat at the inlet to ammonia refrigerant compressor? The engineer in charge at that time instructed the writer to adjust the expansion valves so that the frost line at the entrance to the compressor did not recede or progress beyond a certain marking at the inlet strainer. To do so would reduce the ice-making performance.

Additionally, the ice pullers were on a strict time schedule, making each ice pull at a specified time. Installations were made with a guaranteed ice production per ton of coal or specified number of KW per ton of ice and this author never saw a single failure.

The demand for automatic operation was in full swing in 1920. A major obstacle to automatic operation was the problem of the crank shaft stuffing box. The type of packing material available required a tightening of the gland bolts during shutdown and a release of the bolt tension during operation, to prevent leakage and overheating.

Literature as early as 1880 indicated that system compounding for low temperature work was most efficient. Observation by the writer in field installations and tests in the early 1920's showed that operators resisted the use of refrigeration suction pressure below atmospheric pressure because of stuffing box leakage and air contamination in the system. This resulted in a determined research effort on the part of the company to develop a truly tight shaft seal. This was accomplished and put on the compressors and into the market for replacement of packing in the late 1920's under the patent name of "Flexo Seal," the first successful mechanical seal for large reciprocating refrigeration compressors.

With the shaft seal problem corrected, the next step was the development of a large volume compressor for low pressure service in the first stage of compound compression systems for low temperature service. Note that despite the benefits of compounding being known in the 1880's, wide spread use did not occur until the early 1930's, and then only because of the development of oil pressurized shaft seals, which corrected overheating and air entrance problems.

Recognition of faults, such as compressor clearance volume, valve leakage, etc., were all known then as they are now. Oil injection to reduce leakage and provide internal cooling was recognized long before the advent of the oil injected blade rotary and helical screw compressors, and with the same problem of providing full oil separation from the liquid refrigerant flow to the evaporator.

At the time the author began, ammonia as a refrigerant was never considered a hazard to the operator or other occupants because instructions to all newcomers were:

1. To immediately leave any area where there was a leak, and to then reapproach that area only from the upwind side for repair purposes.

2. To immediately close the service valve in the liquid supply and permit compressor operation to zero pounds gauge.

3. Regardless of an apparent safe pressure, never to open a line if frost continued to exist on any part of the isolated area.

Lubricants for ammonia compressors remain about the same. A paraffin base oil without additives was the exclusive lubricant in the U.S. until 1926. Naphthenic base oils were then introduced, and with catastrophic results. The fault was an excess of resins which caused a varnish build up at piston rings, and eventual seizure. It was the expensive warranty cost which caused compressor manufacturers to buy and sell a proper lubricant. The resin problem of the Naphthenic base oils was later corrected and does not knowingly exist at this time.

So-called "Ice Machine Oil" was even then a problem item because a mineral oil of limited additives has an atmospheric pour point no lower than -40°F. However, it did not prevent the development of low temperature systems. Test chambers at -75°F air temperature and -90°F ammonia operated successfully with an oil rated at 0°F atmospheric pour point being used in the compressors. There exists practically a zero miscibility of ammonia in a paraffin base oil. Thus, with a correctly designed oil removal means, ammonia will be free of contamination and provide the exact liquid temperature as shown in published tables.

Oil separation in the liquid phase in the De La Vergne system was a method used until the early '20's and later abandoned for reasons unknown; but it remains one of the best methods, if an adequate retention time for oil separation is provided.

The writer has not observed any change in the differences of opinion regarding evaporator temperature and oil pour point. Some advocate an oil to give least trouble in the evaporator, others recommend an oil best suited to the compressor requirements. Experience over the years has proved that the lubricant should be selected for best compressor life and any oil problem that might occur in the system can be eliminated by proper oil separation and proper means for oil removal.

The rapid growth of the refrigeration industry, using ammonia almost exclusively up to 1930, caused some application problems which under a slower growth rate might never have occurred. One of the problems encountered then and at the present time is that of so-called "condenser bottling". Condensers in parallel occasioned by plant growth, were assumed to be at the same pressure-temperature equilibrium; a condition that does not exist because of differences in size, cleanliness and location. Equilibrium is obtainable only in the liquid accumulation phase which occurs in the drain lines to the receiver and then with a free vapor flow to the area of lowest vapor pressure of any of the condensers. It is a challenge to present day ammonia system designers to recognize the need for a more careful study of condensate collection and its retention in a multiple-condenser, single-receiver application.

