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# GEOTHERMAL DISTRICT HEATING
# THE ICELAND EXPERIENCE

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UNU Geothermal Training Programme, Iceland
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GEOTHERMAL DISTRICT HEATING

The Iceland Experience

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# Foreword

These notes on the design of geothermal district heating systems have been written for use in the Geothermal Training Programme of the United Nations University in Iceland. They describe and relate the experience gained in the design and operation of geothermal district heating systems in Iceland, where today over three quarters of the population live in homes heated by geothermal energy.

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## LIST OF SYMBOLS

### (a) Latin letter symbols

A : radiator surface area, $m^2$

a = $(k_r + k_l)/m$ building parameter

b = $k_l/(k_r + k_l)$ building parameter

C : heat capacity of items inside a heated building, kJ/K

$c_j$: specific heat of item j inside a building, kJ/kg·K

$c_w$: specific heat of water, kJ/kg·K

DD(T): annual degree days below temperature T

$F_d$: annual district heating energy requirement, kJ/$m^3$·year

$F_h$: annual space heating energy consumption, kJ/$m^3$·year

$F_v$: gross exterior wall area of building, $m^2$

G : annual space heating energy consumption in district, kJ/person·year

$G_h$: annual heating energy consumption in district, kJ/person.year

$G_t$: annual energy consumption for hot tap water, kJ/person·year

H : peak daily thermal power demand for district, kW

H': maximum average daily thermal power demand, kW

h : enthalpy of water, kJ/kg; insulation layer thickness, m

$K_l$: overall heat transfer coefficient of building, W/K

$K_r$: combined heat transfer coefficient for all radiators in building, W/K

k : radiator heat transfer coefficient, kJ/$m^2$·K

$k_l$: normalized heat transfer coefficient of building $= K_l/F_v$

$k_r$: normalized radiator heat transfer coefficient $= K_r/F_v$

L : length of pipe, m

m : normalized heat capacity of items inside building $= C/F_v$

$m_j$: mass of item j inside building, kg

$\dot{m}$ : mass water flow rate, kg/s

$N_T$: number of days in year with mean temperature below T

P : population of heated district

$Q$ : heat loss from building or pipe, W

$q_n$: hot tap water use, l/s·100 persons

$R$ : thermal heat flow resistance, m·°C/W

$r$ : radius, m

$T$ : temperature, °C

$T_b$: return water temperature from radiators, °C

$T_d$: depth of cold wave below system design temperature

$T_f$: inflow water temperature to radiators, °C

$T_g$: system design temperature, °C

$T_i$: room temperature, °C

$T_k$: $T_o - T_g$

$T_m$: annual mean air temperature, °C

$T_{mT}$: mean temperature for $N_T$ days in year, °C

$T_o$: outside air temperature, also borehole temperature, °C

$T_p$: fluid temperature in pipe, °C

$T_r$: mean water temperature in radiators, °C

$\Delta T_m$: logarithmic mean temperature difference, °C

$t$ : time

$t_o$: duration of cold wave, days

$V$ : building volume in district, $m^3$/person

$x$ : double distribution system, fraction of total

(b) Greek letter symbols

$\alpha$ : overall correction factor for district heating system heat demand

$\beta$ : peak load factor for district heating

$\eta$ : efficiency

$\kappa$ : geothermal fluid volume as fraction of 80°C water volume

$\lambda$ : thermal conductivity, W/m·K

$\rho$ : density, $kg/m^3$

$\phi$ : phase angle for sinusoidal cold wave = $\tan^{-1}(\omega/a)$

$\omega$ : frequency of sinusoidal cold wave = $\pi/t_o$

# 1. INTRODUCTION

District heating is defined as the grouping together of several buildings and using a boiler plant or any other source of heat to heat a number of dwellings or blocks of buildings. It has been found that the larger the consumer network in such a case, the more economical the boiler plant can be run. For this reason it is the practice in many countries today to heat entire sections of towns and cities, and in some cases even clusters of towns from a central heating plant. Among countries where extensive district heating systems have been built are the Soviet Union, West Germany, France, Austria, Holland, Belgium, Switzerland, Norway, Sweden, Denmark, United States of America, and Iceland.

Most district heating systems use conventional fuel (oil, natural gas or coal) as the source of heat. Many of these systems are operated in connection with electric generating power plants. For such plants it is not uncommon to use nuclear energy as the source of heat. In some areas geothermal heat is used as the district heating source and experience has shown that low temperature geothermal fluid ($<120^{\circ}C$) is well suited for this purpose. Geothermal district heating systems are operated in France, Hungary and lately such systems have found increased use in the western part of the United States. The best known of the geothermal district heating systems, however, is the hot water supply system of Reykjavik, the capital city of Iceland. Geothermal district heating systems are becoming commonplace in Iceland and by now (June 1981) it is estimated that over 75% of the Icelandic population enjoys geothermal heat in their homes. The Icelandic authorities have set it as a goal to eliminate fuel burning as a source of space heat altogether by the year 1990. For areas where geothermal heat is not available, space heating will then be done by means of electricity. The experience in Iceland in the field of geothermal space heating will be the main subject of this report.

The heat distribution in district heating systems is carried out by the use of either hot water or steam. In the United

States as well as in most countries of Western Europe steam is the more common heat carrying medium whereas in Eastern Europe and in the Scandinavian countries the hot water method is most commonly employed. Geothermal district heating is done exclusively by the use of hot water which is natural since the heat source is generally the low temperature geothermal fluid.

The Reykjavik district heating system was put into operation on a small scale in 1930 when over 80°C hot water from wells within the city was piped into a hospital, a swimming hall, two school buildings and 70 homes in the vicinity of the swimming hall.

In 1943 a major expansion of the Reykjavik district heating system was completed when the system was extended to a good part of the city at that time or to a population of approximately 30,000. The water was at that time piped from the Reykir geothermal area, about 16 kilometers to the northeast of the city. The capacity of the heating system was increased slowly for the next fifteen years while at the same time the city was expanding rapidly. In 1959 a modern rotary drilling rig was purchased and since then a rapid expansion of the Reykjavik district heating system has taken place with the result that today the total population of Reykjavik as well as that of the neighbouring towns of Kopavogur, Gardabaer, Bessastadahreppur and Hafnarfjordur are connected to the system. This means that the Reykjavik heating system today serves a population of over 115,000 people and the capacity of the system is estimated at approximately 475 MW based on utilization of the heat content of the geothermal fluid to a temperature of 40°C (Gudmundsson and Palmason, 1981).

Parallel to the rapid expansion of the Reykjavik district heating system, other heating systems have been designed and built at various places around the country. These are almost exclusively based on geothermal energy as the source of heat. It is projected that by the year 1990 about 80% of the population of the country will be using geothermal heating and the remaining 20% will be heating their homes with electricity (Orkusparnefnd, 1980).

## 2. DISTRICT HEATING DEMAND

When the district heating of a section of a town is being planned it is necessary to determine the heating requirements of the section. The most exact method of doing this would be to consider each building separately and find its requirements for the most severe weather conditions to which the town is expected to be exposed. The total need for the section in question is then found as the sum of all such individual building needs.

This method, however, is very time consuming, and district heating systems are also always designed in such a way that they are flexible and some room left for expansion. Great exactitude in the heating requirements of individual buildings is therefore not warranted and other less tedious ways are available which give sufficiently reliable information about the heating requirements of the district under consideration. Two such methods, which both have been used for the design of district heating systems in Iceland are described below.

### 2.1 Heat requirements based on size and type of buildings

Over the years the following rule of thumb has been developed for the Reykjavik district heating system for the maximum average daily heating requirement of buildings connected to the system with direct hot tap water connection:

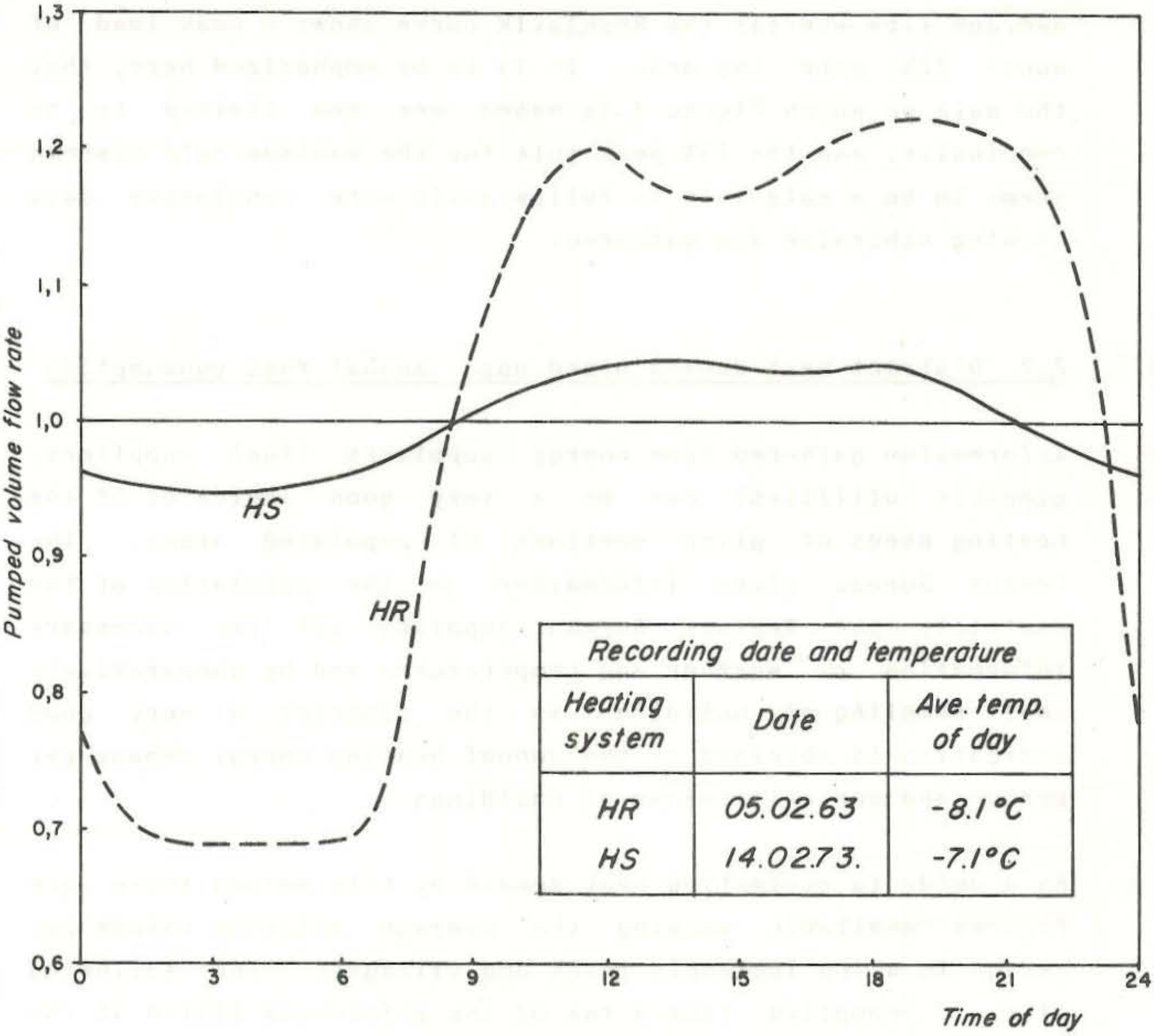
| | |
|---|---|
| One story buildings | 24.4 W/m$^3$ |
| Two story buildings | 22.1 W/m$^3$ |
| Three story buildings | 19.8 W/m$^3$ |
| Four story buildings or higher | 17.4 W/m$^3$ |

Since these figures represent the average <u>daily</u> requirement, the maximum <u>hourly</u> demand is estimated to be 30% higher. It is also estimated that heat losses in the distribution network are on the order of 10% of the above figures.

For the design of district heating systems outside the Reykjavik area the above figures have been modified in accordance with the distribution of degree days (see definition in section 3.1.) at the location in question in comparison with the degree day distribution in Reykjavik.

At this point it might be pointed out, that the method of metering the hot water may affect the maximum hourly demand as compared to the maximum average daily demand. In the Reykjavik district heating system the charge to the customer is based on the total volume of water used as measured by a regular flowmeter. All other district heating systems in Iceland, on the other hand, base their charges on the maximum rate of flow to the customer. The difference between these two methods will be discussed in more detail later, but it is mentioned here since it will influence the way the heat is used. The customer who is charged for the heat on the basis of the maximum rate of flow is more likely to distribute the use of the heat more evenly throughout the day, thus reducing the peak load during the day or even eliminating it altogether. For this reason the maximum hourly demand in district heating systems using the maximum rate method is usually estimated at 15% over the average maximum daily demand instead of 30% as done in Reykjavik.

A reduction of the peak demand over the average daily demand from 30% for the total flow system to 15% for the maximum rate system may be underestimated, although data on this point are insufficient and too limited to be conclusive. An investigation made by this author (Karlsson 1975), however, indicates that the daily variation from the average flow for a maximum rate system is much smaller than the 15% estimated in most designs. This is shown in Figure 1, which shows the pumped volume of water from two different district heating networks in Iceland as a function of time on a cold day. The solid curve is that for the Seltjarnarnes district heating system and the dashed line shows the flow variation at one pumping station in the Reykjavik district heating network.




Variation of pumped volume flow rate on a cold day

Fig. I

Both curves are normalized with an ordinate of 1.0 giving the average flow during the day. Seltjarnarnes is a suburb of Reykjavik with its own independent district heating network which uses the maximum flow rate method. It appears that the peak load at the Seltjarnarnes network is only 4-5% over the average flow whereas the Reykjavik curve shows a peak load of about 22% over the mean. It is to be emphasized here, that the data on which Figure 1 is based are too limited to be conclusive, and the 15% peak rule for the maximum rate systems seems to be a safe rule to follow until more conclusive data showing otherwise are gathered.

## 2.2 District heat demand based upon annual fuel consumption

Information gathered from energy suppliers (fuel suppliers, electric utilities) can be a very good indicator of the heating needs of given sections of populated areas. The Census Bureau gives information on the population of the district, the Weather Bureau supplies all the necessary information on weather and temperatures and by comparatively small sampling of buildings in the district a very good indication is obtained of the annual heating energy demand per person and per unit volume of buildings.

As a guide to estimating heat demand by this method there are figures available showing the average building volume per person in a few Icelandic towns and villages. The following list is compiled from a few of the references listed at the end of this report (Fjarhitun, Ltd., 1970, Virkir 1970, Verkfraedistofa Nordurlands, Ltd., and Verkfraedistofa Sigurdar Thoroddsen, Ltd., 1976):

| | | | |
|---|---|---|---|
| Akranes | 127 $m^3$/person | Kópavogur | 105 $m^3$/person |
| Akureyri | 106 - - | Reykjavík | 145 - - |
| Garðabær | 115 - - | Seltjarnarnes | 116 - - |
| Hafnarfjörður | 115 - - | Siglufjörður | 107 - - |

As a rule, houses in Iceland located outside the geothermal district heating areas are heated by one of two means:

1: Fuel oil
2: Electricity a) Panel heaters, b) Hot water tank

The fuel oil heating systems are almost exclusively hot water heating systems where hot water is piped through the buildings with radiators supplying and distributing the heat to each individual room. A few systems are of the radiant panel heating design with radiant panels embedded either in the ceiling or in the floor, with hot water passing through the panels. Warm air heating systems are also found occasionally, but they may be considered as exception.

In estimating the heating need of a building from the annual consumption of heating fuel oil the heating value of the fuel oil and the efficiency of the heat generating boiler must be known. For the fuel oil used in Iceland the following values have been assumed:

| | |
|---|---|
| Density of fuel oil | 0.85 kg/l |
| Heating value | 10,000 kcal/kg = 41,868 kJ/kg |

The efficiency of the heat generating hot water boiler is often assumed to be $\eta = 0.60$ from which the heat demand of a building heated by fuel oil is easily estimated.

For electric panel heaters the heating efficiency is assumed to be $\eta = 0.95$. Electric heating of a hot water tank is similar to hot water heating with the tank replacing the hot water boiler. The efficiency of the electric hot water heating is assumed $\eta = 0.85$.

