ICE RINK REFRIGERATION SYSTEM WITH CARBON DIOXIDE AS SECONDARY FLUID IN COPPER TUBES

P-O. NILSSON(a), J. ROGSTAM(a), S. SAVALHA(b), K. SHAHZAD(b)

(a) IUC Ref Centre, Kungsgatan 2A, Katrineholm, 641 30, Sweden
Fax: +46 (0)150 488 700, e-mail: jorgen.rogstam@iuc-education.se

(b) Energy Technology Department, Royal Institute of Technology (KTH), Brinellvägen 68, Stockholm, 10044, Sweden
Fax: +46 (0)8 20 41 61, e-mail: samer@energy.kth.se

ABSTRACT

This investigation presents the development of a refrigeration system for ice rinks based on carbon dioxide as phase changing secondary fluid in copper tubes with aim at optimising heat transfer, pressure drop and costs. A miniature ice rink was built and designed with copper tubes imbedded in a concrete slab. The tubes were developed for this purpose to have best possible conduction, protection and manufacturability at low cost.

Experimental tests were conducted with CO₂ circulated in the tubes of two different lengths and diameters by a hermetic pump. Correlations for heat transfer and pressure drop were compared with the experimental results to find the most suitable ones for this application.

Based on the found correlations a theoretical model was developed for numerical optimisation of the geometry. Good agreement was found between the model and the experimental results.

The numerical optimisation shows that the circulation rate has little influence on the overall temperature difference across the tubes when super imposing the influence of pressure drop and heat transfer. Consequently the circulation rate should be chosen as low as possible to ensure no superheat. An economical optimisation illustrates the influence of the tube pitch on the cost of operational and the investment. The higher the annual average heat load is, the tighter the pitch should be chosen. For moderate average loads, up to 100 W/m², the standard 100 mm pitch is found to be sufficient for 1/2” tubes.

1. INTRODUCTION

1.1 Background

Today’s ice rinks are mainly built with indirect refrigeration systems that pump calcium chloride (CaCl₂) solutions as brine, which causes extensive pumping energy. It has been proven that CO₂ can successfully be used as secondary fluid in ice rinks (Rogstam et al 2005). This applies to converted plants using steel tubes that are originally designed for NH₃. New plants made with new steel tubes, and to ice rinks that use small diameter copper tubes. The potential annual saving only from the use of CO₂ and copper tubes vs. pumped calcium chloride (CaCl₂) is 150 MWh.

It is preferable to design an ice rink with the headers on short end of the ice rink since it reduces the header length and the required soldering. Consequently, the tubes in the ice rink bed must be at least 120 meters long since the rink is 60 meters long. When it comes to the choice of tube diameter, the focus has been on standard tubes that can be found on the market and are easy to install; therefore, choice is limited to 3/8” (9.5mm) and 1/2” (12.7mm). Larger diameter could be used but this increases the internal volume which is negative from a system cost perspective. Earlier tests have indicated that the 1/2” tubes are the better choice than the 3/8” tubes due to pressure loss and heat transfer reasons. The tubes will have a thin protective plastic foil as protection against chemical and mechanical wear.
1.2 The objective

The objective of this investigation was to further optimise the concept of using copper tubes in an ice rink bed. The existing prototype, hereafter referred to as prototype 1, was to be thoroughly experimentally tested to verify which correlations can be used in this application. A model for numerical evaluation was developed which could be used to evaluate different layouts such as tube diameter, circuit lengths, pitches and circulation rate, etc. This would enable finding the best possible heat transfer and flow conditions for the lowest possible installation and running costs. A prototype 2 including the final geometry of the tubes and the ice rink floor was built to verify the results of the optimisation.

2. THEORETICAL ASSESSMENTS

The core issue is to keep a certain surface temperature of the ice at varying heat loads. Heat will be transferred from the ice and depending on the ice rink floor resistance a certain temperature of the refrigerant inside the tubes must be held. The circulation rate inside the tubes affects the convective heat transfer and hereby the inside surface temperature of the tube, which is also influenced by the pressure drop. By use of general correlations for pressure drop and heat transfer together with the present geometry of the ice rink a model can be built. The calculated values from the model will be compared with experimental measurements in the ice rink prototype which are described in chapter 3.