Dual effect compression was introduced in the "19 teens." An inlet port, through the cylinder wall at the bottom of the stroke, permitted high pressure ammonia vapor to enter the cylinder which was previously filled with vapor that had entered through the inlet valves in the piston at the nominal plant suction pressure. Frick Company made many of these dual effect compressors, in addition to cylinders for application to the vertical single acting cross head type compressors. Most notable of these were the large compressors at the Commonwealth Fish Pier in Boston.

It was the advent of the mechanical shaft seal in 1930, and refinements in compound system design which greatly added to the growth of the food freezing and storage business. Controls with adequate reliability were being made available and practically full automatic operation was possible. Operator duties were now confined to observations of oil levels, motor lubrication, condenser cleanliness and system tightness.

Flooded evaporator refrigerant control was made a reality in the late 1920's by the introduction of reliable float devices and a scientific means for determining float level head.

In the can ice making field there were many improvements: precooling water by using the refrigeration effect available with the dual acting compressor; development of the low pressure air agitation system; the full row lift; and later, the one man operable plants producing up to 30 tons of ice per day.

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In the time period from 1923 to 1940, the writer served as superintendent of Frick field installations. Changes in all areas of refrigeration were most notable in system design and control. The enclosed type valve-in-piston compressor was gradually refined for higher RPM operation, and higher RPM compressors were introduced which featured multiple smaller cylinders with inlet valves in the head area.

In this same time period production of Dry Ice was increased while cost was reduced by the introduction of cascade systems. Liquifying of $C0_2$ was accomplished by an ammonia system with an evaporator temperature of -20°F, condensing the $C0_2$ at -10°F, thus enabling the use of standard ammonia compressors.

During this same period of development of high speed, valve-in-head ammonia compressors, halocarbon refrigerants were also introduced. The accompanying fanfare momentarily reduced the rate of growth of ammonia applications, partly because of the widespread claims of ammonia danger by those advocating halocarbon refrigerants. Those in the ammonia refrigerant business also assisted in the use of halocarbon refrigerants for air conditioning and in areas of dense human occupancy.

In the cold storage field a lower product insurance rate was applied if a halocarbon refrigerant was used instead of ammonia. The result was that many new installations were a direct halocarbon refrigerant or an ammonia-to-brine system. This continued until the middle of the 1950's when cold storage warehouse companies that had switched to the direct halocarbon refrigerant in the late 40's returned to ammonia in new storage applications.

The development of household refrigerators and freezers started a decline in block ice production in the mid 30's. This was offset by a greater use of refrigeration in the petrochemical field.

The advent of World War II started many low temperature applications in the -75°F area; for engine testing, stratosphere environment testing, etc.

Of interest in that area was the startup of a gun firing test room. A military tank of the best U.S. design in 1940 chilled to -15°F for 8 hours was inoperative in all respects: the breech block could not be opened; the engine could not be started; and on removal by external tow means, the track would not move. A similar observation of test results was for aircraft tires: chilled to -75°F, on removal from the test chamber the tires broke into pieces when subjected to a drop of four feet. In the realm of engine testing, it was observed that below -35°F, low stage cooling coils of plain pipe did not build up excessive frost. Moisture fell out in the air stream as snow, and snow traps were necessary to keep from plugging the carburetors of reciprocating engines.

It required a -90°F ammonia system to produce a -75°F room temperature, and that required an approximate 1.7 PSIA at the compressor inlet. Only the valve-in-piston compressor with the oil pump located below the oil level in the crankcase could reliably provide such performance. Those plants were built for three-stage operations.

Numerous three-stage systems using R-12 and designed for -75°F service were built, using the higher speed valve-in-head type compressors. These did not provide the trouble-free performance available from ammonia installations.

The wartime period brought about many new refrigeration control requirements. The Rubber Reserve Corp. plants that produced artificial rubber had a very difficult control problem in their latex reactors. Various refrigeration manufacturers were asked to submit designs and test units. Frick Co. entered this competition and provided the required cleanable, removable coil, with a control system to prevent the previous problem of overheating that would ruin entire batches of latex. All major artificial rubber producing plants under the war production program were fitted with that unique evaporator design and refrigerant feed control.

At the same time, a butadiene purging and recovery system was re-engineered and placed under an automatically controlled refrigeration system. Ammonia was the refrigerant.