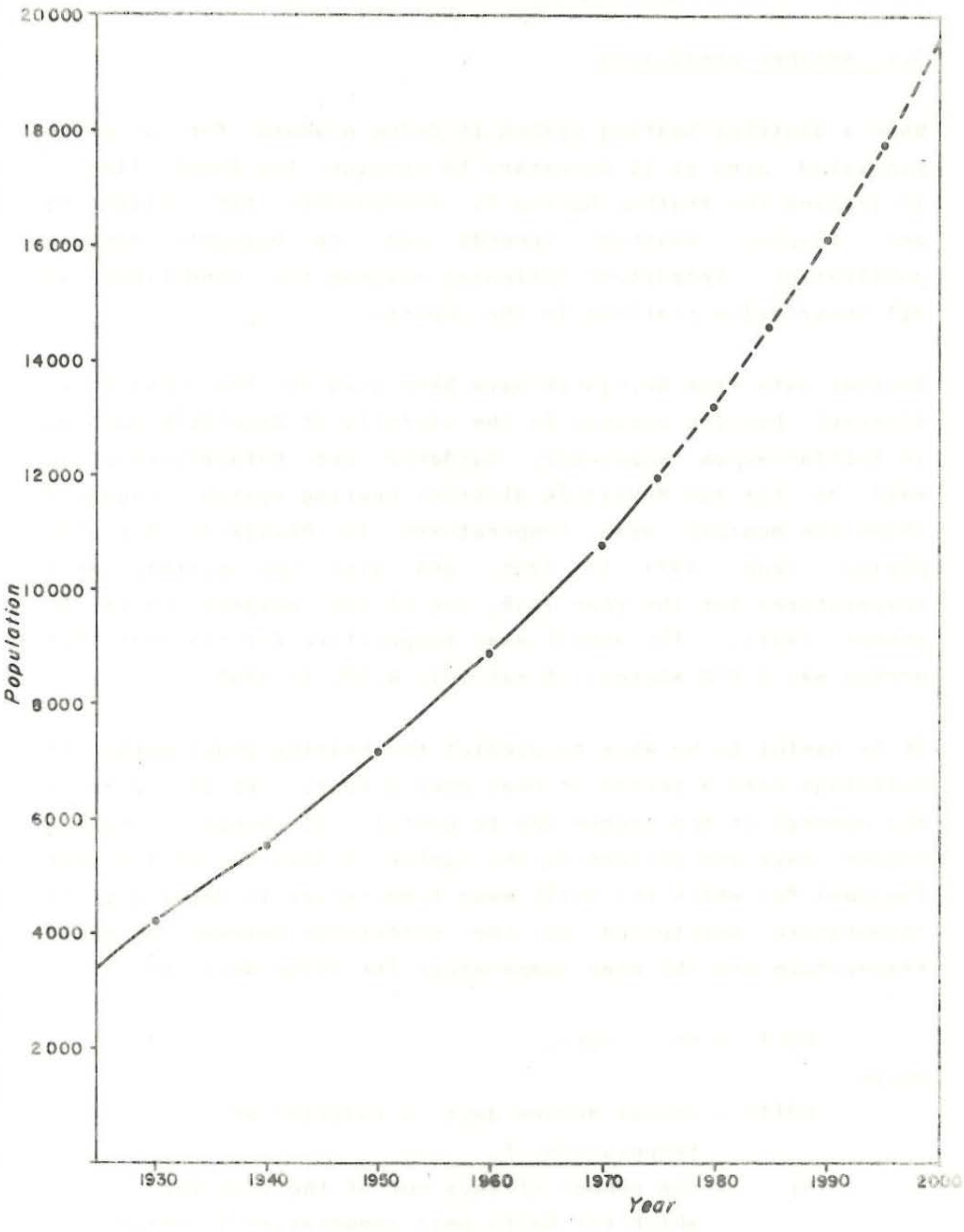
On basis of the results of the above analysis for a limited but representative sample of buildings in the district (approx. 5% of total buildings) as well as information on the total heat energy consumption of the district an estimate is obtained for the heat demand of the district per unit volume of buildings and per person living in the district.

## 2.3. Population trends

Since the district heating system is designed to be in operation for a number of years it is necessary to study how the population of the district has developed in the past and attempt to make a forecast or an educated guess about future developments. An example of such a forecast is presented in Figure 2, showing the population of Akureyri, a town in northern Iceland where the construction of a district heating system was recently completed (Verkfraedistofa Nordurlands and Verkfraedistofa Sigurdar Thoroddsen, Ltd., 1976). The forecast calls for a continuous annual population growth of 2% until the turn of the century. This is a little higher rate of growth than has been projected for the country as a whole in a recent study (Orkusparnefnd 1980), where an annual population increase of 1.0% has been forecast.

Another factor which affects the future heat demand is the size of houses in which the general population lives. It has been observed in Iceland that the space per individual person has been on the increase in the past few decades and this trend is expected to continue. In Akureyri it was estimated that the annual rate of growth of living space per person would be about 2% (Verkfraedistofa Nordurlands, Ltd., and Verkfraedistofa Sigurdar Thoroddsen, Ltd., 1976). For the country as a whole the annual growth rate is assumed to fall off linearly over the twenty years period from 1980 to 2000, starting with 2.5% in 1980 and decreasing to 1.5% in 2000 (Orkusparnefnd 1980).

Both of the above factors must be taken into consideration when a district heating system is being planned. The design of some of the major components of the system such as supply mains and distribution trunk lines are often based on the 15 to 20 years projected future needs.




Population of Akureyri, Iceland

Fig. 2

# 3. HEATING REQUIREMENTS DETERMINED BY ANALYTICAL METHODS

## 3.1. Weather conditions

When a district heating system is being planned for a given populated area it is necessary to consider the local climate. In Iceland the Weather Bureau is responsible for collecting and keeping weather records and the Bureau's monthly publication, "Veðráttan" (Climate) reviews the conditions at all observation stations in the country.

Weather data from Reykjavík have been used in the design of district heating systems in the vicinity of Reykjavík such as in Seltjarnarnes, Kópavogur, Garðabær and Hafnarfjörður as well as for the Reykjavík district heating system. Figure 3 shows the monthly mean temperatures in Reykjavík for the period from 1931 to 1968, and also the monthly mean temperatures for the year 1969, one of the coldest years in recent years. The annual mean temperature for the 1931-1968 period was 4.9°C whereas it was only 4.0°C in 1969.

It is useful to be able to predict the heating requirement of buildings over a season or even over a year. For this purpose the concept of the degree day is useful, but annual (seasonal) degree days are defined as the number of days out of the year (season) for which the daily mean temperature is below a given temperature multiplied by the difference between the given temperature and the mean temperature for these days, or

$$DD(T) = N_T \; (T - T_{mT}), \tag{1}$$

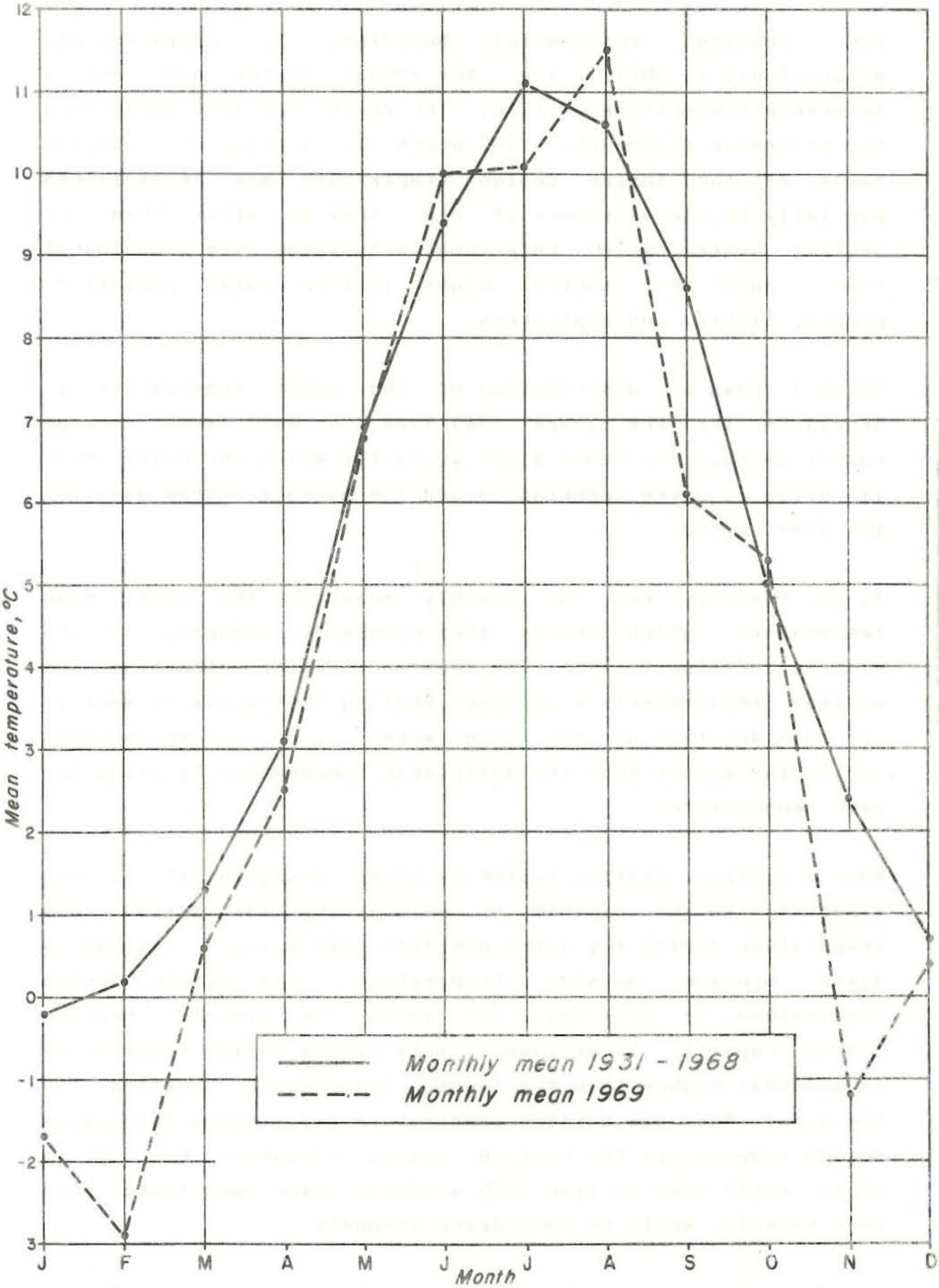
where

$DD(T)$ = annual degree days, a function of temperature, $T$,

$N_T$ = the number of days out of the year for which the daily mean temperature is below $T$,

$T_{mT}$ = the mean temperature for these $N_T$ days.

It has been found (Threlkeld, 1970) that energy consumption




Monthly mean temperatures in Reykjavik

Fig. 3

for heating residential buildings is approximately proportional to DD(17), i.e. the annual degree days for a reference temperature of 17°C. The reason for 17°C being used for reference rather than 20°C which in Iceland is usually taken as the inside design temperature may be explained partially by the existence of heat sources other than the heating system which help heat a building over a period of time. Such heat sources might include solar radiation, people, lights, and appliances.

Table I shows the distribution of the daily temperature in Reykjavik for the years 1961-1968, as well as the average number of days for those eight years for which the daily mean is below a given temperature and the average degree days for the same period.

It is, however, not the monthly mean or the daily mean temperature which decide the necessary capacity of the district heating system. The important factor affecting the maximum requirements a district heating system has to meet is the consideration of the cold waves, i.e. those periods during the winter when the daily mean temperature is below the mean temperature.

When a district heating system is being designed it is not expected to be capable of maintaining the design room temperature during the worst possible cold waves. Instead a given minimum outside temperature, the system design temperature, is selected, for which the district heating system capacity is designed. This system design temperature is somewhat higher than the lowest temperature expected for the area. When the outside temperature falls below the system design temperature the heating system consumers may for a short while have to live with a little lower room temperature than normally would be considered adequate.

For determination of the system design temperature it is necessary to study the available weather data for the area and

## Table I

Distribution of daily mean temperatures and degree days in Reykjavik.
Average values for the years 1961-1968.

| Daily mean, °C T-(T+1) | 1961 | 1962 | 1963 | 1964 | 1965 | 1966 | 1967 | 1968 | Total | Mean | $N_{T+1}$ | DD(T+1) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 19-20 | | | | | | | | | | | 365,3 | 5719,0 |
| 18-19 | | | | | | | | | | | 365,3 | 5353,7 |
| 17-18 | | | | | | | | | | | 365,3 | 4988,4 |
| 16-17 | | | | | | | | | | | 365,3 | 4623,1 |
| 15-16 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0,0 | 365,3 | 4257,8 |
| 14-15 | 0 | 1 | 0 | 1 | 1 | 1 | 0 | 2 | 6 | 0,8 | 365,3 | 3892,5 |
| 13-14 | 2 | 1 | 3 | 3 | 9 | 7 | 0 | 10 | 35 | 4,4 | 364,5 | 3527,2 |
| 12-13 | 8 | 8 | 3 | 9 | 12 | 16 | 4 | 15 | 75 | 9,4 | 360,1 | 3162,7 |
| 11-12 | 25 | 22 | 14 | 12 | 20 | 19 | 21 | 21 | 154 | 19,3 | 350,7 | 2802,6 |
| 10-11 | 29 | 32 | 33 | 26 | 24 | 28 | 33 | 29 | 234 | 29,2 | 331,4 | 2451,9 |
| 9-10 | 47 | 28 | 36 | 34 | 31 | 27 | 26 | 33 | 262 | 32,7 | 302,2 | 2120,5 |
| 8-9 | 43 | 34 | 20 | 33 | 31 | 22 | 31 | 22 | 236 | 29,4 | 269,5 | 1818,3 |
| 7-8 | 25 | 26 | 21 | 42 | 41 | 11 | 21 | 19 | 206 | 25,7 | 240,1 | 1548,8 |
| 6-7 | 8 | 30 | 25 | 46 | 26 | 23 | 22 | 16 | 196 | 24,4 | 214,4 | 1308,7 |
| 5-6 | 15 | 12 | 34 | 35 | 16 | 24 | 18 | 17 | 171 | 21,4 | 190,0 | 1094,3 |
| 4-5 | 25 | 17 | 39 | 30 | 14 | 15 | 22 | 21 | 183 | 22,8 | 168,6 | 904,3 |
| 3-4 | 16 | 20 | 31 | 22 | 22 | 20 | 19 | 21 | 171 | 21,4 | 145,8 | 735,7 |
| 2-3 | 16 | 21 | 24 | 12 | 12 | 27 | 23 | 22 | 157 | 19,6 | 124,4 | 589,9 |
| 1-2 | 21 | 20 | 25 | 12 | 13 | 25 | 23 | 31 | 170 | 21,2 | 104,8 | 465,5 |
| 0-1 | 13 | 13 | 9 | 11 | 19 | 12 | 14 | 13 | 104 | 13,0 | 83,6 | 360,7 |
| -1—0 | 23 | 15 | 7 | 4 | 18 | 16 | 17 | 15 | 115 | 14,4 | 70,6 | 277,1 |
| -2—1 | 5 | 9 | 9 | 6 | 15 | 16 | 17 | 11 | 88 | 11,0 | 56,2 | 206,5 |
| -3—2 | 9 | 18 | 9 | 9 | 7 | 13 | 17 | 9 | 91 | 11,4 | 45,2 | 150,3 |
| -4—3 | 12 | 11 | 4 | 5 | 7 | 15 | 9 | 3 | 66 | 8,3 | 32,8 | 105,1 |
| -5—4 | 8 | 16 | 8 | 4 | 3 | 7 | 9 | 3 | 58 | 7,3 | 24,5 | 72,3 |
| -6—5 | 4 | 6 | 1 | 5 | 9 | 6 | 5 | 7 | 43 | 5,4 | 17,2 | 47,8 |
| -7—6 | 4 | 5 | 2 | 2 | 4 | 5 | 7 | 7 | 36 | 4,5 | 11,8 | 30,6 |
| -8—7 | 2 | 0 | 3 | 0 | 6 | 1 | 4 | 8 | 24 | 3,0 | 7,3 | 18,8 |
| -9—8 | 2 | 0 | 3 | 0 | 1 | 5 | 0 | 2 | 13 | 1,6 | 4,3 | 11,5 |
| -10—9 | 1 | 0 | 2 | 3 | 3 | 1 | 3 | 5 | 18 | 2,3 | 3,7 | 7,2 |
| -11—10 | 1 | 0 | 0 | 0 | 0 | 2 | 0 | 0 | 3 | 0,4 | 1,4 | 3,5 |
| -12—11 | 0 | 0 | 0 | 0 | 1 | 0 | 0 | 1 | 2 | 0,3 | 1,0 | 2,1 |
| -13—12 | 0 | 0 | 0 | 0 | 0 | 1 | 0 | 1 | 2 | 0,3 | 0,7 | 1,1 |
| -14—13 | 1 | 0 | 0 | 0 | 0 | 0 | 0 | 2 | 3 | 0,4 | 0,4 | 0,4 |
| -15—14 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0,0 | 0,0 | 0,0 |
| | 365 | 365 | 365 | 366 | 365 | 365 | 365 | 366 | 2922 | 365,3 | - | |

to estimate the effects of the worst cold waves on the inside temperature of buildings. A cold wave is defined as a period of at least two days for which the outside daily mean temperature is below the system design temperature. The system design temperature must be selected low enough so that the maximum cooling of buildings during the most severe cold wave to be expected will not bring the inside temperature down below a predetermined value. This minimum inside temperature for which district heating systems in Iceland are designed is often taken as 17-18$^{o}$C.

## 3.2 Evaluation of cooling of buildings during cold waves

As stated above the system design temperature is defined as the outside temperature for which a district heating system is designed to maintain the desired room temperature in buildings. In Iceland this room temperature in residential buildings is usually taken to be 20$^{o}$C. A rule of thumb, sometimes used by district heating system designers in Iceland, is to choose the system design temperature so that in an average year there are about two days with a mean temperature lower than the system design temperature (see f.ex. Verkfraedistofa Nordurlands, Ltd., and Verkfraedistofa Sigurdar Thoroddsen, Ltd., 1976).

Most buildings in Iceland from recent years are made of steel reinforced poured concrete. Their mass content is therefore rather large with great heat capacity in walls and floors. The inside temperature will therefore only to a limited degree follow variations of the outside temperature, which is of great importance when the weather changes as frequently and rapidly as it does in Iceland. For this reason cold waves of limited duration will have insignificant effects on the room temperature of the buildings even though the building heating system is only designed for the outside temperature before and after the cold wave. The system design temperature is then selected so that the inside temperature decrease during the most severe cold wave is within reasonable limits. In the following section it will be outlined how this temperature decrease may be determined analytically.

The steady state rate of heat loss of a building is given by the equation

$$Q = K_l (T_i - T_o), \tag{2}$$

where

$Q$ = heat loss, W

$K_l$ = overall heat transfer coefficient of building, W/$^oC$,

$T_i$ = room temperature, $^oC$,

$T_o$ = outside air temperature, $^oC$.

For the steady state conditions it is assumed that the average temperature of inside walls, floors, furniture and any other items inside the building is $T_i$. If the mass of an item contributing to the heat capacity of the building is $m_j$ and its specific heat is $c_j$, the building heat capacity is expressed as

$$C = m_j c_j, \tag{3}$$

where repeated indices indicate summation over all items. The building heat content is then $CT_i$. If the building is insulated on the inside surface of the exterior walls, their mass is not included in C but if the insulation is placed on the outside surface, the outer walls are included in C.

It is now assumed that the building is heated with hot water radiators with the mean temperature $T_r$. The heat emitted by the radiators is

$$Q = K_r (T_r - T_i), \tag{4}$$

where

$K_r$ = combined heat transfer coefficient for all radiators in the building, W/ $^oC$.