2.1 Heat Transfer Analysis

The heat transfer analysis can be divided into two parts i.e. conduction heat transfer, which is analysed with FEMLAB and convective heat transfer which is analysed with EES.

![Figure 1: FEMLAB model of 12.7 mm copper tube with plastic foil](image)

FEMLAB has powerful module to calculate heat transfer. The assumptions made in this model were:

a) Ground was assumed to be perfectly insulated.

b) The plastic around the copper tubes was assumed to be high density polyethylene.

c) The thickness of concrete above refrigerant tubes and thickness of ice was assumed to be same all across the rink bed.

d) The density of concrete as well as ice was assumed to be homogeneous.

For modelling, 12.7 mm copper tubes with and without plastic and 9.5 mm copper tubes with and without plastic were taken. The thickness of the concrete above tubes was taken 14.35 mm for each case. The baseline pitch was chosen to be 100 mm. The pitch is the centre-to-centre...
distance between two rink pipes or tubes. Average heat flux was increased from 50 W/m$^2$ to 300 W/m$^2$ with the increment of 50 W/m$^2$. The inner tube surface temperature was calculated by the model to obtain -4 °C at the ice surface. The pitch was decreased from 100 mm to 75 mm with the decrement of 5 mm.

As an example of the results of the simulation model, a 12.7 mm copper tube with plastic foil is shown in Figure 1. In this model, ground was taken perfectly insulated then a layer of 100 mm concrete was applied which was same in all other cases. Then there was 42 mm layer concrete and 12.7 mm copper tube with plastic foil was in the middle of this concrete layer. Thickness of the plastic foil was 0.45 mm. The top layer was a 25 mm ice sheet. The surface colour showing the temperature distribution and isothermal lines can also be seen in the figure.

3. EXPERIMENTAL EQUIPMENT

Since not enough theoretical correlations are available for this kind of system a small scale model of an ice rink was build together with copper tubes and CO$_2$ as the working fluid. The ice rink bed cross-section is made in full scale to enable direct comparison with the heat transfer model and the tubes are kept in full length for circulation rate and pressure drop evaluations.

3.1 Refrigeration unit

The required cooling capacity for this system is 15 kW since the design load of this ice rink is 300W/m$^2$ and the area is 55 m$^2$ (5.5x10 m). The refrigeration system consists of a tank to store CO$_2$ (see figure 2). This tank is cooled by a cascade condenser that is connected to the refrigeration supply system, which can control the brine temperature down to -11 °C. The brine is pumped into a cascade condenser and by means of self circulation – a siphon – where the CO$_2$ vapour condenses and returns to the tank liquefied. Below the tank there is a down comer leading to a CO$_2$ pump which provides the circuits in the ice rink floor with liquid CO$_2$. In the circuits the liquid CO$_2$ start boiling from the heat load and gas is formed. This gas will end up in the condenser and the loop is a closed.

The system has a pump “by-pass” option to enable evaluation of self circulation for the system. The volume flow of CO$_2$ to the ice rink could be controlled with an adjustment valve after the return line. This allows varying the circulation rate of CO$_2$, which is a parameter when optimising the heat transfer and pressure drop.

3.2 The ice rink

A miniature ice rink was used to further evaluate the copper tube concept. The tubes are installed in 5 circuits. Each was 120 m, which corresponds to the required circuit length in a full size ice hockey rink, and was built from two spools of 60 meter each. The spools are produced in these lengths only and can be handled by one man when it comes to weight. The circuits were attached to the concrete with plastic holders and then covered in concrete according to the standard construction method. The system was instrumented to evaluate the pressure drop and heat transfer in the tubes which is further described in chapter 4. The 120 m circuit runs twelve times in the 5.5x10 m rink which introduces more bends than a full scale plant.
Thermocouples around Copper Tubes and above Concrete Surface

Figure 2. The lay out of the CO\textsubscript{2} system and the rink floor.

However, this was considered when analysing the results. Each circuit was equipped with an adjustment valve to enable individual flow regulation. This is to be able to adjust the flow in each section of the rink. The target is to achieve as even ice surface temperature as possible to assure good ice quality.

