Energy-Saving Technology for Skating Rinks

The results of an experiment with a new energy-saving technology tested on five skating rinks

By CLAUDE DUMAS,
Project Engineer,
The Professional Building Services Module, City of Montreal,
Montreal, Quebec, Canada

Montreal owns 25 indoor skating rinks distributed throughout the city. Construction dates vary from 1958 to 1982. In the last few years, we have experienced recurring steel corrosion problems with the brine header pipes. Pipe corrosion has led us to replace six header systems between 1989 and 1995.

This corrosion provided a unique opportunity to modify the design and cut energy consumption. In brief, here is how we proceeded. First, we updated the brine specification. Then, we took two-pass brine distribution systems with evaporators connected in parallel and replaced them by building four-pass brine distribution systems with evaporators in series.

The original two-pass design

The original slab system used 296 1-in. diameter pipes placed every 3.5 in., as shown in Fig. 1. Most of the recent installations have used polyethylene pipe in the slab in contrast to the steel pipe found in the older installations. Slab pipes were connected in two-pass circuits. The brine header system was reversed-return, using 8-in. Schedule 40 steel pipes. Brine chillers used the direct expansion technique with three independent HCFC-22 circuits within one evaporator shell connected to air-cooled remote condensers, suction accumulators, refrigerant receivers, and head pressure controls. Refrigeration controls were electromechanical, and the solenoid cut the suction upon compressor shutdown. The chiller evaporators were piped in parallel, and each evaporator used 750 gpm.

Steel pipe headers at Arena Mont-Royal.

Each skating rink had six open-type reciprocating compressors producing 12 tons of refrigeration effect each and driven by six open 25-hp electric motors. The brine was composed of calcium chloride with a specific gravity of 1.25 at 60 F circulated by a single brine pump. The pump, driven by a 50-hp, 1800-rpm motor, had a capacity of 1500 gpm and a head of 43 psig. One spare pump and a few compressors were kept in storage for installation on short notice in case of a failure at any of the rinks.

In four rinks, the three chillers were water-cooled with two independent HCFC-22 circuits within one evaporator shell. They released heat to a cooling tower equipped with one winterized reservoir and a water diverting valve. Three evaporators were piped for parallel flow, each evaporator using 500 gpm.