For steady state conditions the heat flow from the radiators is equal to the building heat loss. Combining equations (2) and (4) gives then

$$T_i = K_rT_r/(K_l+K_r) + K_lT_o/(K_l+K_r). \qquad (5)$$

It is assumed that the temperature of all items within the building follows that of the inside air, $T_i$, even though $T_i$ may vary slowly. It may be shown (Bodvarsson 1954) that this is not an unreasonable assumption. The building heat losses are now normalized by dividing through by the gross exterior wall area, $F_v$, giving the following normalized parameters:

$$k_l = K_l/F_v; \quad k_r = K_r/F_v; \quad m = C/F_v.$$

During a cold wave the outside air temperature, $T_o$, is lowered below the system design temperature. This will result in changes of the inside air temperature according to the differential equation

$$-mdT_i/dt + k_r(T_r-T_i) = k_l(T_i-T_o)$$

or

$$mdT_i/dt + (k_l+k_r)T_i = k_rT_r + k_lT_o, \qquad (6)$$

where m, $k_l$, $k_r$, and $T_r$ are constants and $T_i$ and $T_o$ are changing with time. Now let

$$T_o = T_g + T_k; \quad T_i = (k_rT_r+k_lT_g)/(k_r+k_l) + T, \qquad (7)$$

where $T_g$ is the system design temperature. The differential equation (6) is then reduced to the form

$$mdT/dt + (k_l+k_r)T = k_lT_k. \qquad (8)$$

It has been suggested (Bodvarsson 1965) that all cold waves may be approximated by one of these basic types:

a) Rectangular cold waves, where the outside air temperature drops rather suddenly, then stays relatively constant for the duration of the cold wave, and finally increases suddenly again up to and above the system design temperature. This cold wave is shown in Figure 4(a)

b) Triangular cold waves, where the outside air temperature drops gradually down to a minimum and then increases gradually again up to and above the system design temperature. The shape of this cold wave is shown in Figure 4(b).

c) Sinusoidal cold waves, where the outside air temperature variation is somewhere between the above two types. The temperature-time history is then approximated by a sine curve as shown in Figure 4(c).

The differential equation (8) can be solved analytically for the above three cases. The solutions are as follows:

a) Rectangular cold wave:

$$T = bT_d\,(1-\exp(-at)), \quad 0<t<to,$$
$$T = bT_d\exp(-at)(\exp(at_o)-1), \quad t>t_o, \qquad (9)$$

where

$a = (k_r+k_l)/m$,

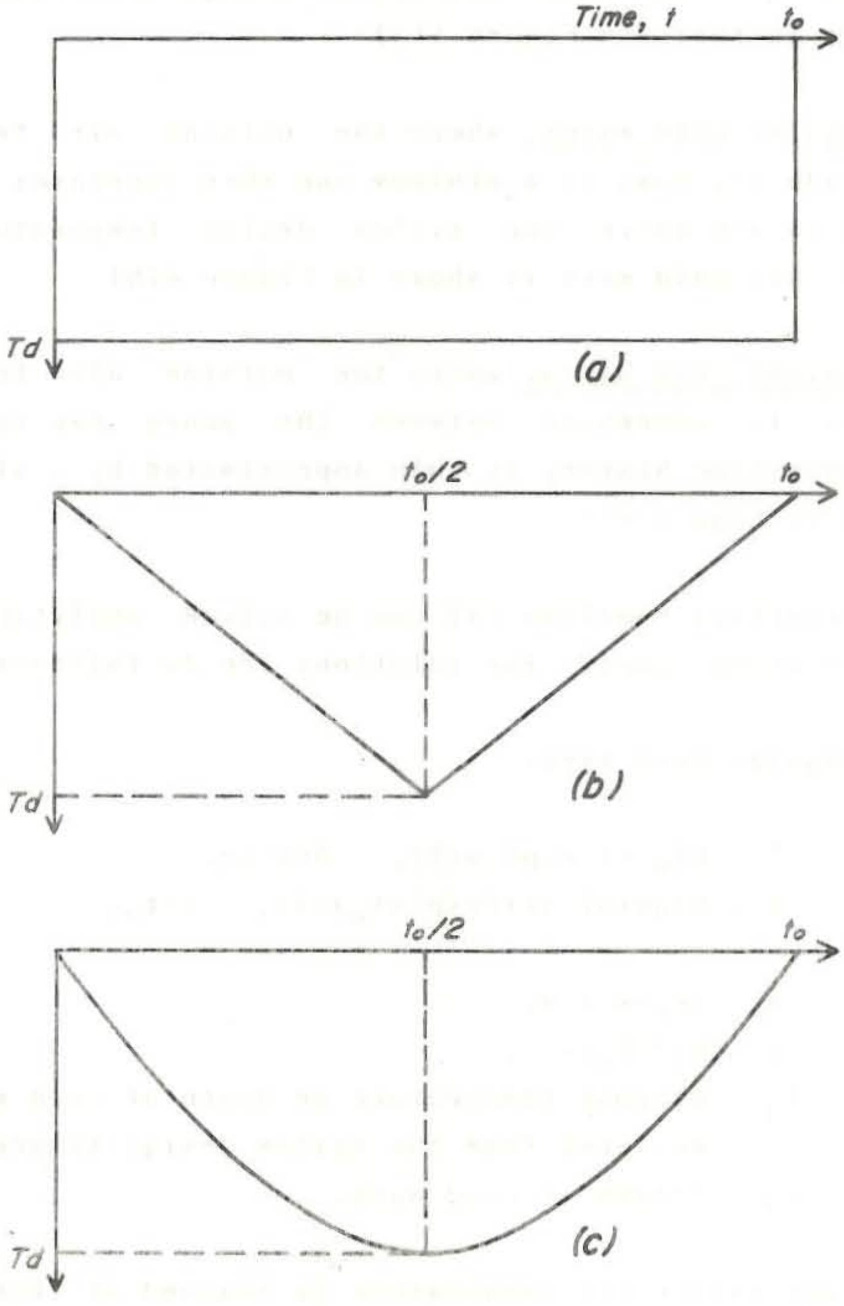
$b = k_l/(k_r+k_l)$,

$T_d$ = minimum temperature or depth of cold wave as measured from the system design temperature,

$t_o$ = length of cold wave.

The minimum inside air temperature is reached at the end of the cold wave, for $t = t_o$, i.e.

$$T_{min} = bT_d(1-\exp(-at_o)). \qquad (10)$$




Cold wave approximations

Fig. 4

b) Triangular cold wave:

$$T = 2bT_d(at+\exp(-at)-1)/at_o, \quad 0<t<t_o/2,$$
$$T = 2bT_d(1+a(t_o-t) - \exp(-at)(2\exp(at_o/2)-1))/at_o, \quad t_o/2<t<t_o. \tag{11}$$

The minimum inside air temperature is encountered at the time

$$t = \ln(2\exp(at_o/2)-1)/a, \tag{12}$$

and its value is

$$T_{min} = 2bT_d(1-\ln(2\exp(at_o/2)-1)/at_o). \tag{13}$$

c) Sinusoidal cold wave

$$T = bT_d \cos(\omega t-\phi)/\sqrt{1 + (\omega/a)^2}, \tag{14}$$

where $\omega$ is the frequency of the temperature oscillation, i.e. $\omega = \pi/t_o$, and $\phi = \tan^{-1}(\omega/a)$. The minimum temperature is

$$T_{min} = bT_d/\sqrt{1 + (\omega/a)^2}. \tag{15}$$

The depth of the cold wave, $T_d$, is determined in such a way that the degree days for the cold wave measured from the system design temperature, $DD(T_g)$, is equal to the degree days of the approximate cold wave type. For the three types described above, the value of $T_g$ is given by

| | | | |
|---|---|---|---|
| Cold wave type: | Rectangular | Triangular | Sinusoidal |
| Depth, $T_d$: | $DD(T_g)/t_o$ | $2DD(T_g)/t_o$ | $\pi DD(T_g)/2t_o$ |

If none of the above cold wave approximations seems to fit a given cold wave, the differential equation (8) can be solved on a day to day basis through the duration of the cold wave using the daily mean temperature as constant throughout the day. An even better approach would be to use the period between weather observations (3 hours at most weather stations in Iceland) as a basis for the calculations of the cold wave effect on the inside air temperature. Assuming a linear variation of temperature between observations, the solution to (8) is

$$T = T_1 \exp(-at) + b((T_{k1}-T_{ko})t/\Delta t + (T_{ko}-(T_{k1}-T_{ko})/a\Delta t)(1-\exp(-at))), \qquad (16)$$

where $T_1$ = inside air temperature at beginning of period,
$t$ = time measured from beginning of period,
$\Delta t$ = time period between observations,
$T_{ko}$ = outside air temperature at beginning of period,
$T_{k1}$ = outside air temperature at end of period.

If it is assumed that the outside air temperature stays constant, $T_{k1} = T_{ko}$, between observations, Equation (17) is reduced to the form

$$T = T_1 \exp(-at) + bT_{ko}(1-\exp(-at)). \qquad (17)$$

In this form it is easy to evaluate the effect of any given cold wave on the inside air temperature of a given building provided that the building parameters a and b are known. This subject will be treated in the next section.

### 3.3. Evaluation of building parameters

By far the most common method of house heating in Iceland is hot water heating using radiators in each room for transmitting the heat. In recent years these heating systems have been designed as 80/40,-15°C systems, which means that they are designed to maintain the desired room temperature, usually 20°C, at -15°C outside air temperature, with 80°C inflow water temperature to radiators and 40°C return flow temperature. This temperature drop through the radiators is somewhat larger than commonly used in hot water heating systems in other countries and has been brought about by the rapid development of geothermal district heating systems in the country. For maximum utilization of the geothermal fluid, the radiator surface area is enlarged which results in larger temperature drop of the water flowing through.

For hot water radiator heating systems operating at conditions other than design conditions the German standard, DIN 4703, specifies the following rule

$$Q = Q_o(\Delta T_m/\Delta T_{mo})^{4/3}, \tag{18}$$

where $Q_o$ and $\Delta T_{mo}$ denote the radiator heat load and the radiator mean temperature above the room temperature, respectively, at design conditions and Q and $\Delta T_m$ are the same quantities at some other conditions. The radiator mean temperature difference is determined by the equation

$$\Delta T_m = (T_f - T_b)/\ln((T_f - T_i)/(T_b - T_i)), \tag{19}$$

i.e. the logarithmic mean temperature difference, where

$T_f$ = inflow water temperature to radiators,
$T_b$ = return water temperature from radiators,
$T_i$ = room temperature.

If $(T_b - T_i)/(T_f - T_i) > 0.7$, the difference between the logarithmic mean temperature difference and the arithmetic mean temperature difference given by

$$\Delta T'_m = (T_f + T_b)/2 - T_i, \tag{20}$$

is insignificant and (20) may then be used instead of (19). Otherwise Equation (19) should be used for the evaluation of off-design conditions.

It follows from Equation (18) that the radiator heat transmission coefficients vary with the load, since

$$Q/Q_o = k_r \Delta T_m / k_{ro} \Delta T_{mo} = (\Delta T_m / \Delta T_{mo})^{4/3}, \qquad (21)$$

leads to

$$k_r / k_{ro} = (\Delta T_m / \Delta T_{mo})^{1/3}. \qquad (22)$$

When a district heating system is designed for a system design temperature different from the design temperature for the radiator heating system (usually -15°C in Iceland) the mean temperature difference will also be different from that at design conditions. This mean temperature difference is given by the equation

$$Q/Q_o = (T_i - T_g)/(T_i - T_{go}) = (\Delta T_m / \Delta T_{mo})^{4/3}, \qquad (23)$$

where

$T_g$ = system design temperature,

$T_{go}$ = radiator system outside air design temperature (= -15°C).

At the system design temperature, $T_g$, the heat balance for the building gives

$$k_r \Delta T_m = k_l (T_i - T_g), \qquad (24)$$

where $k_r$ and $k_l$ are previously defined. The value of the parameter b is then found to be

$$b = k_l/(k_l + k_r) = \Delta T_m / (\Delta T_m + T_i - T_g), \qquad (25)$$

i.e. b is independent of the building type. Assuming an 80/40, -15°C radiator system, room temperature of 20°C, and an inflow water temperature of 80°C, Equations (23) and (25) give the following values:

| | | | | | |
|---|---|---|---|---|---|
| $T_g$ | -6 | -8 | -10 | -12 | -15 |
| $T_b$ | 31.2 | 33.1 | 34.9 | 36.9 | 40.0 |
| $\Delta T_m$ | 29.1 | 30.8 | 32.4 | 34.0 | 36.4 |
| b | 0.528 | 0.524 | 0.519 | 0.515 | 0.510 |

For evaluation of the parameter a, the type of buildings to be considered should be specified. The percentage of windows and door openings of the gross exterior wall area is needed as well as the volume of interior partitions made of poured concrete (supporting walls) or lightweight aggregate and poured concrete floor volume. This is the type of houses most common in Iceland and in what follows other types will not be considered. For heat losses from buildings, the heat transfer coefficients specified by the Icelandic Standard, IST 66, will be used. This Standard specifies the following maximum values of heat transfer coefficients (k-values):

| | k-value $W/m^2 \cdot {}^oC$ | | k-value $W/m^2 \cdot {}^oC$ |
|---|---|---|---|
| Exterior walls | 0.55 | Windows | 3.2 |
| Roof or ceiling | 0.30 | Doors on ext.walls | 2.5 |
| Floor | 0.30 | Air change, 0.8/h | 0.29 $W/m^3 \cdot {}^oC$ |

According to the Standard the heat loss through the floor is evaluated by assuming the full difference between inside and outside air temperature through a 1.0 m wide area along the exterior wall whereas the heat loss through the remaining portion of the floor is based on a temperature underneath the floor equal to the annual mean (5.0°C in Reykjavik).

The mass of interior walls and floors is based on a mass density of $2.5 \cdot 10^3$ kg/m$^3$ for poured concrete and $1.5 \cdot 10^3$ kg/m$^3$ for light aggregate walls. The specific heat for both poured concrete and light aggregate is assumed to be c = 0.88 kJ/kg.°C.

In Iceland the size and type of buildings in which people live is quite variable although the buildings are almost exclusively made of concrete as discussed previously. In Reykjavik large blocks of apartment buildings, four to ten stories high, are quite common as are one, two and three story houses for one to four families. The same is true for some of the larger towns around the country, but in the smaller towns and villages it is noted that the one story, one family house is the most common. It is obvious that this type of building will exhibit a relatively greater heat loss than will a larger building and therefore it will be cooled down more during a cold wave than the larger buildings. For this reason the one story, one family house will be the only type of building considered in the following.

The Iceland State House Agency, which is responsible for the public financing of family housing in the country, has designed a number of one story, one family houses, which have been and are being built all around the country. Although these houses do not cover all types of houses being built, they provide a good, representative cross section of the Icelandic one family home. An average one story home has been constructed by using 17 different designs from the State Housing Agency. This average home has the characteristic values given in Table II from which the building parameter a is evaluated.

From Table II, assuming 20°C room temperature the following values are obtained for the average Icelandic one family home:

$$\begin{aligned} k_l &= 2.3522 + 2.9786/(20-T_g) \quad W/m^2\cdot{}^oC, \\ m &= \Sigma m_i = 397.28 \ kJ/m^2\cdot{}^oC, \qquad\qquad (26) \\ a &= 0.2175(k_r + k_l) = 0.2175 \ k_l/b \ days^{-1} \end{aligned}$$

With the house parameters established the cooling effects of the most severe cold waves can now be evaluated.