3.3 Measurement equipment

To measure the cooling capacity of the CO\textsubscript{2}-module, the brine side was equipped with an inductive type flow meter and the temperature difference across the CO\textsubscript{2} condenser was measured. To have a comparative method for this evaluation, the CO\textsubscript{2} side of the siphon was equipped with a CO\textsubscript{2} flow meter of turbine type. The capacity on this side is calculated as the CO\textsubscript{2} mass flow times the latent heat of vaporization/condensation of the CO\textsubscript{2} at the measured condensing temperature. Another CO\textsubscript{2} flow meter was installed on the header feeding the ice rink, which would measure the flow of CO\textsubscript{2} into the circuits and consequently the CR could be calculated.

Since essentially the entire system is two-phase, there is little interest in measuring temperatures of the CO\textsubscript{2}. However, pressures are measured on the circuits as can be seen in figure 2. For the analysis of the tubes with and without the PE foil, the different types of tubes were equipped with thermocouples on the surface and between the tubes. These measurements would show the temperature values of different points in the geometry and to be compared with the FEM analysis referred to in section 2.1.
4. EXPERIMENTAL RESULTS

It was not an objective of this project to find the perfect correlations for designing ice rinks, but to find a “close candidate”. These correlations were used to evaluate pressure drop and heat transfer and verification is made in the described prototype.

4.1 Heat transfer analysis

The heat transfer model use temperature of the CO\textsubscript{2} as input and from this the ice surface temperature is reached by stepping through the layers of the ice rink bed. The CO\textsubscript{2} temperature is given by measuring the pressure in the tubes of the ice rink and this will be the input CO\textsubscript{2} evaporating temperature for the model. The results for different values of CR can be seen in table 1.

Table 1. Comparative table between calculations and experiments

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>-8.86</td>
<td>136</td>
<td>1.67</td>
<td>-8.53</td>
<td>-3.17</td>
<td>-3.00</td>
</tr>
<tr>
<td>-8.85</td>
<td>135</td>
<td>1.90</td>
<td>-8.54</td>
<td>-3.02</td>
<td>-3.05</td>
</tr>
<tr>
<td>-8.83</td>
<td>138</td>
<td>2.18</td>
<td>-8.54</td>
<td>-2.96</td>
<td>-2.93</td>
</tr>
</tbody>
</table>

As can be seen in the table the overall results from the measurement and the simulations are very close. However, when studying temperature differences across different layers of the geometry there are deviations, which is further discussed in (Shazhad 2006). It is anyway concluded that the model is good enough to give good indications on the thermal behaviour of the different geometries tested.

4.2 Pressure drop analysis and CR evaluation

There will be a trade off between pressure drop and heat transfer as a function of the circulated carbon dioxide mass flow. It was investigated if there is a circulation rate that achieves best possible heat transfer without causing too high pressure drop. The temperature level of the CO\textsubscript{2} will affect the evaporation temperature in the refrigeration machines for the ice rink, and this is directly connected to the energy efficiency (COP). There will be a solution for how many meters of tube (loops) that has to be used to manage the heat transfer and keep ice surface temperature equally distributed. The length will cause a pressure drop that has to be overcome by higher pumping power and also a lower temperature of the CO\textsubscript{2} to compensate the accompanying temperature drop. Different values for CR are calculated with different pressure drop correlations and the results are compared with measurements (see table 2).

Table 2. Pressure drop in 120m long 12.7 mm copper tubes

<table>
<thead>
<tr>
<th>CR</th>
<th>Heat Flux W/m\textsuperscript{2}</th>
<th>Average Homogeneous Pressure Drop kPa</th>
<th>Average Friedel Pressure Drop kPa</th>
<th>Experimental Pressure Drop kPa</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.67</td>
<td>136</td>
<td>6.84</td>
<td>10.44</td>
<td>8.57</td>
</tr>
<tr>
<td>1.90</td>
<td>135</td>
<td>7.86</td>
<td>12.20</td>
<td>8.98</td>
</tr>
<tr>
<td>2.18</td>
<td>138</td>
<td>9.65</td>
<td>15.01</td>
<td>9.57</td>
</tr>
</tbody>
</table>

Two pressure drop correlations are compared with the experimental data. The first is a one-phase correlation in which the “average homogenous” properties along the tube are used. The second
correlation is the Friedel two-phase correlation (Reiberer 1998) where the two-phase multiplier factor is calculated for inlet and outlet conditions and then averaged (average two-phase).