Block ice plants of 8 and 15 ton per day capacity were built and packaged in portable sections for use by the Army. A demonstration for the War Department proved a field assembly time of 10 to 15 days. Ammonia was the refrigerant.

One of the war efforts was the concentration of citrus fruit juices. The most promising procedure was to pass orange juice through an ammonia system condenser for heating of the juice, and then evaporate water from the juice under a vacuum. The water vapors were then condensed by the low side of the ammonia system and removed in liquid form. Heat balances using ammonia as the refrigerant were supplied by Frick Co.

Penicillin production was an early war priority and was accomplished with the freeze-dry process, using a three-stage ammonia system at a refrigerant temperature of -75°F.

Reciprocating compressors in both inlet valve-in-piston type and inlet valve-in-head type have been improved over the years by the use of better materials and lighter weight moving parts. Replaceable cylinder sleeves were introduced and one of the latest designs has a cylinder sleeve fixed into the housing well below the area of highest compression temperature and with a flat head assembly held in place by long bolts. This permits a relatively free expansion of the cylinder in the top area, greatly reducing the chance for piston seizure in the event of excessive heat caused by discharge valve breakage. (Figure 7)

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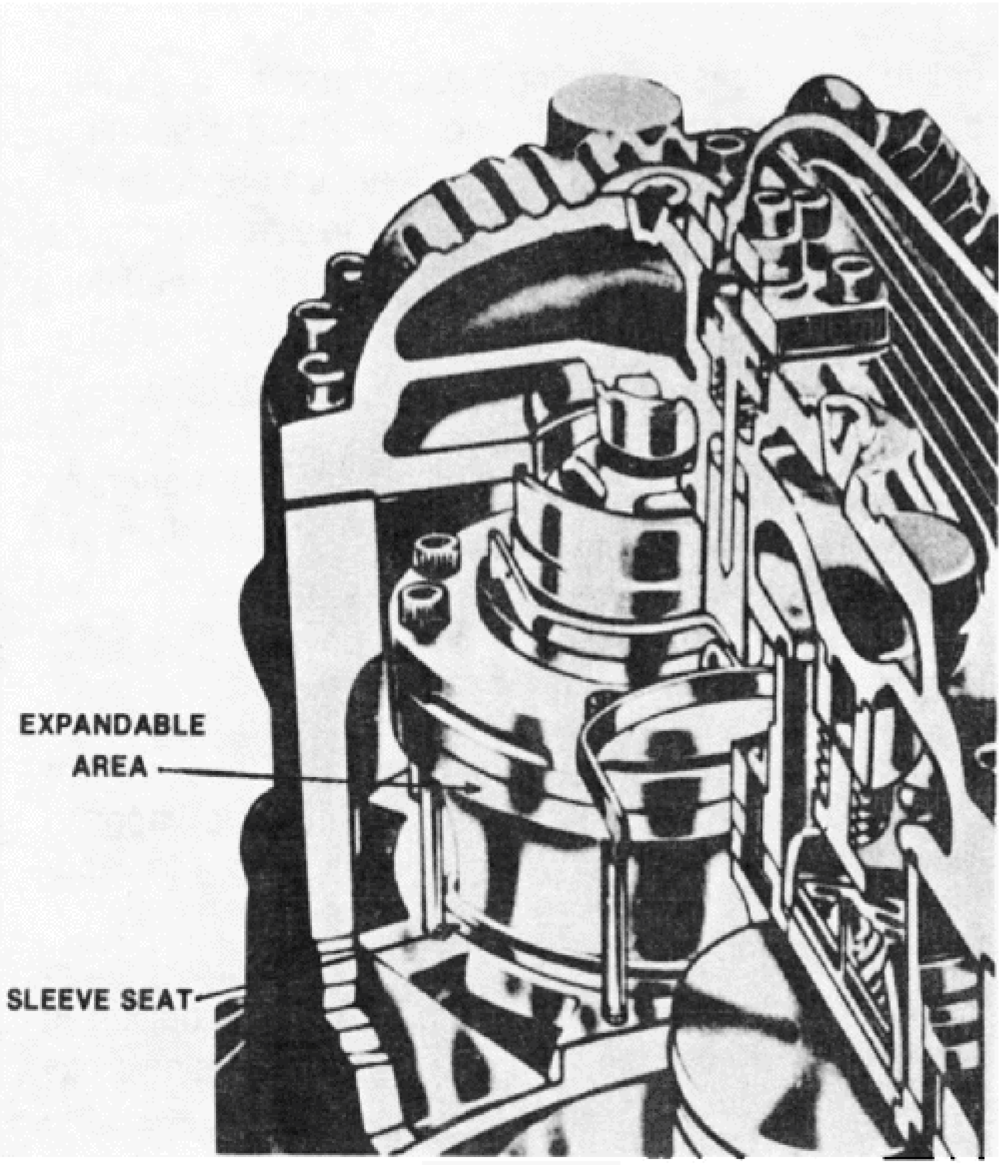