## 3.4. Evaluation of system design temperature

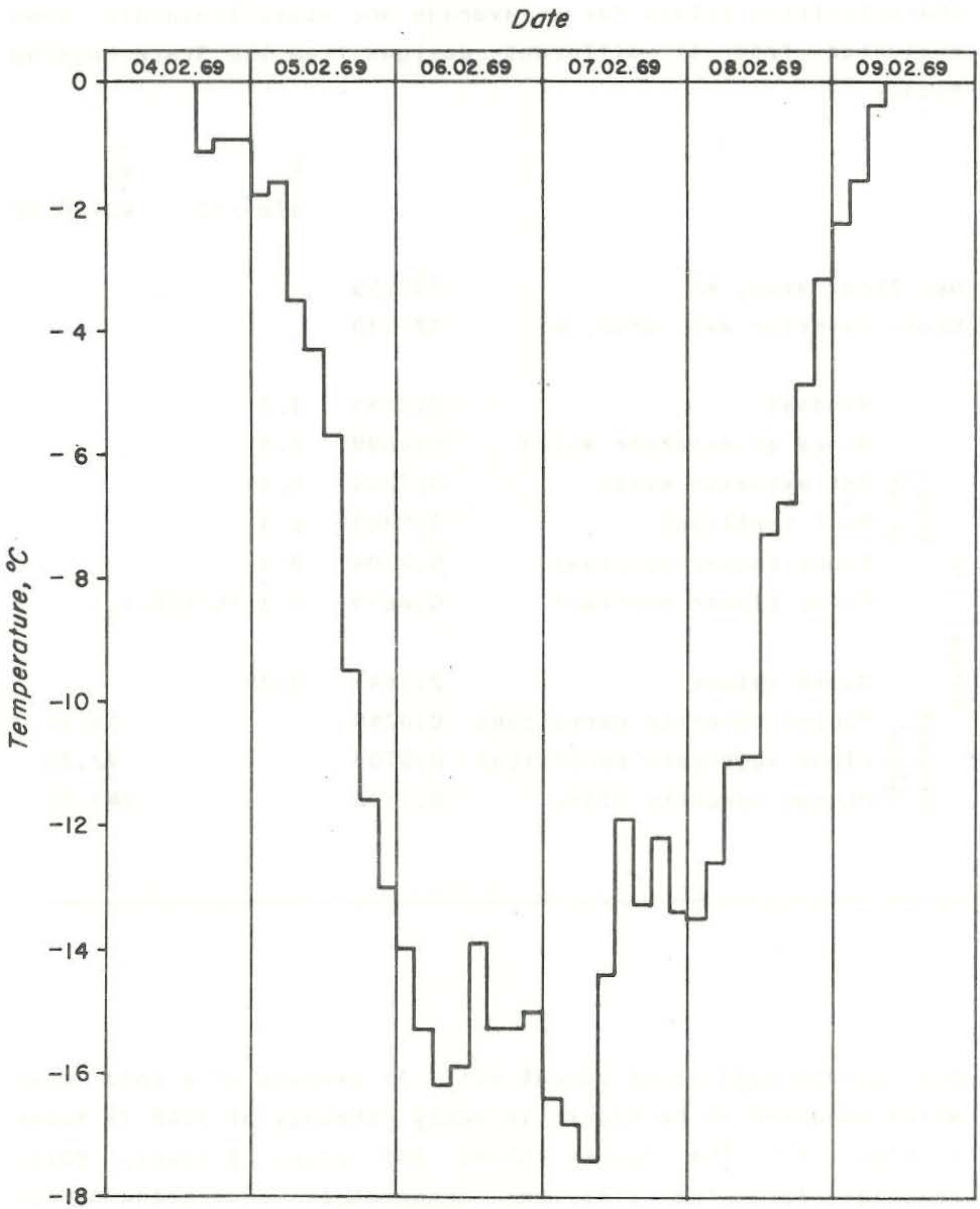
In order to evaluate the extent of cooling of buildings during cold waves the local weather records must be studied and the

## Table II

Characteristic values for an average one story Icelandic home evaluated from 17 different designs from the State Housing Agency.

| | | | $k_i$ $W/m^2 \cdot {}^oC$ | $m_i$ $kJ/m^2 \cdot {}^oC$ |
|---|---|---|---|---|
| | Net floor area, $m^2$ | 109.55 | | |
| | Gross exterior wall area, $m^2$ | 127.30 | | |
| Normalized Areas $m^2/m^2$ | Windows | 0.2185 | 3.2 | |
| | Doors on exterior walls | 0.0609 | 2.5 | |
| | Net exterior walls | 0.7206 | 0.55 | |
| | Roof (ceiling) | 0.9323 | 0.3 | |
| | Floor (outer section) | 0.2704 | 0.3 | |
| | Floor (inner section) | 0.6619 | $0.3 \cdot 15/(20-T_g)$ | |
| Normalized Volumes $m^3/m^2$ | House volume | 2.5641 | 0.29 | |
| | Poured concrete partitions | 0.0248 | | 54.56 |
| | Light aggregate partitions | 0.0703 | | 92.80 |
| | Poured concrete floor | 0.1136 | | 249.92 |

most severe cold waves sought out. An example of a cold wave which occurred in Reykjavik in early February of 1969 is shown in Figure 5. The figure shows the eight 3-hourly daily readings from the time the temperature went below 0°C on February 4 until it reached 0°C again on February 9. This is probably the most severe cold wave to occur in Reykjavik in recent years.

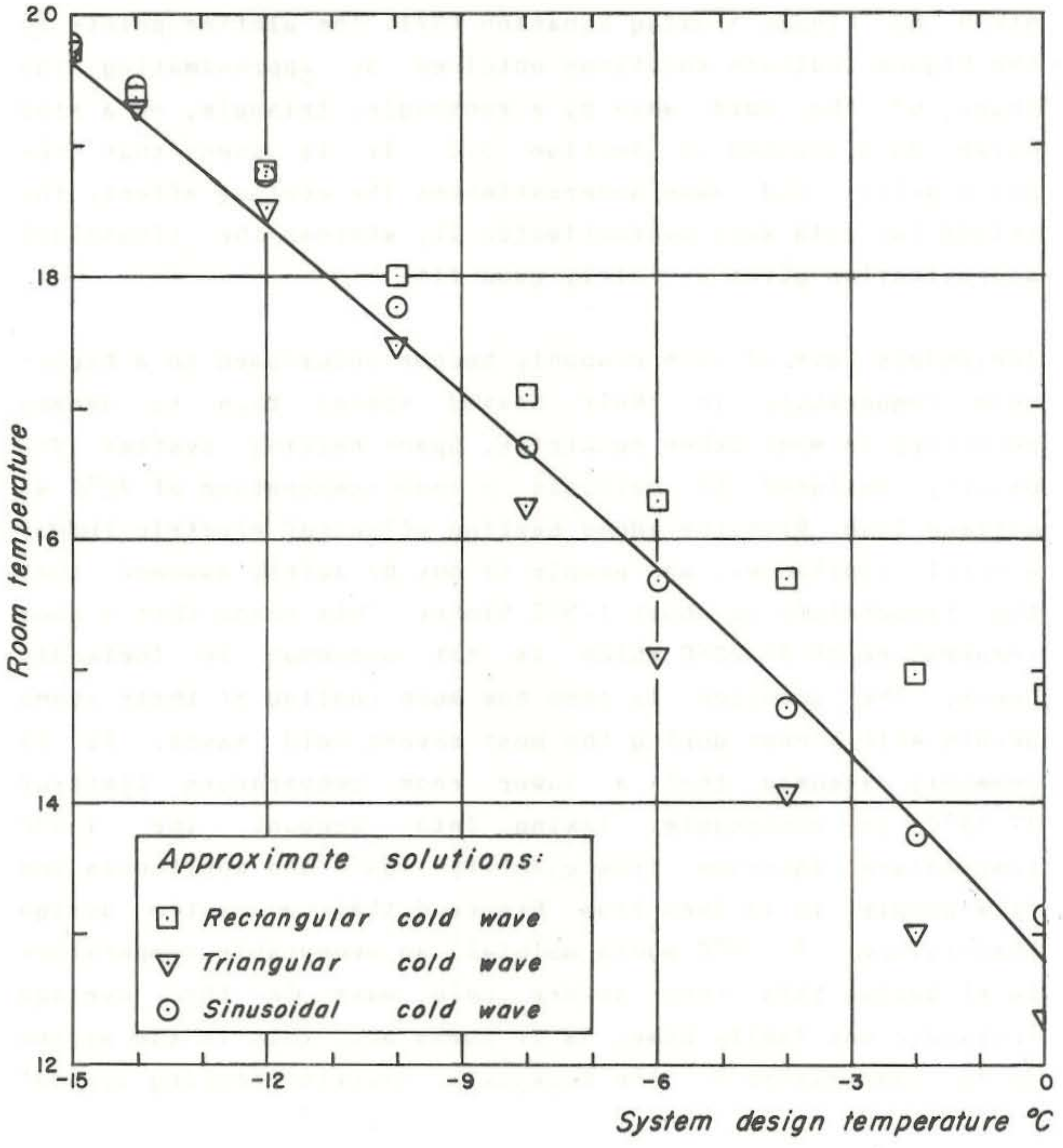



Observed temperatures in Reykjavik during a severe cold wave

Fig. 5

The results of the calculations of the cooling effect of this cold wave on the average Icelandic one family home are shown in Figure 6. The solid curve gives the step by step solution through the 3-hourly temperature readings of the cold wave as given by Figure 5 using Equation (17). The plotted points on the Figure indicate solutions obtained by approximating the shape of the cold wave by a rectangle, triangle, or a sine curve, as discussed in section 3.2. It is seen that the rectangular cold wave underestimates the cooling effect, the triangular cold wave overestimates it, whereas the sinusoidal approximation gives a fairly good fit.

Icelanders have of late probably become accustomed to a higher room temperature in their heated spaces than is deemed necessary in most other countries. Space heating systems are usually designed to maintain a room temperature of 20°C at maximum load. With the added heating effects of electric lights, electric appliances, and people it may be safely assumed that the temperature is about 1.5°C higher. This means that a room temperature of 21-22°C which is not uncommon in Icelandic homes. The question is then how much cooling of their rooms people will accept during the most severe cold waves. It is commonly assumed that a lower room temperature limit of 17-18°C is acceptable. Taking into account the 1.5°C temperature increase from electric lights and appliances and from people, it is seen from Figure 6 that a system design temperature of -8°C would maintain an acceptable temperature level during this very severe cold wave in the average Icelandic one family home. As it turns out, this is the system design temperature for the Reykjavík district heating system.




Minimum inside temperature during cold wave of Figure 5

Fig. 6

## 4. THE DISTRICT HEATING SYSTEM POWER DEMAND

In chapter 2 it is outlined how the heating needs of a district heating system may be estimated. One method, discussed in section 2.1, is based on experience gained from operation of the Reykjavik district heating system, where the heat requirements are based on the volume of buildings of different types. Included in the figures used in this method are 10% heat losses in the distribution network as well as the requirements for hot water tap use.

With the other method, discussed in section 2.2, the district heat demand is based upon the annual fuel consumption for heating in the district. This method gives an estimate of the average annual heating needs of buildings and does not include several losses which are inherent in district heating systems as well as other factors which influence the heating requirements. These factors will be discussed briefly in the following sections.

### 4.1 Correction factors due to heat losses and other effects

4.1.1. Heat losses in building pipe network. It has been estimated that 5-10% of the heat energy supplied to ordinary hot water central heating systems is lost. These losses have been accounted for in the utilization factors or heating efficiency factors discussed in section 2.2. and will therefore have to be added when the district heating system power is being evaluated.

4.1.2. Heat losses in the distribution network. Several measurements of heat losses in the Reykjavik district heating system distribution network have been made, both in street branches and in the house connection pipes, as reported by Zoega (1964) and Jonsson and Theodorsson (1969). These measurements indicate heat losses of about 45 W/m in an ordinary street branch (40-100 mm OD) with double pipe (supply and return) but considerably smaller heat loss in house connection pipes or about 20 W/m. On basis of these measurements Zoega (1964) has estimated heat losses in the distribution system of about 6.6% of the total annual heat energy use and about 4.4% of the heat energy use at -6°C outside air temperature. Experience from Denmark (Olufsen 1965) indicates distribution network losses of 7-10% of the total heat demand. In Iceland these losses are generally estimated at 5% when the district heating power demand is being estimated.

4.1.3. Effects of solar radiation. Studies of the effects of solar radiation on the heating requirements of buildings are very limited. It is estimated that the solar radiation in spring, summer, and fall provide for approximately 10% of the annual heating needs. For this reason the heating system power demand is increased by 10% in order to account for this effect.

4.1.4. Increased heat demand with district heating. Experience from the operation of geothermal heating systems in Iceland indicates that people´s heating habits change with the advent of district heating in such a way that the heat demand may be increased by 10 to 15%. This phenomenon may no longer be true in the more recent district heating systems in Iceland where the heating cost is not so favourable in comparison with fuel heating cost as was the case with the older district heating systems such as in Reykjavik. In the following, however, the increased heat demand will be assumed.

4.1.5. Increased heat demand due to strong winds. Strong winds will increase the heat demand of buildings. This effect should be accounted for when estimating the district heating system power. It is estimated that a power increase of 10% will take care of this extra load on the heating system.

4.1.6 Overall correction factor. When all the effects of sections 4.1.1 through 4.1.5. are combined the overall correction factor is found to be

$$\alpha \approx 1.05 \cdot 1.05 \cdot 1.1 \cdot 1.12 \cdot 1.1 \approx 1.5. \quad (27)$$

## 4.2. Hot tap water consumption

In homes where fossil fuel is used for heating it has been estimated that the annual hot tap water consumption amounts to 10-15 metric tons/person of hot water at 80°C. The Reykjavik district heating system experience indicates a considerable increase in hot water use when the district heating system is connected. From measurements of this use the annual hot tap water consumption in geothermal heating district is estimated at 30 metric tons/person.

The use of hot tap water is for the most part independent of the weather so that the heat energy for its heating must be separated from the space heating energy. Assuming heating of the tap water from 4°C to 80°C in homes heated with fossil fuel and an annual consumption of 10 metric tons/person, the annual energy consumption for hot tap water is

$$G_t = 10 \cdot 10^3 (h_{80} - h_4) = 3.18 \cdot 10^6 \text{ kJ/person} \cdot \text{year}, \quad (28)$$

where

$h_{80}$ = enthalpy of water at 80°C = 334.9 kJ/kg,
$h_4$ = enthalpy of water at 4°C = 16.8 kJ/kg.

The above enthalpy values are given in U.K. Steam Tables in SI Units (1970).

## 4.3. Total district heating system power demand

In section 2.2 it was described how the annual heating energy demand of a district can be obtained by studying the annual heating fuel consumption of a limited but representative sample of buildings in the district. With the known population of the district the annual heating energy per person in the district is obtained, $G_h$ kJ/person·year. Subtracting the annual energy consumption for hot tap water (28) the annual space heating energy consumption is obtained

$$G = G_h - G_t \quad \text{kJ/person·year} \qquad (29)$$

With the known building space per person in the district, V $m^3$/person, the space heating energy consumption is obtained on the basis of building volume

$$F_h = G/V = (G_h - G_t)/V \quad \text{kJ/m}^3\text{·year.} \qquad (30)$$

With the correction coefficient found in section 4.1.6. the space heating energy to be supplied by a district heating system is

$$F_d = \alpha \cdot F_h = (G_h - G_t)\alpha/V \quad \text{kJ/m}^3\text{·year.} \qquad (31)$$

If the population of the district is P, the annual mean temperature of the district $T_m$, the system design temperature as determined by section 3, the maximum daily average thermal power demand for the district system is

$$H' = G \cdot P(T_i - T_g)/(8760(T_i - T_m) \cdot 3600) \quad \text{kW.} \qquad (32)$$

As discussed in section 2 the peak demand is somewhat higher than the maximum average daily demand. In the Reykjavik district heating system the peak load is found to be on the order of 30% over the daily average load, but in other district heating systems the peak load is somewhat lower as previously discussed. Calling this peak load factor $\beta$ (= 1.3 for Reykjavik) the peak power which the district heating system must supply is

$$H = \beta \cdot H' \quad \text{kW} \qquad (33)$$

This power demand forms the basis fór the design of the district heating system. If geothermal fluid is available which might be used as a source for supplying this power, the quantity needed depends on the fluid temperature. The relationship between the water temperature and the necessary flow of water for the heating system will be treated in the next chapter.

# 5. DETERMINATION OF DISTRICT HEATING WATER SUPPLY

## 5.1. Radiator heat emission

The heat emission by hot water radiators is usually expressed by the equation

$$Q = k\,A\Delta T_m , \tag{34}$$

where k is the radiator heat transmission coefficient, A is the radiator surface area, and $\Delta T_m$ is the mean difference between the radiator water temperature and the room air temperature. The radiators may be regarded as counterflow heat exchangers for which the mean temperature difference is given by the equation

$$\Delta T_m = (T_f - T_b)/\ln((T_f - T_i)/(T_b - T_i)), \tag{35}$$

which is the same equation as discussed previously (Equation (19), see chapter 3.). If the radiator system is designed for an outside air temperature of $T_{go}$ the heat emission by the radiators is in equilibrium with the heat loss from the building to the environment:

$$Q_o = k_o A_o \Delta T_{mo} = K_{lo}(T_{io} - T_{go}) = \dot{m}_o c_w (T_{fo} - T_{bo}), \tag{36}$$

where the subscript o refers to design conditions. The quantities not previously defined are the following

$K_{lo}$ = overall building heat transfer coefficient, kW/°C,
$\dot{m}_o$ = hot water mass flow, kg/s,
$c_w$ = specific heat of water, kJ/kg·°C, assumed constant.

If conditions are different from design conditions the heat balance equations (36) become

$$Q = kA\Delta T_m = K_l(T_i - T_g) = \dot{m}c_w(T_f - T_b). \tag{37}$$

In most cases several quantities in Equation (37) are unchanged from (36) such as

$A$: Radiator surface area. In many cases, however, it may be considered advantageous to increase the radiator surface area.

$K_l$: Building heat transmission coefficient. It is not likely that this coefficient will change although it is always possible to improve the building insulation and reduce its value.

$T_i$: Room temperature. In Iceland this is usually 20°C and its value is not likely to change.

$c_w$: Specific heat of water. This is taken to be constant.

According to the German Standard, DIN 4703, there is a relationship between the load on a hot water radiator system, $Q$, and the mean temperature difference, $\Delta T_m$. With other parameters such as radiator surface area unchanged this relationship is given by the expression

$$Q/Q_o = (\Delta T/\Delta T_{mo})^{4/3}, \qquad (38)$$

which leads to the following equation for the radiator heat transmission coefficient

$$k/k_o = (\Delta T_m/\Delta T_{mo})^{1/3}. \qquad (39)$$

Combining Equations (35) through (39) leads finally to the following system of equations for a district heating system:

$$\begin{aligned} Q/Q_o &= kA\Delta T_m/k_oA_o\Delta T_{mo} = (A/A_o)(\Delta T_m/\Delta T_{mo})^{4/3} \\ &= K_l(T_i-T_g)/K_{lo}(T_i-T_{go}) \\ &= \dot{m}(T_f-T_b)/\dot{m}_o(T_{fo}-T_{bo}). \qquad (40) \end{aligned}$$

With these equations the various parameters needed for the design of a district heating system can be evaluated.