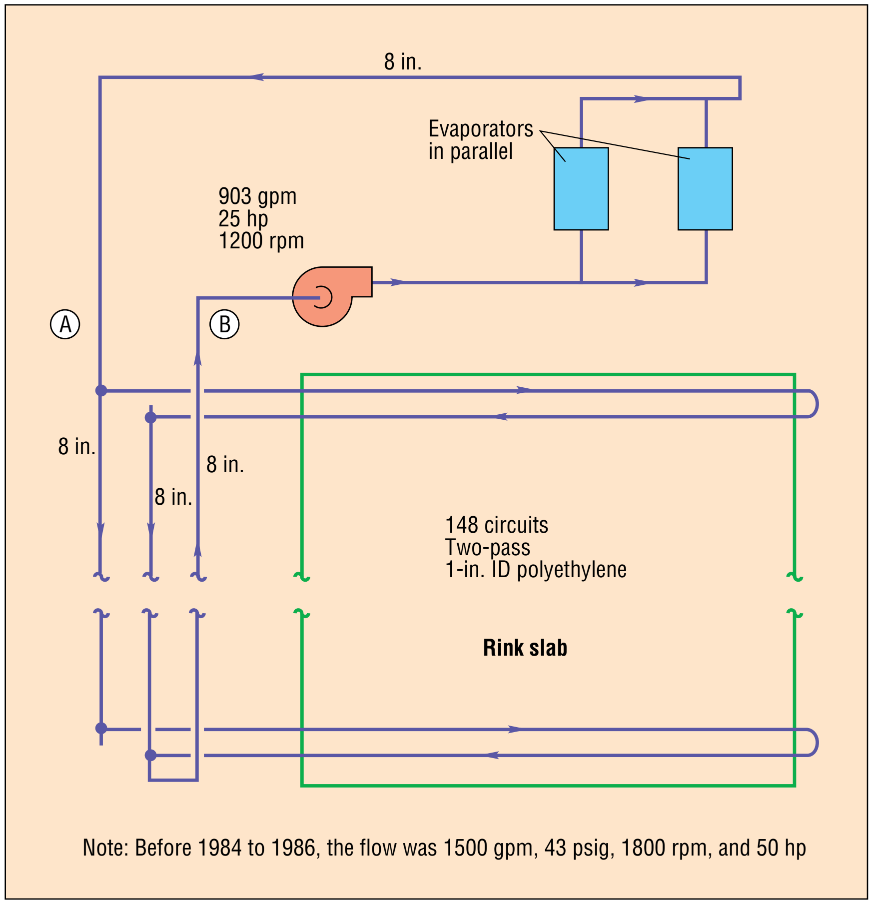
Modifications

From 1983 to 1986, we modified all our ice systems to save energy. On all our brine pumps, the original motors produced 50 hp and rotated at 1800 rpm. They were rewound to produce 25 hp and 1200 rpm. The brine flow produced by the original pump rotating at 1200 rpm was nominally 903 gpm. By dividing 903 gpm by 148 circuits, we obtained a ratio of 6.1 gpm per two-pass circuit (Fig. 1). We found that five rewound motors were excessively noisy due to magnetic vibrations and needed replacement by new 1200 rpm motors.

All compressor control systems were modified to stop and start by auxiliary low-pressure controls and pump-down. Timers were added to the systems to stop the refrigeration machines and the brine pump for seven hours every night to save energy. Timers were also added to control ventilation systems to save energy. Two of the six compressors in each arena were fitted with desuperheaters to heat domestic and resurfacing hot water. The pump energy consumption shrank 25 hp, resulting in less heat dissipation into the brine. Compressors and condensers did less work over shorter hours. These modifications produced very successful results.

At the end of 1991, budgetary restrictions on city energy consumption led us to search for other possible modifications. We decided to reduce the brine flow, decrease the specific gravity of the brine, and use a smaller, high-efficiency motor on the brine pump.

The brine specifications originated in 1971. They were revised for three reasons. First, reducing the specific gravity from 1.25 to 1.18 also reduced the pump energy consumption. In the past, brine freezing temperature was set at –20 F whereas now it is set at 0 F. This is sufficient, considering the low operating brine temperature of 15 F. Second, following a repair, we save money because less salt is required to top off the system. Third, environmental protection regulations prohibit the use of sodium chromate to inhibit corrosion. We now use a biodegradable inhibitor.



1 Schematic of two-pass system with evaporators in parallel (not to scale).

The new four-pass design

The pump horsepower is a direct function of head, flow, and specific gravity.[1] The heat-transfer performance of the pipes located under the ice is dependent on the heat exchanger area, the temperature differential, and the heat-transfer coefficient. Heat-transfer coefficients are experimental and are influenced by the nature of the fluid and the nature of the motion imposed to the fluid.[2] Actual brine flow is 6.1 gpm in 1-in. pipe, which translates into a velocity of 2.3 fps.

Our experience over many years gives us confidence in the above parameters. System pump flow can be reduced if sufficient fluid velocity is maintained. Reducing flow by half—from 903 to 451 gpm—requires the modification of the slab heat exchanger. These modifications make a four-pass circuit out of two two-pass circuits. In the original design, 1500 gpm was split in two evaporators piped in parallel for a flow of 750 gpm of brine each.

Ten years ago, we reduced the brine flow to 903 gpm by rewinding the pump motor to 1200 rpm. Therefore, the flow of each evaporator became 450 gpm. Within the

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[1]*Superscript numerals indicate references listed at end of article.*