Although the homogenous model predicts the level better it under predicts at lower CRs, which lead the choice to the Friedel correlation which over predict more but gives a better trend. In general it was also concluded that it is better to over than to under predict, so the Friedel correlation was used for the following analysis.

4.2 Circulation rate (CR) evaluation
The choice of CR may be a trade off between pressure drop and heat transfer. In this section these two phenomena are compared to each other to find the minimum “temperature drop” consisting of pressure drop and heat transfer. This evaluation is based on calculations with the Friedel correlations, mentioned above, and the heat transfer correlation of Pierre 1953 (Granryd et al) which is originally used for R12, R22 and R502.

CR is plotted on x-axis and “temperature drop” is plotted on y-axis. With the increase of CR, temperature drop due to convections is decreasing while temperature drop due to pressure drop is increasing. The sums of the temperature drop are also plotted to find the optimum CR. The variation in heat load makes it difficult to find a general CR that fit all situations. When there is a high heat load, a low CR is better and when heat load is low the CR can be higher. The conclusion from both theoretical and experimental tests is that a CR as close to 1 as possible is preferable.

![Figure 3. Comparison of CR in ½” copper tubes for different heat flux](image)

The practical benefits of the results from the graphs in figure 3 are interesting. It shows that at lower heat loads a higher CR is beneficial whereas at higher heat loads a lower CR is desired. In practice the differences are very small so the CR should be kept as low as possible to save pump energy.

4.3 Investment estimation
One of the objectives of this project was to find a good design solution for a new full scale ice rink that uses CO₂ as refrigerant. The core issue is the cost of the investment versus the potential benefit in operational cost.
4.3.1 Investment per circuit

Headers are located on the short side of the ice rink and both supply and return headers are at same side of the rink. The size of the ice rink is 60x30 m and this makes one circuit 120 meter long. Since copper tubes with plastic foil are not available in 120 meter length, it was decided to use two 60 meter long copper tubes and make a soldered connection between them at the rink side opposite of the headers.

Labour cost for one circuit is calculated by a rough estimation of time. 5 min for rolling down the whole circuit on its place, 5 min for soldering the two ends of tube to both supply and return headers and 10 min for soldering the both 60 meter copper tubes with each other. This makes together 20 minutes and with a Swedish labour rate of 480 SEK/hr (1 SEK ≈ 0.11Euro) and a 120 meter long tube that cost 18 SEK/meter the total cost of one loop is 2356 SEK.

Investment for the whole rink is calculated in table 3. With the decrease of the pitch, number of circuits of copper tubes increases. The depreciation time of 10 years is taken which gives investment per year. Investment per year increases with the decrease of pitch of the copper tubes inside the rink bed.

Table 3. Investment per year of 12.7 mm copper tubes with plastic foil in different pitches

<table>
<thead>
<tr>
<th>Pitch [mm]</th>
<th>No. of Circuits</th>
<th>Investment [SEK]</th>
<th>Investment per Year (10 year) [SEK/Yr]</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>150</td>
<td>353400</td>
<td>35340.0</td>
</tr>
<tr>
<td>95</td>
<td>158</td>
<td>372248</td>
<td>37224.8</td>
</tr>
<tr>
<td>90</td>
<td>166</td>
<td>391096</td>
<td>39109.6</td>
</tr>
<tr>
<td>85</td>
<td>176</td>
<td>414656</td>
<td>41465.6</td>
</tr>
<tr>
<td>80</td>
<td>188</td>
<td>442928</td>
<td>44292.8</td>
</tr>
<tr>
<td>75</td>
<td>200</td>
<td>471200</td>
<td>47120.0</td>
</tr>
</tbody>
</table>

4.3.2 Economical analysis of copper tube layout for ice rink

Operational cost per year is calculated for different pitches in which only cost for compressor energy is taken. It is assumed that pressure drop will remain same with the change of pitch and hence pump power will remain same for each pitch that’s why cost for pump energy is not included in this analysis. The average heat flux is 50 W/m² and area of the ice rink is 1800 m² which gives a cooling load for the ice rink of 90 kW.