## 5.2 Water requirement of district heating system.

Equations (40) are now used to determine the rate of flow of water needed for the district heating system. If it is assumed that the building radiator heating systems are designed as 80/40,-15°C systems the equations to be solved are

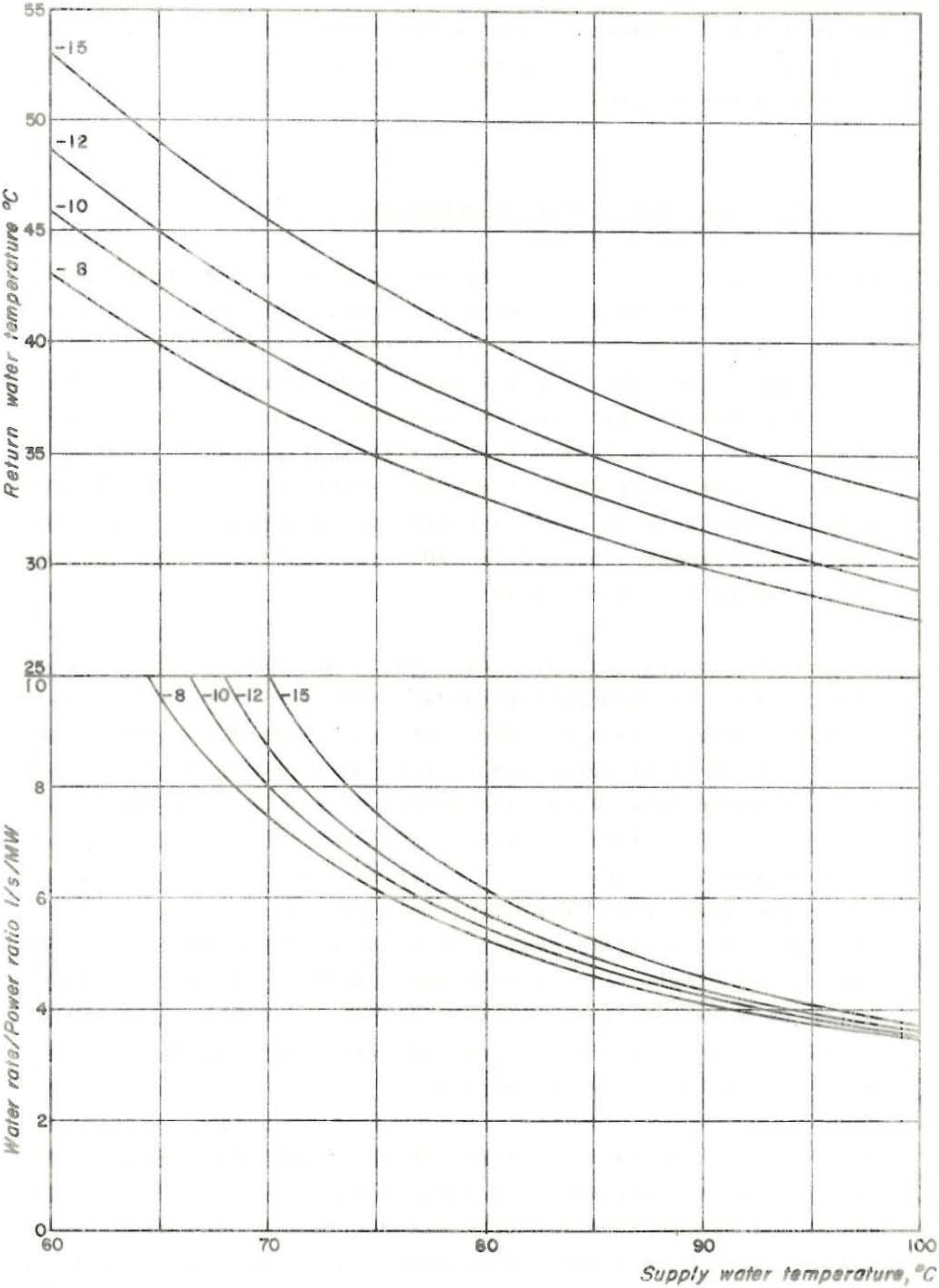
$$(\Delta T_m/36.41)^{4/3} = (T_i-T_g)/35 = \dot{m}(T_f-T_b)/40\dot{m}_o, \quad (41)$$

where a room temperature of 20°C is assumed. The mass rate of water flow will depend on the supply temperature of the water, $T_f$, and the necessary system design temperature, $T_g$. The solutions are shown in Figure 7. The upper part of the figure shows the return water temperature as a function of the supply water temperature for a few values of the system design temperature. The lower part of the figure shows the rate of hot water flow in litres/sec required per megawatt heating output power. With the heating power requirement established in the manner described in the last section, the corresponding hot water requirement is easily obtained from Figure 7.

As previously discussed the annual hot tap water requirement is estimated at 30 metric tons/person. This requirement is independent of seasons and its use is assumed to be fairly evenly distributed over a daily period of 10 hours. It is also assumed that this use is not much affected by the water supply temperature. The hot tap water use is therefore estimated as

$$q_n = 30 \cdot 10^4/365 \cdot 3600 = 0.23 \text{ l/s} \cdot 100 \text{ persons.} \quad (42)$$

This quantity must be added to the hot water requirement for heating obtained from (41) or Figure 7. Here it is assumed that the hot water may be used directly for tap water which is common for Icelandic geothermal district heating systems.




Return water temperature and rate of water flow required

Fig. 7

Occasionally, however, the water chemistry prohibits its direct use for taps, in which case heat exchangers are used for tap water heating.

## 5.3. High geothermal fluid temperature

It is clear from Figure 7 that the rate of water requirement for a district heating system is rapidly reduced as the water temperature increases. It is, however, not considered desirable that the supply water temperature be higher than 90°C and 80°C seems to be considered an optimum supply temperature. The reason for this is that the supply water is commonly used directly for hot water taps, and higher temperature than 80°C may actually be dangerous for this use. For water temperature over 90°C therefore, some special arrangements have to be made.

Hot water temperatures well over 100°C are encountered in at least two low temperature areas which are used for district heating. These areas are the hot water area in Reykjavik where an average water temperature of 128°C is produced, and the hot water area of Seltjarnarnes, a town just outside the Reykjavik city limits, where the water temperature from drilled wells is over 115°C. In spite of the proximity of these two geothermal reservoirs there is no connection between the two. The Reykjavik water is produced from water bearing layers of less than 1200 m depth whereas the Seltjarnarnes water is produced from a depth of close to 2000 m. The water chemistry of the two layers is also considerably different pointing to the different origin of the two.

When high temperatures like those discussed above are encountered the distribution network which ordinarily consists only of a single supply pipe, is made double in part. The return water from the double distribution system is used for cooling down the hot water making the supply water of suitable temperature. A temperature of 80°C at buildings is commonly chosen as supply water temperature. This arrangement is shown schematically in Figure 8.

If it is assumed that radiator systems are 80/40,-15°C systems, temperature drop in supply lines from pumping station to consumers is 3°C and in return lines from consumers to pumping station is 1°C, and that 10% of the supply water to the double distribution network is consumed as hot tap water, the size of the double distribution network as a fraction of the total network, for 83°C supply water leaving the pumping station, must be

$$x = (T_{o}-83)/0.9(T_{o}-T_{b}+1). \qquad (43)$$

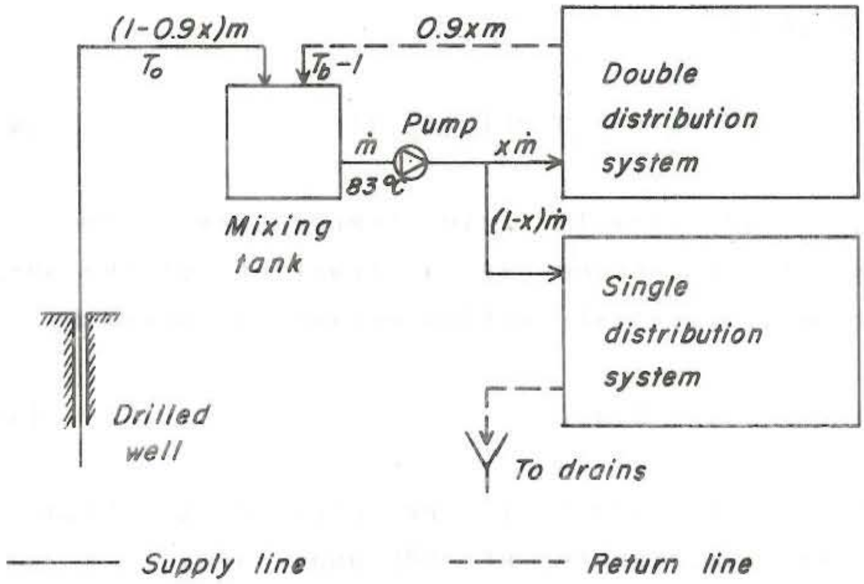
where $T_o$ = geothermal fluid temperature. The rate of geothermal fluid needed as a fraction of the 80°C supply needed for a single distribution network is given by

$$\kappa = 1 - 0.9x. \qquad (44)$$

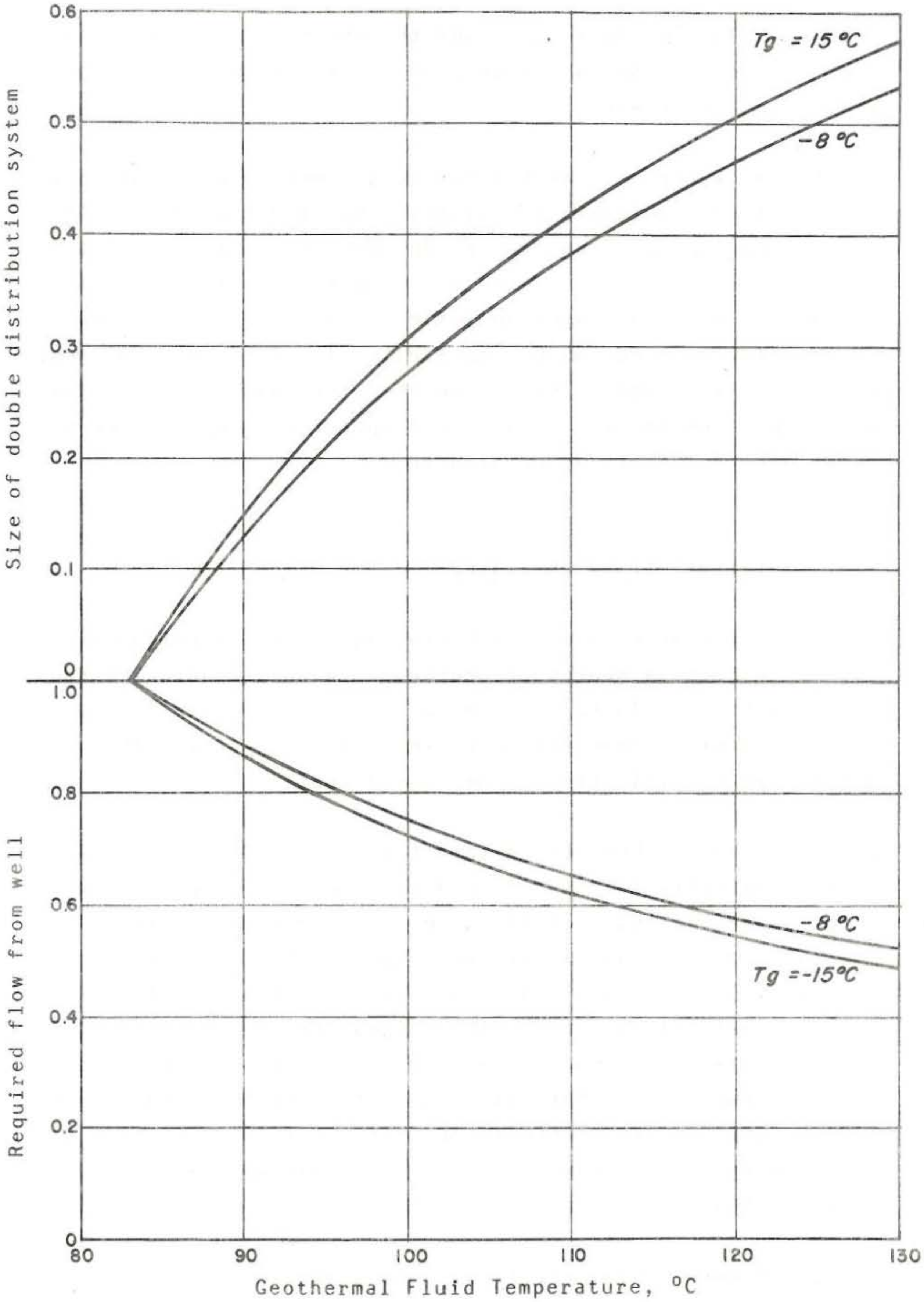
The results of (43) and (44) are plotted in Figure 9 for system design temperatures of -8°C and -15°C.

## 5.4. Low geothermal fluid temperature

From Figure 7 it is clear that the needed volume rate of water flow for heating increases rapidly as the temperature of the supply water is reduced. In retrofitted buildings not designed to handle the added rate of flow, this may pose problems such as excessive noise in pipes and too much friction in the radiator systems to move the necessary volume of water through. In Iceland there are several geothermal district heating systems where the temperature of the water is as low as 60°C. The general method of dealing with the problems of excessive fluid flow in these systems is to design the house heating systems with oversize radiators as compared with the customary 80/40,-15°C systems commonly used in the




Schematic line diagram for a district heating system with overheated water




Required size of double distribution network and relative rate of flow

Fig. 9

country. As seen from Eq. (40) the oversize radiators will increase the temperature drop of the water thus reducing the needed volume of water.

Geothermal water at temperatures much under 60 °C is generally not considered suitable for district heating systems except by further heating of the water during the coldest weather. Actually such water could be very well suited for radiant panel heating or air heating systems. Such systems, however, have never gained much popularity in Iceland, and whole districts with radiant panel heating or air heating which would be needed for the low temperature supply water have never been contemplated or planned.

## 5.5. Geothermal fluid chemistry

If a town for which a district heating system is being planned has access to a source of geothermal water, a detailed study of the water chemistry is required in order to determine various design features for the heating system. There are mainly two possibilities to be considered:

(a) The water chemistry allows the use of the geothermal fluid directly for the distribution network as well as for the house heating systems. The distribution network is then designed as a single pipe system if the water temperature is not too high (see section 5.3). The geothermal fluid is then most often also used directly as hot tap water. If people object to the direct use of the water for their hot taps they are free to install heat exchangers to heat fresh water for their hot taps. A schematic line drawing of this arrangement is shown in Figure 10a.

(b) The water chemistry is of such a nature that it is not considered advisable to use it directly for the system distribution network. The reason for this may be due to corrosive effects of the water or due to danger of pipe clogging caused by excessive deposits of minerals dissolved in the geothermal fluid. For this reason the distribution network must be designed as a double closed pipe system containing fresh water which is heated in a heat exchanger station by the geothermal fluid. The hot tap water is then heated in heat exchangers in each house. A schematic line drawing of this arrangement is shown in Figure 10b. The geothermal fluid temperature has a strong influence on the cost of pumping stations and heat exchanger stations.

Geothermal district heating systems in Iceland are generally designed so that the geothermal fluid is used directly for distribution of the heat energy. Double distribution systems are so expensive to construct that their use has been avoided whenever possible and other measures attempted to minimize the risks involved in using the geothermal fluid directly such as chemical treatment of the water.

The main components of geothermal fluids acting as corrosive agents are oxygen and sulfides. These components are not to any appreciable extent found simultaneously in the fluid since they tend to eliminate each other (Hermannsson et al. 1975). Oxigen is probably the most corrosive agent in Icelandic geothermal water and must be removed if at all possible in order to minimize the risk of corrosion.

Other components dissolved in the geothermal water may act as inhibitors, slowing down the corrosive effects of oxygen and the sulfides, but they may also in some instances increase these effects. Dissolved silica may be deposited at spots where corrosion has started and it is generally regarded as an inhibitor to corrosion. If, however, the silica content of the water is not sufficient for the formation of a completely protective layer of deposits its presence can actually increase the danger of pitting corrosion. Chlorides in the water will speed up corrosion by the increased conductivity of the water due to their presence. It may also be mentioned here that the pH value influences the corrosive effect of the water in such a way that the higher the pH value the less corrosive the water will be, other things being equal. This fact has been used with good results in the chemical treatment of geothermal fluid for the purpose of reducing the risk of corrosion.

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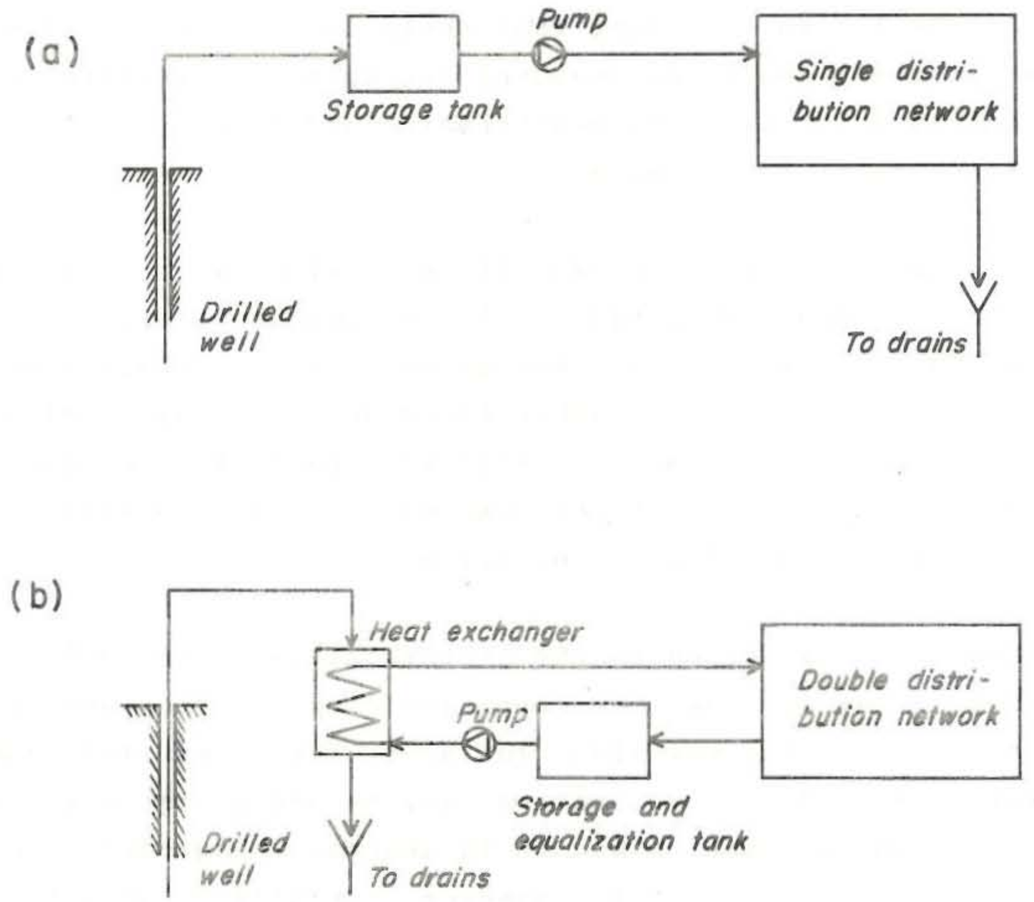