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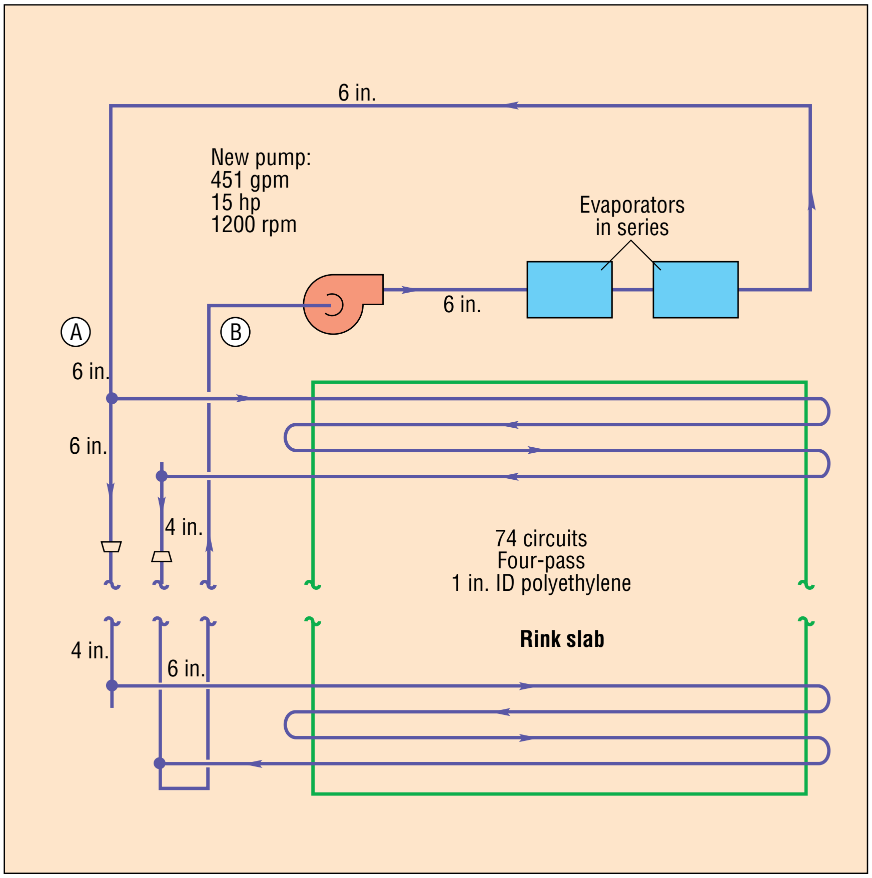
limitations of a two-pass system with evaporators in parallel, it is impossible to reduce evaporator brine flow further without jeopardizing the heat transfer on the shell side. To maintain the minimum flow and velocity within the shell side, one must pipe the evaporators in series.

In the modified slab system, the standard 296 1-in. diameter pipes are kept without change. Our modifications of the slab pipe arrangements create a four-pass circuit, as shown in Fig. 2, by adding a supplementary return bend to two two-pass circuits. The original 8-in. brine header system is replaced by a 6-in. header because brine flow decreased from 903 to 451 gpm. The header stays reversed return, using 6-in. Schedule 40 steel pipe throughout the system, with the end suction using 4-in. Schedule 80 pipe. All 148 nipples are Schedule 80 pipe. The use of Schedule 80 pipe is required for small diameter steel pipe because increased wall thickness results in added resistance to corrosion. The two (or three) evaporators in our systems are piped in series for a flow of 451 gpm each.



2 Schematic of four-pass system with evaporators in series (not to scale).

Each skating rink is equipped with six open-type reciprocating compressors; however, to save on the electricity bill, we allow a maximum of four compressors to run simultaneously—and this without endangering the ice quality. The brine is calcium chloride, adjusted for 1.18 specific gravity at 60 F. Brine is circulated by a single brine pump sized for 451 gpm and 64 ft of brine head, driven by a 15-hp high-efficiency motor rotating at 1200 rpm. The original pump is replaced even if it is still serviceable because the operation is inefficient; our tests show a penalty of 2 to 3 hp. On the brine pipe, the valves are butterfly with bodies of PVC and shafts of stainless steel.

The total cost of the modifications brought about between 1992 and 1994 was $77,000 per rink. (All sums of money are expressed in Canadian dollars and include taxes.) The energy savings resulting from a four-pass system with evaporators in series are estimated at $8500 per year. Ice quality is the same as with a two-pass system, and users do not notice any difference. By transforming the two-pass system into a four-pass system, we not only saved money but also generated a recurring saving of $8500 each year.