Figure 7

The Cylinder Sleeve is seated to the housing at about mid-section with long bolting to permit tip area freedom for expansion and contraction.

The rotary screw type compressor was under full scrutiny by the Navy during World War II in development work performed by the Elliott Co. for gas or liquid fuel turbine driven electrical generators. The Lysholm design of dry compression was used. U.S. production rights were offered to Frick Co. for refrigeration service, but a test unit of the dry design on a refrigerant calorimeter failed to show the required efficiency.

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As of this time, the efficiency of the rotary screw compressor is fully acceptable and is obtained by use of a large volume of lubricant which seals leakage areas and removes some of the heat of compression.

All rotary type compressors have had the penalty of a fixed compression or volume ratio. The advent of an automatically adjustable compression or volume ratio is now a reality and greatly improves the performance of this style of compressor. (Figure 8)

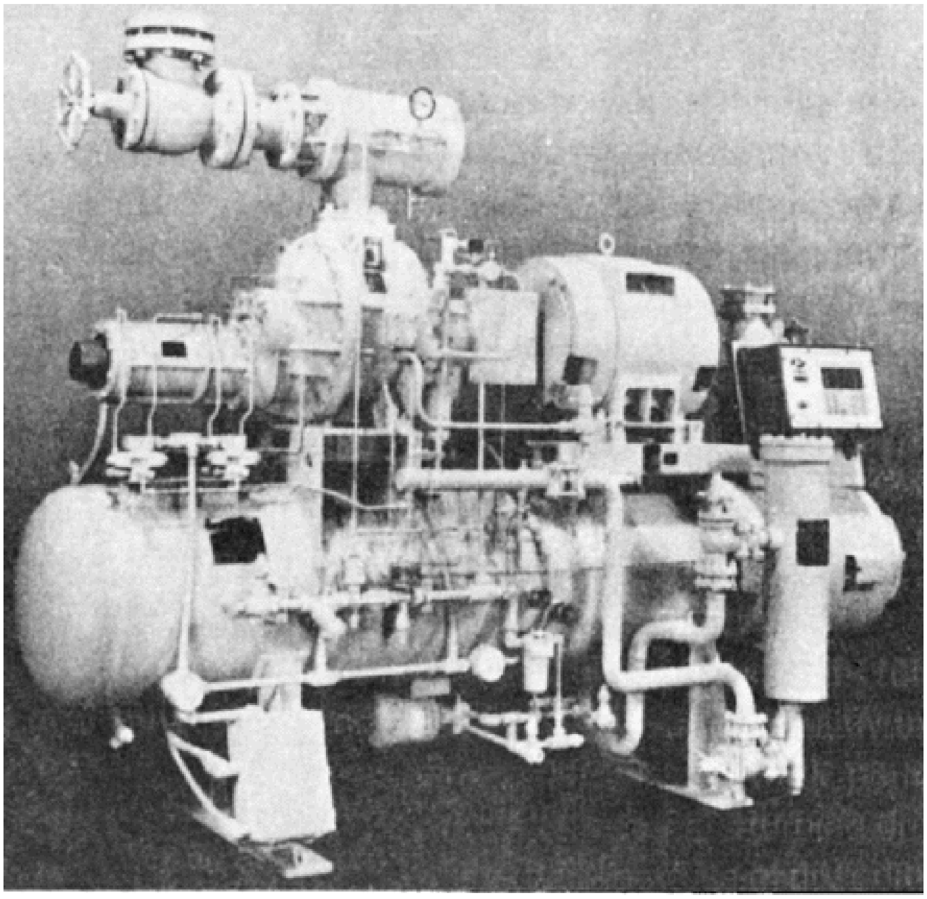

Figure 8

Screw Type Compressor with automatic volume ratio control. Available up to 1250 H.P. Larger sizes in prototype stage, all with Microprocessor control.