Table 4. Operational cost per year at heat load of 50 W/m², 8000 hours at a cost of 1 SEK/kWh

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>-6.03</td>
<td>-9.03</td>
<td>3.65</td>
<td>24.658</td>
<td>197260</td>
</tr>
<tr>
<td>95</td>
<td>-5.97</td>
<td>-8.97</td>
<td>3.66</td>
<td>24.590</td>
<td>196721</td>
</tr>
<tr>
<td>90</td>
<td>-5.91</td>
<td>-8.91</td>
<td>3.67</td>
<td>24.523</td>
<td>196185</td>
</tr>
<tr>
<td>85</td>
<td>-5.86</td>
<td>-8.86</td>
<td>3.68</td>
<td>24.457</td>
<td>195652</td>
</tr>
<tr>
<td>80</td>
<td>-5.81</td>
<td>-8.81</td>
<td>3.69</td>
<td>24.390</td>
<td>195122</td>
</tr>
<tr>
<td>75</td>
<td>-5.76</td>
<td>-8.76</td>
<td>3.70</td>
<td>24.324</td>
<td>194595</td>
</tr>
</tbody>
</table>

The CO₂ is assumed to be cooled by ammonia refrigeration units with a temperature difference of 3°C between the CO₂ temperature and ammonia evaporation temperature. COP of the Ammonia refrigeration system is based on the BOCK selection program for compressors. A condensing temperature of 30°C is assumed and the corresponding ammonia evaporation temperature is used. Since the cooling load is known the compressor power can be calculated. The Swedish electricity rate is assumed to be 1 SEK/kWh which gives the operational cost per year. With the
increase of average heat load the CO\textsubscript{2} fluid temperature must be lowered resulting in lower NH\textsubscript{3} evaporation temperature. This will lower the COP of the machinery and an increase in compressor power and finally operational cost per year as can be seen in table 5.

Table 5. Annual operational at different average heat loads

<table>
<thead>
<tr>
<th>Pitch mm</th>
<th>50 W/m\textsuperscript{2}</th>
<th>100 W/m\textsuperscript{2}</th>
<th>150 W/m\textsuperscript{2}</th>
<th>200 W/m\textsuperscript{2}</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>197260</td>
<td>426036</td>
<td>690096</td>
<td>993103</td>
</tr>
<tr>
<td>95</td>
<td>196721</td>
<td>423529</td>
<td>683544</td>
<td>982935</td>
</tr>
<tr>
<td>90</td>
<td>196185</td>
<td>421053</td>
<td>679245</td>
<td>972973</td>
</tr>
<tr>
<td>85</td>
<td>195652</td>
<td>419825</td>
<td>675000</td>
<td>966443</td>
</tr>
<tr>
<td>80</td>
<td>195122</td>
<td>417391</td>
<td>670807</td>
<td>956811</td>
</tr>
<tr>
<td>75</td>
<td>194595</td>
<td>416185</td>
<td>664615</td>
<td>947368</td>
</tr>
</tbody>
</table>

The results in table 3 and 5 can be used to support investment decision based on what type of ice rink application to be designed. A plant with high load and long operational time benefit from a tighter pitch whereas a smaller ice rink with less heat load will manage with a baseline 100mm pitch.

4. CONCLUSIONS

The headers are preferably placed on the short end side of the ice rink which implies that the length of the copper tubes will be 120 meter. This circuit length does not lead to extensive pressure drop for \( \frac{1}{2} \)” tubes even at design loads as high as 300W/m\textsuperscript{2}.

A heat transfer model has been developed that predicts the temperature distribution very well in the ice rink floor. This model has been used to analyse the influence of different geometries of tubes, pitches, concrete thickness, etc.

The experimental results have been compared to pressure drop and heat transfer correlations although it was not within the scope to develop correlations. The best suited correlations were used to perform numerical optimisations.

The influence of the circulation rate was found to have little impact on the heat transfer so no optimum could be seen. In practice this means the CR should be chosen as low as possible to reduce pump power.

An attempt to economically optimise the geometry based on the average heat load is presented which yields investment and operational cost for different pitches.

5. ACKNOWLEDGMENTS

Special thanks to the financiers of the project: Katrineholms Kommun, Outokumpu Copper Products AB, Sörmlands Sparbank/Tillväxtbanken and The Swedish Energy Agency.

REFERENCES