The electrical invoice is based on a combination of two parameters: the demand for kilowatts and the consumption of kilowatt hours. The reduction of both flow and specific gravity of the brine allowed us to replace the 25-hp motor with a smaller 15-hp high-efficiency motor. The smaller motor resulted in a saving of 10 hp (7.5 KW), thereby decreasing both the demand and the consumption of electricity. The pump horsepower (kilowatts) is decreased by 10 (7.5 KW), and consequently less heat dissipates into the brine and must be removed by the compressors using 2 hp per ton (0.42 KW per KW), thus again decreasing demand and consumption. The condenser fan motors run less time and need to dissipate less heat, which also leads to decreased consumption. An arrangement in series for the evaporators allows one chiller to operate more efficiently at a higher suction temperature than was possible in the parallel arrangement, thereby leading to less energy usage. Four compressors do the work of six, saving 2 × 25 hp = 50 hp (2 × 18.6 KW = 37.2 KW), once again lowering the power demand.

Design considerations

The lessons learned from a job where we replaced only part of the header have convinced us to extend the scope of work for the next header replacement jobs. All steel pipe up to the evaporator flanges must be replaced. When pressure testing the new pipes, we found unexpected leaks located on the steel pipe sections that were not part of the contract. Some sections of pipe were so corroded that welding was impossible. The pipe was easily punctured with just a screwdriver. Corrosion originating from the exterior wall of the pipes could be traced to small leaks of brine originating at a flange, coupling, vent, defective polyethylene pipe, or defective weld.

We no longer use Y-strainers in the piping of our brine systems; they are useless. We found that existing strainers were never serviced for the life of the systems and consequently removed them. Moreover, strainers are costly and cause additional consumption of energy. We now use a small line cartridge filter located across the pump to remove fine particles in suspension in the brine. The filter removes rust particles coated with chromate.

We no longer use flexible joints in the piping of our brine systems, resulting in a saving of four flanges and two flexes and, consequently, in the elimination of many potential leaks. Noise and vibration are not a problem with a pump running at 1200 rpm.

We have learned that in Quebec in 1992, approximately half of the brine header replacements, originally in steel, were done with PVC. The life expectancy of a steel header is 18 to 22 years. Exceptionally, we do have a steel system that is 36 years old. Although little is really known about the durability of the new headers, we believe that PVC will last longer than steel, and we will therefore use PVC piping to eliminate steel when replacing headers.

Delivery time for a 1200-rpm high-efficiency motor is 12 to 14 weeks. One pump/motor assembly was purchased as a spare and is made available to the successful bidder whenever a very short construction schedule is required. The contractor must replace our spare pump upon reception of the new one.

For competition, reliability of supply, and ease of maintenance, the pump must be vertical in-line and available from a minimum of two manufacturers. On the header system, the space available for the pump is standardized, the flange diameters are 6 in., and the spacing between flanges is 36 in. One brand of pump is 30 in. from flange to flange, and it requires a 6-in. long spool piece to fit the space available. The second brand of pump is 36 in. from flange to flange and does not require a spool piece. The brine pump can handle 451 gpm and 64 ft of brine head, driven by a 15-hp high-efficiency 1200-rpm motor.

Brine flow is 540 gpm and occasionally greater than in the design. We feel that it is premature to cut down the pump impeller to reduce flow immediately without further experimentation because we suspect that the fluid pressure drop in our evaporators varies considerably from system to system. Pressure gauges are not set up to verify drops in evaporator pressure, and often the installation is too corroded to install them. On renovated systems, we do not install gauges on evaporators but on the pump to limit the potential for leaks or small pipe breakage.

In the two-pass system, the sequencer used to control the refrigeration machines is electromechanical. This technology works well with the small brine temperature difference when combined with the slow rate of temperature change inherent in the large brine volume in the 8-in. pipe system. For the four-pass system, the situation is different as we must control a bigger temperature difference combined with a fast rate of temperature change due to a reduced brine volume in the 6-in. pipe system. We need only one or two compressors running to maintain the ice under minimal conditions when it is unoccupied. The

The large number of joints in the 1-in. polyethylene pipe are due to the demolition subcontractor's cutting them all short before reading the specification.

technology of the sequencer must be direct digital control (DDC) for stable operation. We use a personal computer, modem, and telephone lines to control the DDC systems in three arenas.

On a PVC pipe system, the weight of the pump must rest on its base, not on the piping system.

In the rinks where we use three evaporators, we pipe them in series, where one evaporator is valved off the brine circuit by leaving the bypass valve open. The brine pump head is underdesigned with the intent to circulate only two of the evaporators, have a foolproof system, and save energy. Above all, the underdesign is a strong incentive to follow the energy guideline, operate with only four compressors, and save on the power demand.