Condensers used in most plants prior to the early 1920's were the atmospheric type consisting of a vertically mounted coil, such coils being about 12 pipes high and 20 feet long, placed side by side and with top vapor inlet and bottom liquid outlet. Many variations in gas and liquid circulation were offered and there were names such as "Atmospheric Drip" type, and "Atmospheric Bleed" type. When assisting on tests in the field it was obvious that too many parallel circuits resulted in liquid bottling in as much as 50% of the coil. This problem was universally recognized in the industry, but never realistically corrected.

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Double pipe condensers were used for indoor service, usually in the smaller plants and with fewer coils in parallel than the larger exterior atmospheric applications.

The vertical shell-and-tube type condenser came into use in the 1920's and atmospheric coil condensers became obsolete. Many of the previous condensing problems were corrected, and the vertical open type shell-and-tube and horizontal closed type shell-and-tube condensers were universally applied.

The evaporative type condenser came into the market in the early 1930's. Early models used pull-through air flow with the fans above the coils. Later, designers went to push-through air flow. Structural changes have been in the direction of shorter coil length and more parallel circuits, to accommodate shorter fan assemblies.

Evaporators, up until the early 1920's, were made of plain iron pipe for air cooling; shell-and-tube and double-pipe type for liquid cooling. In block ice making, the parallel stands of pipe between cans were eliminated and a V.W. design compact coil arrangement took the place of the submerged brine cooler. This eliminated tube damage due to freeze-up, and permitted a greater brine movement with reduction in agitator horsepower.

The return to ammonia in the early '50's as the refrigerant for commercial refrigeration service occurred because of lower first cost, lower BHP of refrigeration, and less chance of unobserved refrigerant loss. An outstanding example was the rapid conversion in fishing boats from the halocarbon refrigerants to ammonia. The reason was that with the halocarbon refrigerant, in spite of extra labor cost to check for leaks, operators would find themselves with a boatload of fish and inadequate refrigeration due to loss of refrigerant charge.

The favorable BHP of refrigeration with ammonia is most easily observed by comparing the pounds of refrigerant flow rate per ton of refrigeration; the example here being at 0°F evaporation and 95°F condensing:

AMMONIA	.4325 lb/min/Ton Ref
R-12	4.2221 lb/min/Ton Ref
R-22	2.9958 lb/min/Ton Ref
R-502	4.7248 lb/min/Ton Ref

Because it is the compressor which provides the pressure difference necessary to move the refrigerant, moving less than 1/2 pound per minute of ammonia per ton of refrigeration requires the least BHP.

System pressure and flow conditions have been a subject for debate. For example, providing ammonia recirculation by a pump, in a system first observed and tested by the writer in 1926, contained practically the same features as put into today's installations. Controversial then and today is the recirculation rate. Why recirculate 3 to 4 times the evaporation rate? The only valid reason is that some evaporator designs require a large overfeed rate for the express purpose of obtaining distribution into the many parallel circuits. From a performance viewpoint the least possible overfeed will provide a wetting of the internal surfaces, the least pressure difference between the inlet and the outlet of the evaporator, and the lowest overall average surface temperature. Pumping excess and non-evaporated liquid causes excessive power input to drive the pump and generates a greater pressure difference between the inlet and outlet of the evaporator. The result is a higher overall average surface temperature for a given evaporator pressure than with a minimum of overfeed. Excessive overfeed results in a two phase flow in the suction line; the pressure drop in some observed operations resulted in the equivalent of a ten percent loss in compressor pumping capacity.

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Having served approximately 25 years in direct charge of the installation, testing and customer acceptance of more than 1,000 large ammonia refrigeration systems, and directing the engineering of such plants for 20 years, the writer feels safe in saying that practically all differences between the buyer and the seller resulted from one or more of the following:

A. Specifications of writing

B. Lack of component balancing

C. Deficiency in piping provided by others

A. As an example, for a very large water chilling system, a simple specification was written: "Chill—thousand gallons of water from 44°F to 40°F; compressors, condensers, water chiller and controls by the bidder. Installation, water piping and pumps by others". A resulting performance problem was caused by the use of chilled water to suddenly stop a hot reaction. The jacketed vessels had been in a no-flow condition and full of water during the reaction at a temperature of about 100°F. Then suddenly the chilled water flow was started and the heated water content of the jackets was returned to the chiller. This suddenly applied hot water load caused a serious liquid refrigerant surge out of the chiller.