Schematic line diagrams for geothermal district heating systems

Fig 10

# 6. PIPELINES AND PIPING NETWORK

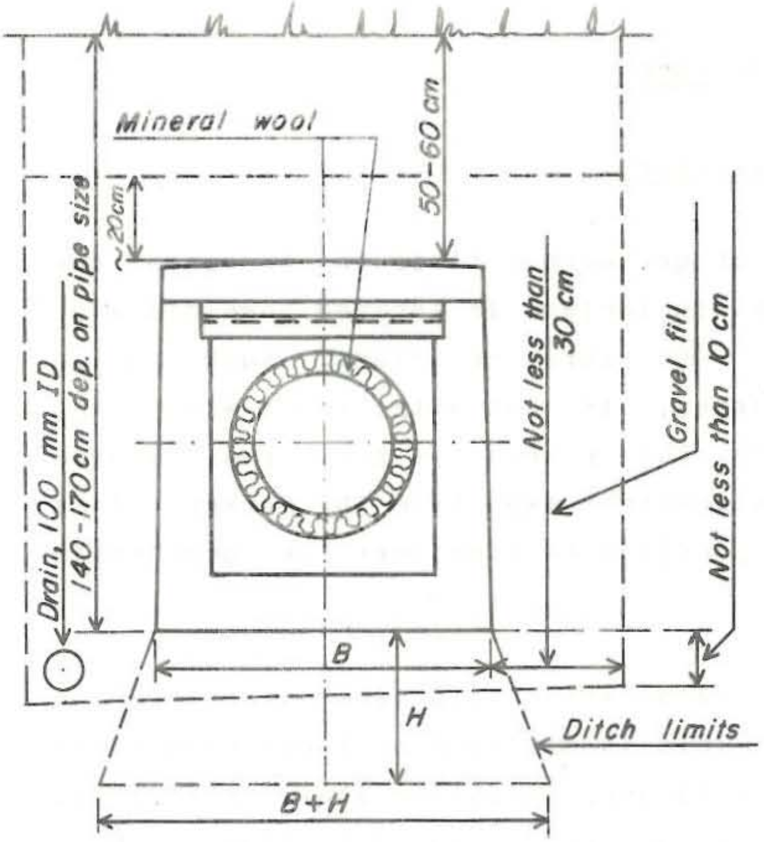
## 6.1. Main transmission pipeline

In most cases the source of geothermal fluid to be used for district heating systems in Iceland is located some distance away from the consumers. On rare occasions, such as in Reykjavik and Seltjarnarnes, is hot water found within the district to be heated. In other places the geothermal source is located up to 60 kilometres away from the market. This calls for a transmission pipeline to transport the geothermal fluid.

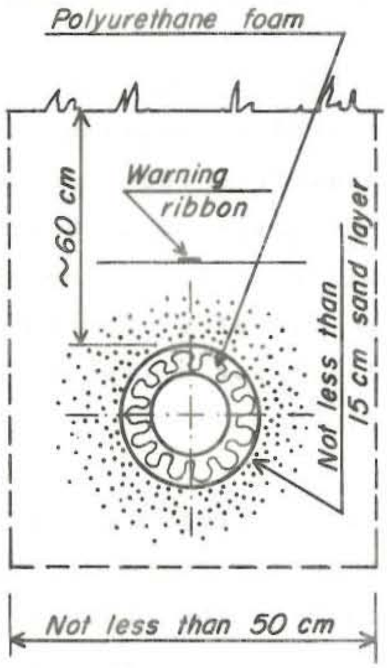
For determination of the transmission pipeline diameter, no general rules do exist. This is dictated by local conditions such as the grade of the land over which the pipeline is laid, available power for pumping, etc. An approximate rule of thumb is to design the diameter of the pipe so that the pressure drop in a straight section of pipe at maximum rate of flow is on the order of 0.5-1.0 bar/km.

In Iceland transmission pipelines for geothermal heating systems have been of various makes and types with great difference in cost and durability between the various types. The pipeline material is either steel or asbestos-cement. Asbestos-cement pipe is usually insulated only by an earth and grass cover, but in recent years some experiments have been made with partial and full insulation of these lines. A short description of the various types of pipelines follows.

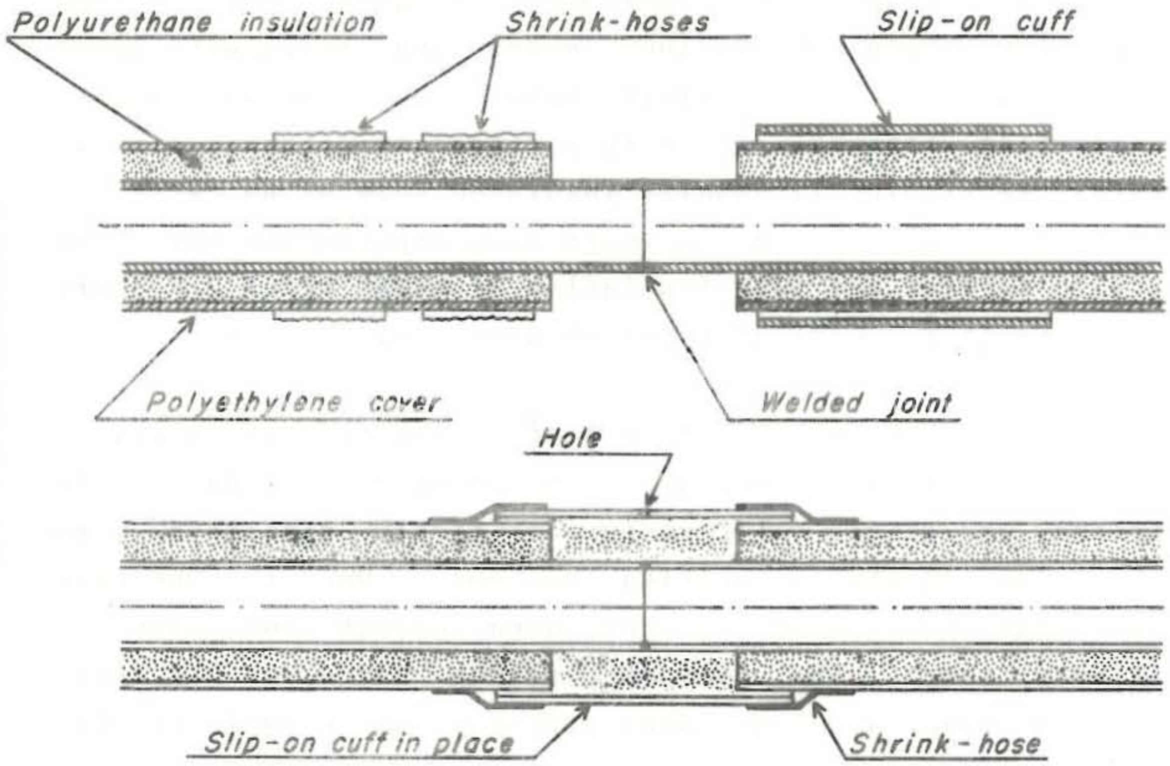
6.1.1. Steel pipe in a concrete duct. This type of pipeline is the most expensive and also the one with the longest life expectancy and lowest maintenance cost of all pipelines used in Icelandic district heating systems. Due to the high initial cost this type is not used unless the line is relatively short, the consumer market large or if local conditions demand an underground pipeline. An example of this type of pipeline is shown in Figure 11a.




a. Steel pipe in a concrete duct




b. Steel pipe in a polyethylene cover-cross section




c. Steel pipe in a polyethylene cover-longitudional section through a pipe joint

Fig. 11

The major transmission pipeline in Iceland of the concrete conduit type is the pipeline from Reykir to Reykjavik, approximately 16 kilometres long, where two 14" OD(350 mm) and one 28" OD(700mm) steel pipes are laid in above ground concrete channels. In the Reykjavik hot water distribution system as well as in several other district heating distribution systems the large diameter underground mains (> 8" or 200 mm nom diam.) are generally of this type.

6.1.2. Steel pipe with a protective polyethylene cover. This type of pipe is almost exclusively used in distribution networks with the exception of the largest diameter pipe (>200 mm OD). Until recently these pre-insulated pipes were not available in Iceland in sizes over 200 mm but now the larger sizes have become available and it may be expected that these will in the future replace the concrete conduit pipe in distribution networks. This pipe has been used to some extent in relatively small capacity transmission lines. Such lines may be constructed either above ground or underground. An example of the underground line is seen in Figure 11b.

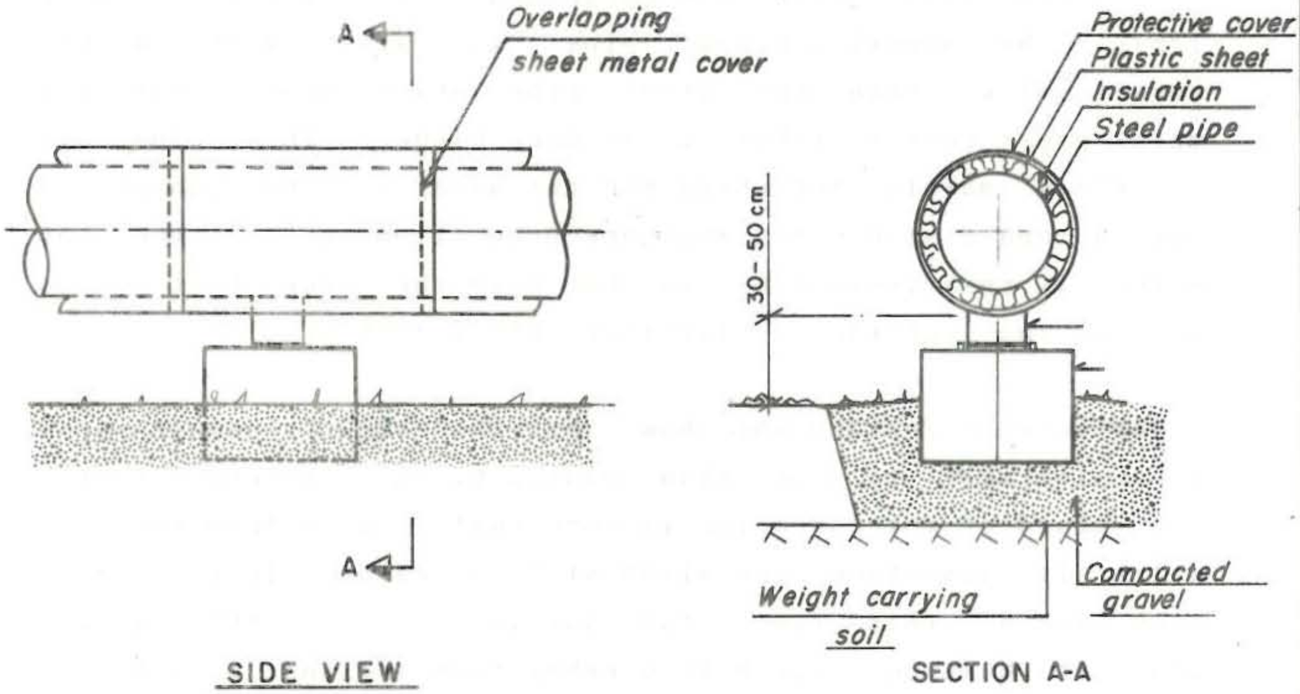
The polyethylene-covered pipe is pre-insulated with polyurethane foam. This means that the pipe is factory insulated in lengths of approximately 5 metres. Since the pipe sections are welded together in the field the factory made insulation does not cover the ends so the steel pipe is therefore somewhat longer than the protective polyethylene cover as shown in Figure 11c. When two pipe ends have been welded together and tested the bare pipe ends are covered by a slip-over cuff as shown in Figure 11c. Two different liquids, component A and component B, are then poured simultaneously through a hole in the slipover cuff. These two components are Polyol, a polyhydroxide compound, and MDI, an isocyanate compound mixed with a foaming agent such as Freon (Björnsson, G.K. 1981). When the two compounds are mixed together a polyurethane foam insulation fills the empty space. The hole is then closed with a plug and wrapped with a waterproof tape, the ends of the slipover cuff are covered by a shrinkhose, which shrinks and makes a watertight joint by heating, and the pipe joint is insulated and waterproof. These pipes are somewhat less expensive than the concrete conduit type and easier and quicker to lay and install. They are, however, more vulnerable to external damage, and directional changes create problems.

6.1.3. Above ground pipeline with a sheet metal protective cover. These pipelines are well suited for areas where the line can be placed above ground. The pipe is resting on supports every 5 to 10 metres depending on the size and strength of the pipe. These supports are made of concrete blocks resting on a bed of sand as shown in Figure 12a. If the ground carrying capacity underneath the supports is insufficient it may be necessary to undertake a soil change in these places. The blocks are generally lined up in such a way that the pipe will be straight over a distance of 100 m.

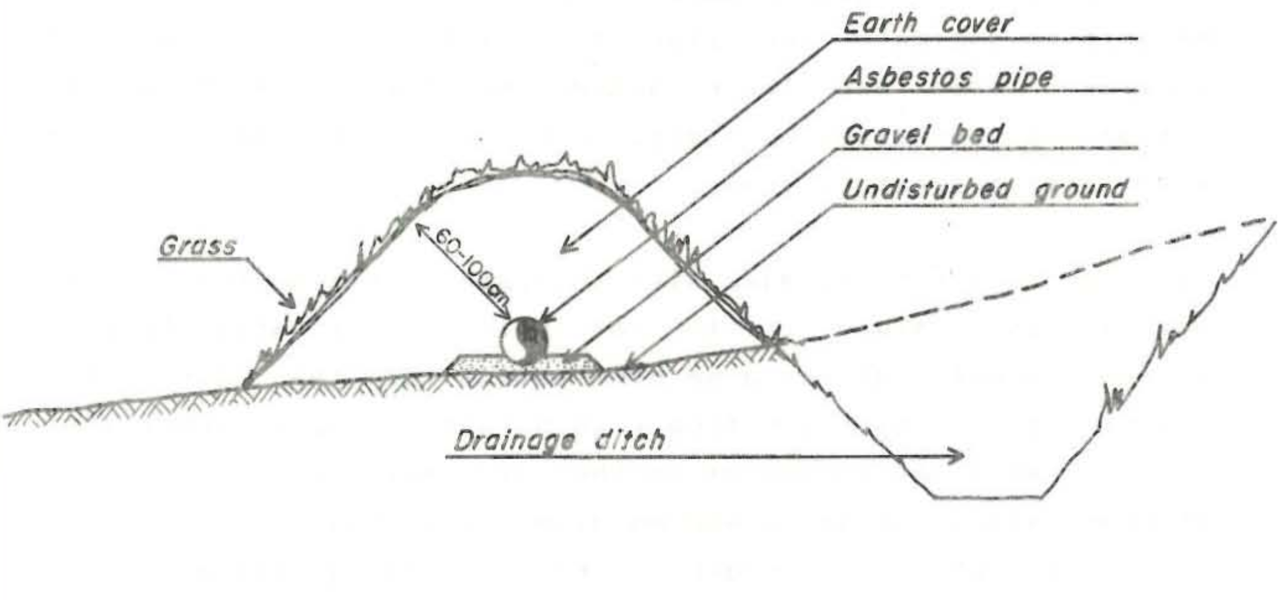
The pipe rests on special steel supports fixed to the concrete blocks. These supports allow the pipe to move within limits in an axial direction due to thermal expansion. Special arrangements have to be made to meet these thermal expansions which will be discussed later. In order to distribute the thermal expansion evenly the pipe is anchored approximately every 100 metres.

When the pipe has been welded together and all supports properly aligned it is insulated with mineral wool and finally covered by a protective sheet metal cover, either aluminium or galvanized steel.

6.1.4. Asbestos-cement pipe with an earth and grass cover. Asbestos-cement pipe is commonly used for geothermal transmission lines to district heating systems in Iceland. These pipes are attractive because of low initial cost compared with steel pipe. Their disadvantage is, however, much greater heat losses than from the insulated steel pipe, since the asbestos-cement pipe is generally used with no




a. Pipe line with a sheet metal cover




b. Asbestos cement pipe with earth and grass cover

Fig. 12

insulation other than the earth and grass cover (see Figure 12b). The asbestos-cement pipe is also much weaker structurally than the steel pipe which means that the maintenance cost is likely to be much higher. These pipes are therefore mainly used where the hot water must be transported over a long distance and when the production cost of the hot water is relatively low. In that case the larger heat losses are not too important, (Fjarhitun, Ltd., 1975).

Experience in Iceland has shown that the asbestos-cement pipe is vulnerable to corrosion caused by the geothermal fluid. The nature of the corrosion is such that calcium from the pipe walls is dissolved and mixed with the water. In the oldest pipe line of this type, the supply line to the Husavik district heating system in northeastern Iceland, by now over 12 years old, the phenomenon was detected early and the corrosion process has been followed closely by the Technological Institute of Iceland. It appears that the speed of corrosion is greatest at the beginning and is reduced with time. The corrosion process, however, is not fully understood, and more complete studies are needed in order to be able to estimate the pipe life time in this kind of service. From experience gained so far, it seems safe to assume that the asbestos-cement pipe in geothermal service will last at least 20 years.

Experiments are being made with insulated asbestos-cement pipe but at this stage it is not quite clear what type of insulating material would be best suited for the pipe. The trouble is that moisture from within (water vapour) penetrates the pipe wall and condenses on the outer surface. The amount of penetration through a 400 mm diam. pipe has been estimated at 2.5 kg/h per 100m length. The insulating material for asbestos-cement pipe, therefore must be sufficiently porous to stay dry with the moisture transfer from within, or a moisture proof layer must be provided between the pipe surface and the insulating layer with provisions for removing the moisture. It has been estimated (Björnsson, O.B., 1981) that mineral wool insulation covering only the upper 7/10 of the pipe surface and with the earth and grass cover would reduce the heat loss from the pipe by almost 50%.