The pressure testing of brine headers, specifically the polyethylene pipes, must be done without excess and using caution. When testing, maximum pressure must be limited to a value not exceeding the pump head plus 10 percent. In one rink, when testing the new pipe for leaks, we found many unexpected leaks located in the slab's polyethylene pipes. Leaks were found at the two ends of the slab and showed up as cracks and pin holes. The defective polyethylene pipes, manufactured in 1973 and found to be fragile and brittle, developed pin holes and cracked for no obvious reason, and new leaks developed after each pressure test. Brine leaks were so numerous and difficult to repair that we thought we would have to close the rink and replace the slab.

Conclusion

We estimate 5.2 gpm per circuit as the minimum turbulent flow.[3] The minimum acceptable flow will be confirmed by further experimentation.

The experiment using the four-pass brine system with evaporators in series has culminated three years of tests on two rinks and one-and-a-half years on three rinks, all with very positive results. The results have been verified for regular rinks with a seating area for 300 people located on one side of the rink. By transforming the two-pass system into a four-pass system, we not only saved money but also generated a recurring saving of $8500 each year. System pump flow can be reduced if sufficient fluid velocity is maintained. Ice quality is the same as with a two-pass system, and users do not notice any difference.

Our successful results have led us to wonder what the ice quality would be if a four-pass brine system with evaporators in series was used on a rink suitable for the National Hockey League—in an arena with seating for 25,000 fans and lighting for color television. Moreover, we also asked ourselves if a six-pass system could be used reliably due to the greater difference in temperature produced by such a system.

We suggest that engineers responsible for constructing new arenas or replacing existing brine headers use the new energy-efficient technology known as a four-pass brine system with evaporators in series. The design is sound and has already been tested in five arenas. The energy saving made possible by this technology is 8 to 9 percent of the energy invoice—an important contribution to the conservation of a precious resource. We hope that the industry will copy our new technology as it is available to the public and is not patented. **HPAC**

References

1) *ASHRAE Pocket Handbook for Air Conditioning, Heating, Ventilating, and Refrigeration*, p. 23, American Society of Heating, Refrigerating and Air-Conditioning Engineers, 1987.

2) Kern, Donald Q., *Process Heat Transfer*, p. 3, McGraw-Hill, 1950.

3) Carlson, G. F., "Liquid Viscosity Effect on Pumping Systems," *Heating/Piping/Air Conditioning*, p. 81-82, July 1979.

Heat-transfer fluids

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tional water-based fluid systems, such as ethylene and propylene glycol, offer excellent performance with minimal regulatory impact. The wide use of glycol systems in standard HVAC environments allows these systems to be installed at competitive costs with familiarity in system design and installation. Their temperature range is, however, limited, and more specialized fluids are required for modern systems ranging to –60 F.

The chlorinated solvents, methylene chloride and trichloroethylene, though offering superior heat-transfer characteristics, are both classified as hazardous chemicals. Their heavily regulated use and hazardous classification leave them completely unsuitable for use in low-temperature systems. They should not be given even passing consideration for modern thermal transfer systems.

TABLE 4—Cost data for thermal transfer fluids.

Fluid	Cost per lb, $	Cost per gal, $
Methylene chloride	0.40	4.39
Trichloroethylene	0.60	7.27
Diethylbenzene	2.43	17.42
Dimethylpolysiloxane I	3.78	26.72
Dimethylpolysiloxane II	4.16	29.83
Heavy naphtha hydrotreated	3.04	19.50
Citrus terpene	0.95	6.75
20 percent propylene glycol	0.15	1.34
60 percent propylene glycol	0.46	4.01
20 percent ethylene glycol	0.14	1.30
60 percent ethylene glycol	0.41	3.90

Citrus terpene offers superior thermal properties to both diethylbenzene and dimethylpolysiloxane but cannot maintain those thermal characteristics over time due to its thermal and chemical instability. In addition, its highly corrosive nature places the mechanical system in which it is used at substantial risk. Furthermore, the product requires frequent replacement to maintain its thermal performance and therefore creates concern over its proper disposal. Finally, citrus

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