NOTE:

The specification indicated a continuous water flow at specification temperatures. Had the variable water flow rate been specified, the refrigerant handling means would have been designed to accommodate it.

B. Observed over the years is the fact that many users of refrigeration in chemical, pharmaceutical, and similar specialized fields, choose to build their own evaporators; investigation of performance problems always showed an ample heat exchange surface, but inadequate vapor release and separation area.

Another observation leads to the conclusion that many users, specification writers and designers do not have a complete and adequate view of the liquid-to-vapor volumes in an evaporator at clean surface full load conditions, when related to the applied refrigerant control.

A so-called float device senses the head of liquid and not the height of the boiling level. Thus a heat load greater than design may stop the liquid feed but cannot prevent boil-over.

C. Observe that one pound of liquid ammonia at 20°F evaporation temperature becomes 5.91 cubic feet of vapor. One pound of liquid ammonia at -40°F evaporation temperature becomes 24.86 cubic feet of vapor. Thus an evaporator of good design for +20°F ammonia will be a poor design for -40°F ammonia, for the same Btu per degree temperature difference. At the low temperature condition, the suction line piping may be undersized, causing an excessive line pressure drop.

Pressurized refrigerant delivery is rapidly developing and the dual drum type of alternative liquid delivery to the evaporators may be the most efficient of all such overfeed systems, providing the overfeed is restricted to not over two times the evaporation rate. Efficiency claims because of a high overfeed rate have been, in this writer's opinion, a half truth because an improvement of an internal film factor does not change the transfer rate of a frosted heat exchange surface.

Ammonia as a refrigerant serves very well as a heat transfer medium. In an ammonia storage plant, heating of the ammonia from -28°F to +40°F is a costly process when natural gas or fuel oil is used to

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supply the heat. However, water from a solar pond could provide the necessary heat if some means were provided to prevent water from freezing, which would occur in a direct water-to-refrigerant heat exchanger. The problem of freezing water has been eliminated in a specific ammonia heat transfer system. This heat transfer system is an evaporator in a horizontal shell-and-tube form and an overhead condenser in a horizontal shell-and-tube form; a thermal gravity system with a fixed ammonia liquid charge. Warm water from a solar pond provides the evaporation heat and cold ammonia from storage provides the condensing effect and is thereby heated. The liquid return to the evaporator is by a gravity head sufficient in height to provide the necessary flow rate of ammonia in the heat transfer system. The first unit of this kind heats 600 GPM of ammonia from -28°F to +40°F.

Ammonia is a proven low cost refrigerant both in first cost and power requirement. Energy saving points to recovery of the condenser heat.

Low power cost is directly related to a low condensing temperature.

Low condensing temperature means that the Btu available at the low condensing temperature may not serve the need for a higher temperature water than that available from the condensing system. This may tempt some users to switch to another refrigerant because higher condensing temperatures may be available and at a reasonable condensing pressure. However, that will result in an unwarranted increase in horsepower per ton of refrigeration.

The correct answer is to stay with ammonia as the refrigerant for the main plant and then in a cascade application use another suitable refrigerant for a boost to a final high temperature of the water.

Such a cascade system should be used only part time; that is, during those hours when the hot water is required.

CONCLUSION

The author's sixty-two years of experience with more than fifteen different refrigerants has shown that ammonia is an ideal refrigerant; however, in system application, much needs to be done to improve operating efficiency. For evaporators, improved refrigerant distribution is needed so that the loss due to pressure difference is eliminated—such losses are extensive in systems with an overfeed rate of more than two times the evaporation rate.

Condensing and condensate collection must be taught more scientifically than at present. This is one area in which there is no consensus of scientific reasoning by application engineers.

We need greater emphasis on details, such as a recognition that a two psi pressure drop in an ammonia suction line at 0°F evaporation results in a 7% increase in vapor volume at the entrance to the compressor. The use of a globe type valve in a suction line is the equivalent of a straight pipe run of 100 feet. An angle valve is equivalent of 28 feet.

Refrigeration is a growing field in application and will continue to expand tremendously in the future. In large systems, ammonia will be a dominant refrigerant.