## 6.2. Distribution network

Distribution networks in Icelandic district heating systems are all made of steel pipe, either in a concrete conduit or in a polyethylene protective cover as described in sections 6.1.1. and 6.1.2. This part of the district heating system is the most important part and it is of great importance for the successful operation and proper functioning of the heating system that the distribution network be designed with care and good planning.

The distribution network may in general be divided into mains or trunk lines, branch lines and house lines. The trunk lines or mains constitute as implied by their name the main lines going out from a pumping station. There may be only one trunk line or more depending on local conditions and the size of the network. The usual arrangement is to lay the trunk lines in such a fashion that the branch lines extend out from the trunk lines similar to branches on a tree with the line diameter decreasing until it ends. There are also lines forming closed loops which are considered to have some advantages over the other type. Both types are seen in Figure 13, showing a part of the Akureyri district heating system (Verkfraedistofa Nordurlands, Ltd., and Verkfraedistofa Sigurdar Thoroddsen, Ltd., 1976). The advantages of the closed loop type are for example less danger of closing of complete sections of town in case of repairs and reduced likelihood of low water pressure for those consumers located furthest away from the pumping station. The closed loop systems are naturally more expensive to construct than the simple branch systems, which probably is the reason why they are not more frequently used.

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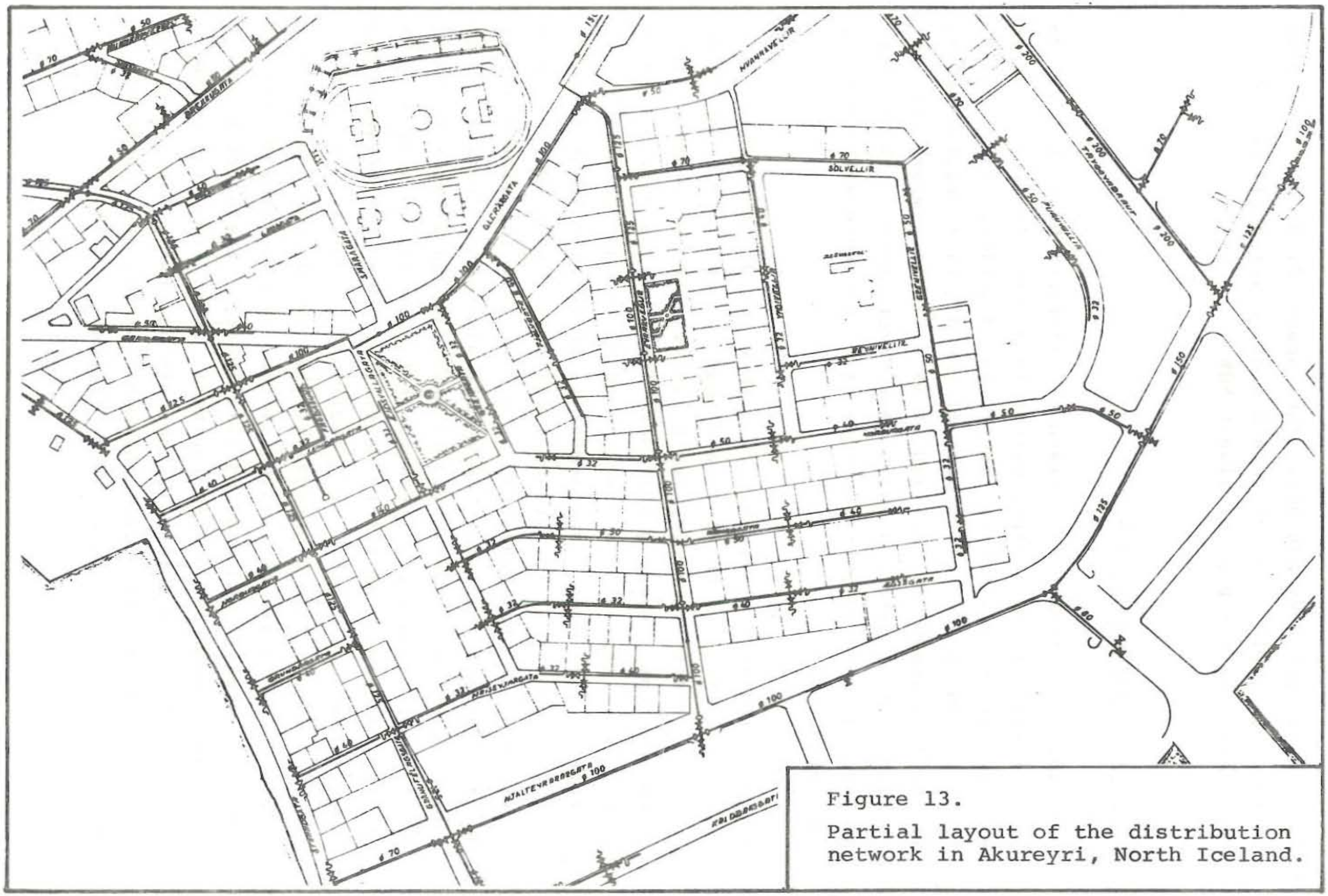


Figure 13.

Partial layout of the distribution network in Akureyri, North Iceland.

Fig. 13

Reinforced concrete service wells are located at all branch points other than a few house lines, for easy service and repair access. All the larger wells are built and poured on location but smaller wells are prefabricated and transported to the location. The wells contain valves, expansion compensators, flexible expansion hose connections, and most often pipe anchor points. The wells are provided with manholes on top for the entering of maintenance and inspection crews. An example of a service well as well as its location in relation to the street and sidewalk is shown in Figure 14, (from Verkfraedistofa Nordurlands, Ltd., and Verkfraedistofa Sigurdar Thoroddsen, Ltd., 1976).

If at all possible pipes in the distribution system should be located underneath sidewalks. There are primarily two deciding factors for the location of the distribution pipe in the street cross section.

a. The service manholes must be within the sidewalk area.

b. Space must be left for other utility lines (electricity, telephone) past service well within the sidewalk.

In order to satisfy both of the above conditions the smallest distance from the private property line to the pipe is about 1.0 m and the sidewalk width must be about 2.0 m. The distribution line is usually buried to a depth of about 0.70 m or similar to electric cables.

Distribution network pipe is commonly made from pipe designed for 12 bar internal pressure. The network is so designed that the lowest pressure at the intake to houses at maximum load is not less than 1.5-2.0 bar. Furthermore the radiator systems in buildings may not be able to withstand pressures in excess of 7-8 bar. This means that in districts where there is a large difference between the elevation of the highest and lowest buildings, the distribution system must be in two or more separate sections, each one serving buildings within a given range of elevations. In districts of extensive areas with great pressure losses the above pressure limits necessitate more than one pumping station in order to keep the maximum pressure within the allowable limits. During construction, the distribution network is tested at 12 bar pressure.

A good example of a large difference in building elevations within a district is in Akureyri in northern Iceland. The town is built from sea level up to 80 m above sea level. The distribution system is so designed that the town is divided into two separate districts where the boundary between the two is approximately at the 45 m level.

Pressure drop in distribution networks is considerably more complicated to evaluate than pressure drop in the main transmission line. The main rule is to design the system so that the average pressure drop in the branch lines is on the order of 0.5-1.0 bar/km at maximum flow as in the main transmission line. The calculation of the pressure drop in the trunk lines and in the branch lines can be very complex, especially in the case of closed loops, and iteration methods must be applied. In general, however, these lines are designed with the possibilites of growth of hot water demand in mind, so that their diameters are frequently on the large side. Another reason for selecting especially the trunk lines of rather large diameter is to minimize the risk of insufficient pressure at those buildings located furthest away from the pumping station.

As stated above, the distribution network is designed in such a way that the minimum water pressure to any user is not less than 1.5-2.0 bar. The maximum pumping pressure, on the other hand, should be chosen so that the maximum pressure to any user is not over 7.0-8.0 bar.

## 6.3. House lines and house connections

House lines vary in size depending on the size of the building. The most common sizes are 20 and 25mm nominal diameter, insulated with polyurethane foam inside a polyethylene protective pipe as previously described (section 6.1.2.). Larger buildings need larger pipe, anywhere from 32mm to 100mm nominal diameter, depending on the building size and heating needs.

6.3.1. Water metering methods. As previously discussed (section 2.1.) there are two methods by which district heating systems in Iceland charge their customers for the hot water. The Reykjavik hot water supply system has always sold the hot water by volume as measured by a regular flow meter at the intake to each house. All other district heating systems in Iceland charge the customers by the socalled maximum rate method, where the customer has at his disposal a predetermined maximum rate of flow which he is free to use as and when he pleases. The maximum rate is set by adjusting the pressure drop in the water flowing through an orifice of a given size.

The maximum rate method is considered to offer certain advantages over the total volume method, among which the following may be mentioned:

(1) The maximum rate method encourages a more even rate of flow because the consumers try to make maximum use of the flow rate to which they subscribe. Experience from the Reykjavik district heating system indicates a peak hourly rate of flow of 30% higher than the average daily rate. Designers of district heating systems in Iceland have assumed a peak hourly rate of 15% over the average daily rate when the maximum rate method is used. The very limited available data on this point indicate an even lower peak load as seen in Figure 1 (Karlsson 1975). The maximum load for which the distribution system must be designed is therefore considerably smaller with the

maximum rate method than with the total volume method.

(2) The maximum rate meters (regulating valves) are less expensive than the flow meters and the operational cost is much lower since they need little maintenance and no meter reading is required. The billing is therefore much simpler. The Reykjavik district heating system has now started estimating the water volume used by consumers over much of the year and the flow meters are read only once a year. This way the cost of meter reading is held to a minimum.

An objection has been raised against the maximum rate method on the grounds that it will result in greater total volume flow of water than will the volume flow meter method. This could be critical in areas where the amount of water available is limited. Data on this point are rather limited although one comparison is available between the Reykjavik district heating system and the Seltjarnarnes district heating system (Figure 15). The Reykjavik data were obtained from unpublished results collected by Fjarhitun, Ltd., (1974) whereas the Seltjarnarnes data were collected by Karlsson (1975). The data show the average daily hot water consumption in $l/h \cdot m^3$ versus outside air temperature, all measured on overcast days when solar radiation did not influence the heating demand. It appears indeed from Figure 15 that the total consumption at Seltjarnarnes is somewhat higher than it is in Reykjavik, especially on relatively mild days. These results lend support to the contention mentioned above.

6.3.2. House connections. In order to get his house connected to the district heating system the individual home owner must pay a house connection fee. For this fee the house line is laid to the house and into the house ending in the flow metering assembly as described later. In houses already built and lived in when the district heating system is constructed the house connection usually enters the boiler room if at all possible. The house connection itself can be

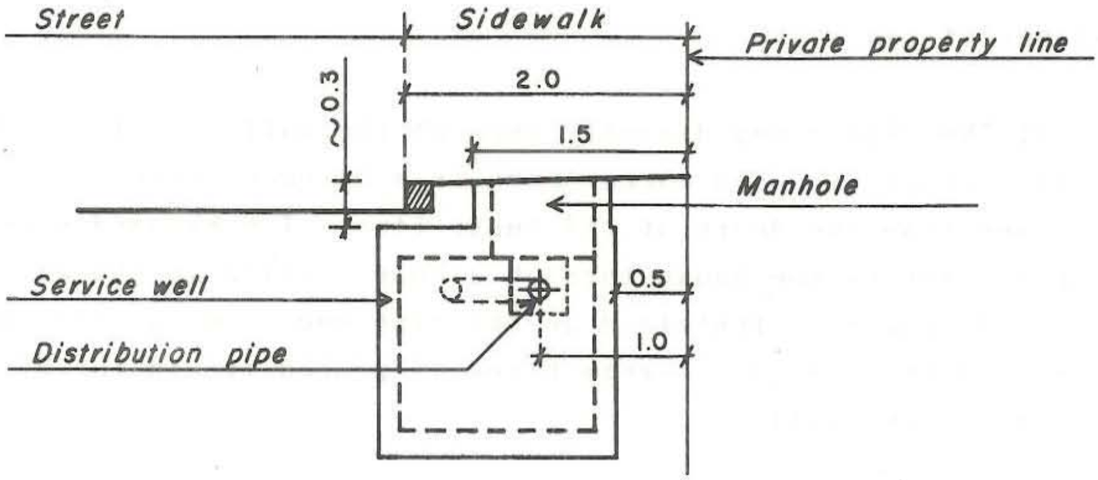



Fig 14

Cross section of sidewalk showing typical service well

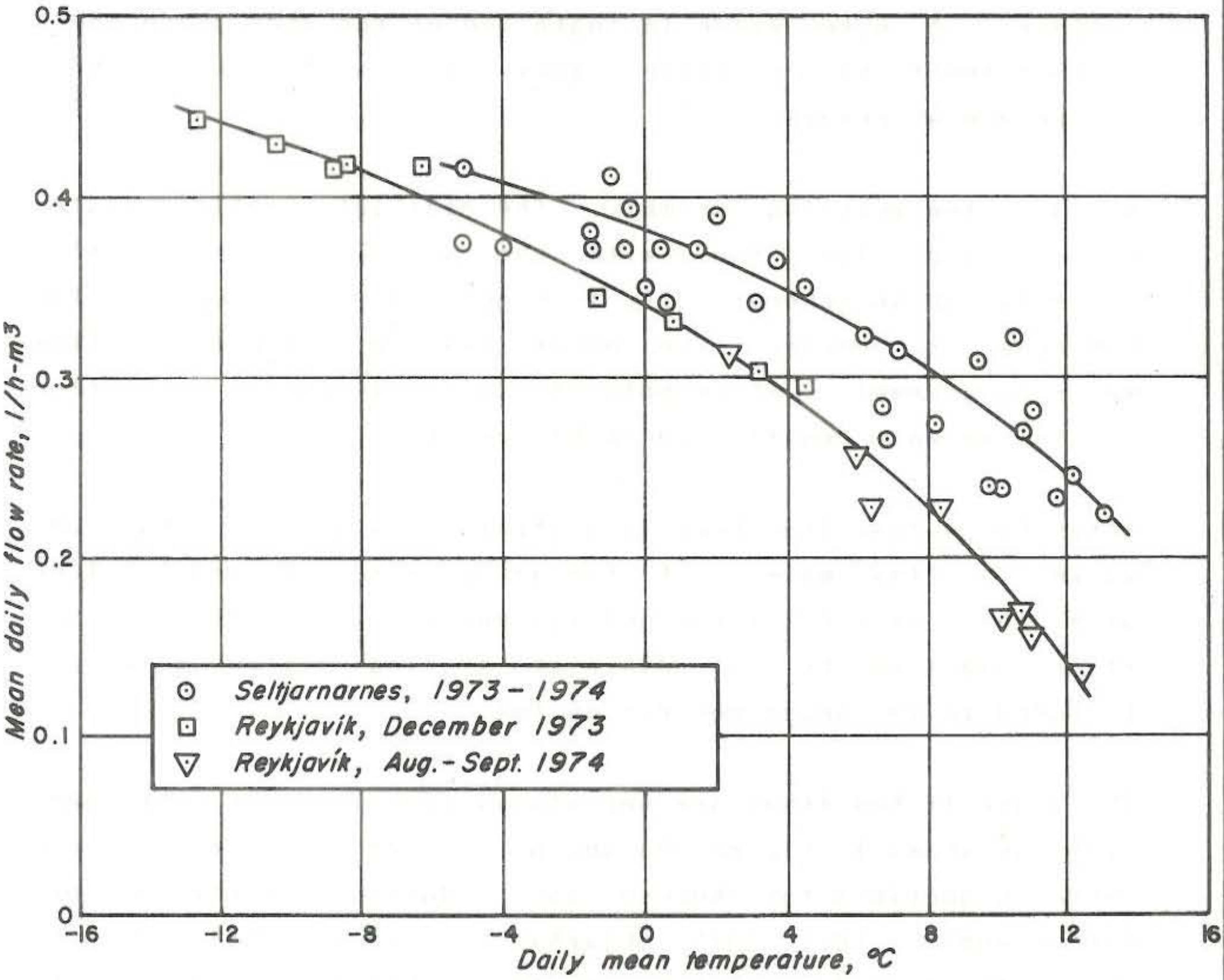



Fig. 15 Comparison of pumped relative volume of hot water in Reykjavik and Seltjarnarnes

either of two ways:

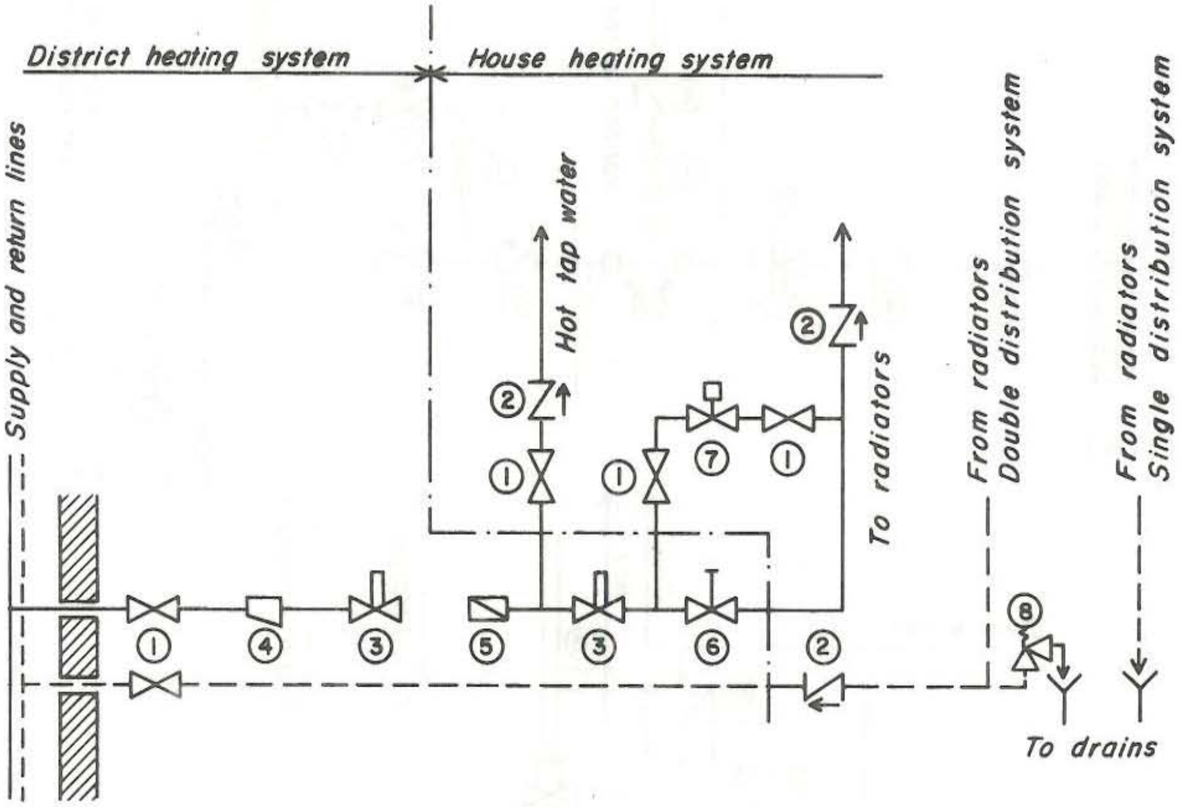
(1) The pipe comes directly through the wall. This type of intake is used where there is a basement with a floor lower than the depth of the house line. The exposed steel pipe enters the house through a hole drilled in the wall. A gate valve is installed on the pipe end coming through the wall and a concrete block is poured around the pipe outside the wall.

(2) The pipe enters the house up through the floor. This type is used on houses with no basement or when the floor sits higher than the depth of the house line. The exposed pipe enters the house through a hole drilled in the wall foundation and then vertically through a hole in the floor. A gate valve is installed on the pipe above the floor level and a concrete block is poured around the entrance as before.

6.3.3. Flow metering assembly. The district heating system house connection ends with the socalled flow metering assembly, shown schematically in Figure 16. It is up to the homeowner to connect the house heating system to the flow metering assembly. An example of a possible house connection is also shown schematically in Figure 16.

After the gate valve there is a strainer, a sealed regulating valve, a flow meter, if the water is sold by total volume used, and a branch for the hot tap water. This completes the items supplied by the district heating system which are included in the house connection fee.

The above listed items are not always connected in the same order as shown in Figure 17a and b. Figure 17a shows proposed house connections for Akureyri and Sudurnes (Verkfraedistofa Nordurlands, Ltd., 1975, Fjarhitun, Ltd., 1975). In both proposals it is suggested to use two regulating orifices, one before the hot tap water branch and another after the hot tap

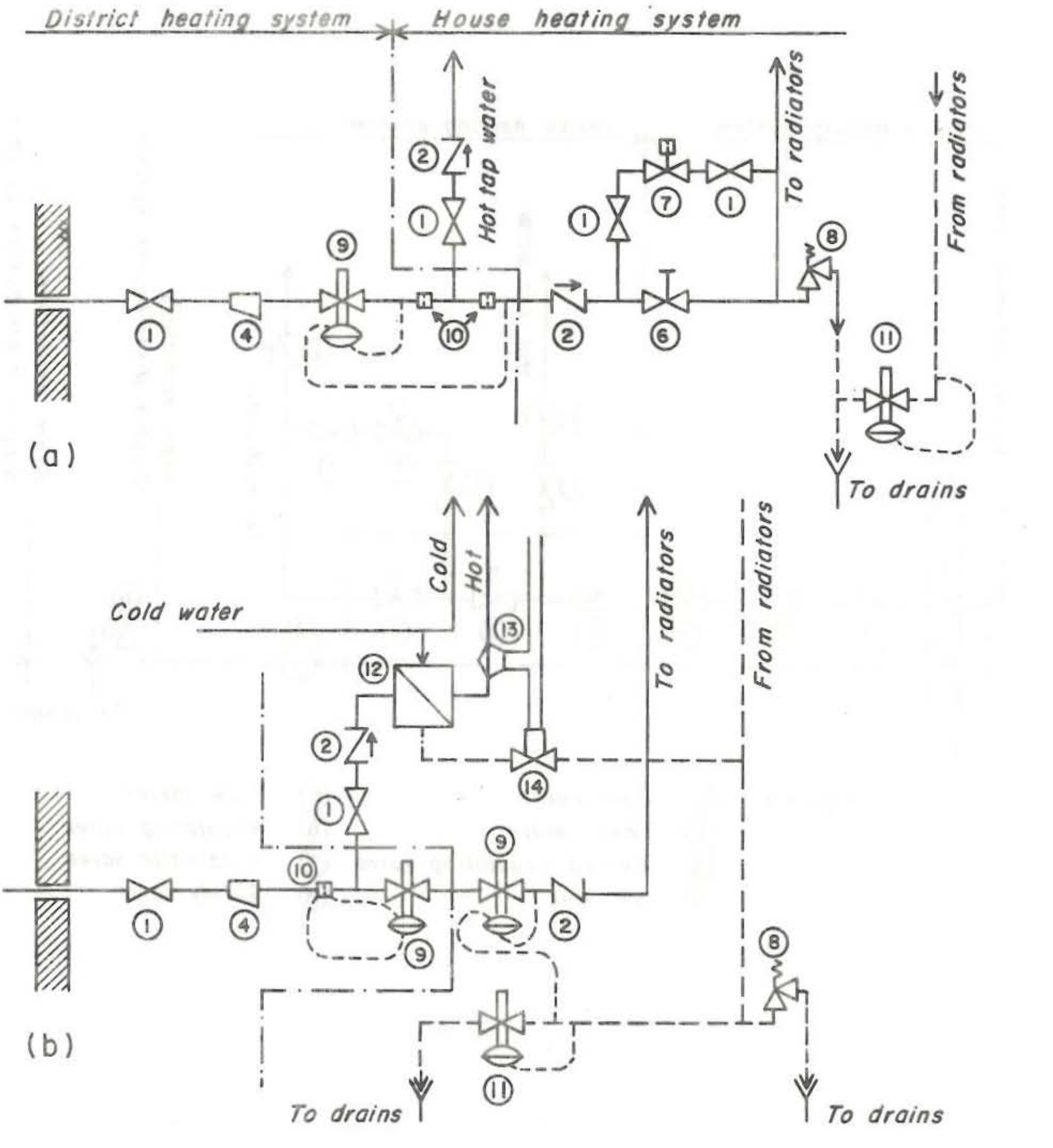



*Legend:*

1. *Gate valve*
2. *Check valve*
3. *Sealed regulating valve*
4. *Strainer*
5. *Flow meter*
6. *Regulating valve*
7. *Automatic valve*
8. *Safety valve*

Typical house connection - Reykjavik District Heating Service -

Fig. 16




LEGEND: (1) Through (8) Same as in Fig. 16
(9) Regulating valve (12) Heat exchanger
(10) Orifice (13) Pressure sensor switch
(11) Regulating valve (14) Solenoid valve

Examples of flow metering assembly arrangements and house connections

Fig. 17

water branch. The maximum flow rate for the heating system would then be somewhat lower than the hot tap water flow.

Figure 17b shows a house connection in the Seltjarnarnes district heating system. It shows the branch for the hot tap water before the regulating valve but after an orifice connected to the regualting valve. With this connection the regulating valve will not affect the hot tap water flow rate and the consumer gets the hot tap water he needs limited only by orifice resistance. This system has been quite successful and as far as is known the consumers are generally satisfied with the arrangement. The only drawback is that the regulating valve shuts off the flow of water to the heating system if the hot tap water usage exceeds the preset maximum rate but in general this is of no concern since only a short time is involved each time.

## 6.4 Thermal expansion of steel pipe

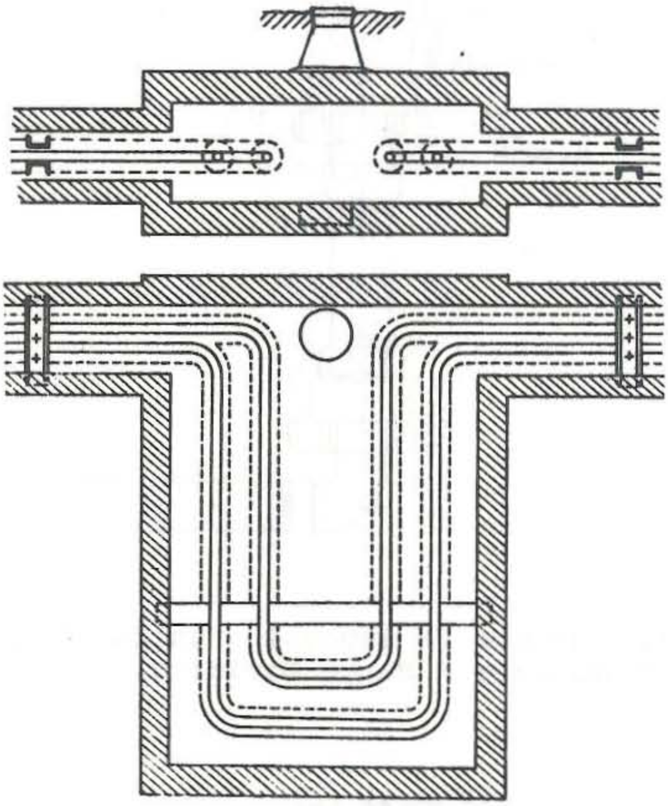
Since steel when heated expands on the order of $1.2 \cdot 10^{-5}$ $m/m \cdot {}^{o}C$ it is necessary in all hot water lines of steel to provide expansion compensators to enable the pipe to move in a longitudinal direction. The pipe is provided with fixed points or anchors where the pipe is securely fastened and prevented from moving. Between these fixed points the pipe must be flexible enough to move as determined by the thermal expansion coefficient without too high stresses being set up. In district heating pipelines a longitudinal expansion of 10-12 cm/100 m may be expected.

Thermal expansion is often most easily met at places where the pipelines change directions. If the pipe is laid as a long straight line the most usual form of compensators are U-shaped pipes consisting of straight sections of pipe and four welding bends. Sometimes such expansion compensators are placed in special concrete wells, which may also serve as drainage places, inspection wells and junction boxes. An example of such construction, used in Denmark, is shown in Figure 18a (Olufsen, 1963).

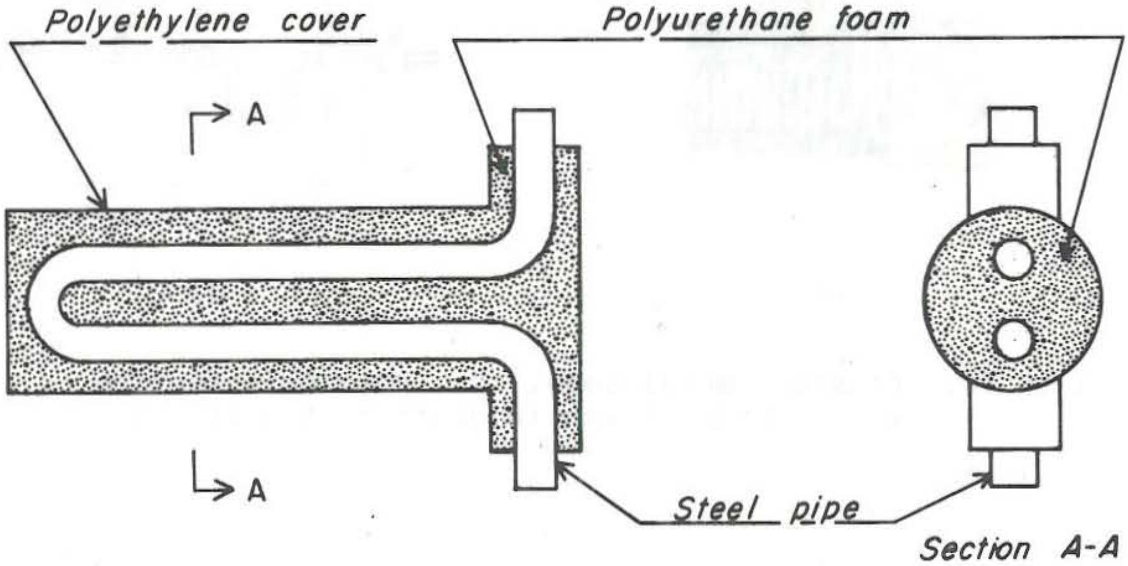
Another example of a U-shaped expansion compensator is shown in Figure 18b. This type has been used in Iceland for the smaller diameter pipe (< 80 mm nom.dia). This construction consists of two straight pipe sections, one 180$^{0}$ bend and two 90$^{0}$ bends, all welded together. The unit is enclosed in a polyethylene protective cover and insulated with polyurethane foam, much the same way as the pipelines previously described (section 6.1.2.).

In many cases it may be inconvenient or impossible to install the number of bends in a pipeline needed to compensate for the expected thermal expansion. Other types of thermal expansion compensators are then needed and there are several types on the market. One system which is widely used in the United States but has not found much use in Iceland is the ball joints expansion unit shown in Figure 19. These are rigid steel or malleable iron units, which are able to flex through 15 to 40 degrees, and to swivel through 360$^{0}$ if required. These ball joints are able to withstand high temperatures without leakage.

Flexible metal hose and tubing are available for a wide range of conditions for temperature and pressure, and are made in two basic constructions: corrugated or interlocked. The corrugated type shown in Figure 20a may have either annular or helical corrugated formations, usually covered with metal braid and is well adapted to high pressure, high temperature leakproof service. These hoses are essentially intended to absorb only those strains which are perpendicular to their own axis. For this reason, they should generally be installed at right angles to the movement involved. In Icelandic district heating systems this type of expansion joints has been extensively used for the connection of house lines to the branch lines, usually in sizes not exceeding 50 mm nominal diameter (see figure 20b).



a. Expansion chamber with U-shaped welded compensator and drainage duct (Olufsen 1963)




b. U-shaped expansion joint in polyurethane foam insulation and polyethylene cover

Figure 18.

Fig. 18

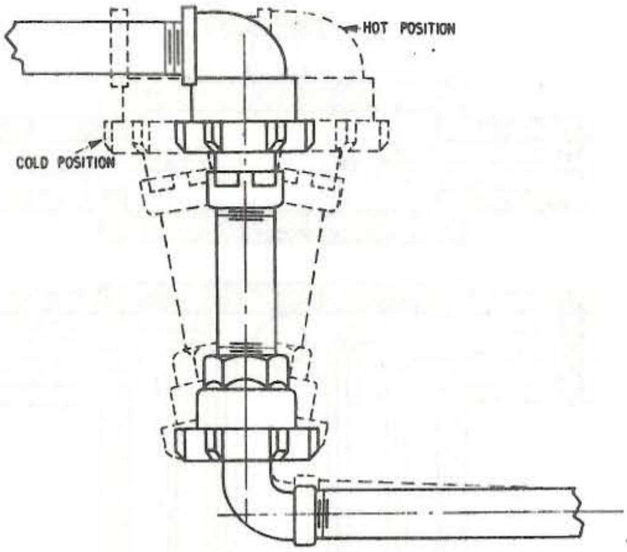
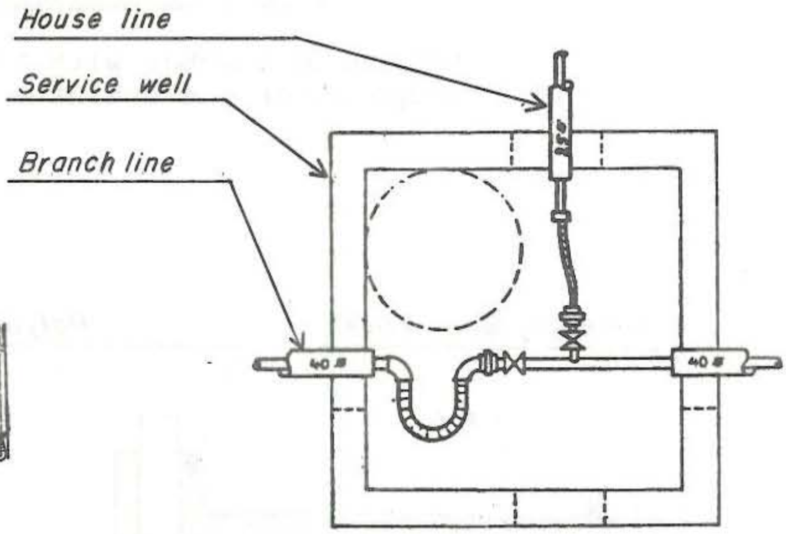



Figure 19. Ball joints as expansion units (from Aeroquip Corporation - Barco Division).




(a) (b)

Figure 20. Flexible metal hose. a. Hose construction b. Example of use in district heating service.

Bellows type expansion joints (Figure 21) are used very extensively in all types of heating pipework. These compensators are usually employed with large diameter pipelines and Z-shaped swing compensators. Their main advantage is their completely sealed costruction, the expansion movement being absorbed by the flexing of a carefully designed convoluted section of high grade material of thinner section than that of the pipe.

There are two basically different types of bellows systems (Diamant and McGarry, 1968):

(1) The axial type (Figure 21, top) is a plain bellows used in compression and extension along its axis in the conventional manner. This method is useful where relatively small amounts of expansion have to be absorbed, and where the pipe runs in a confined space in close proximity to other pipes in trenches or ducts.

It is nevertheless essential when using the axial type, to ensure that the pipe can be anchored very firmly and that the pipe run be guided effectively. The action of the bellows as a spring, in addition to its thrust on the pipe due to internal pressure tending to open it up lengthwise, can impose considerable forces on the pipe and its anchors. Owing to the compressive forces acting on the pipe run, it is essential to prevent any tending for the pipe to bend at any point. Hence the following rules should be carefully adhered to:

a. A pipe containing an axial type bellows must have a strong anchor on each side of the bellows.

b. The pipe guides must be able to resist both vertical and horizontal forces at right angles to the pipe run, while allowing free movement along the pipe axis.

Because of these requirements it is clear that a pipe containing axial type bellows must never be supported on hangers.

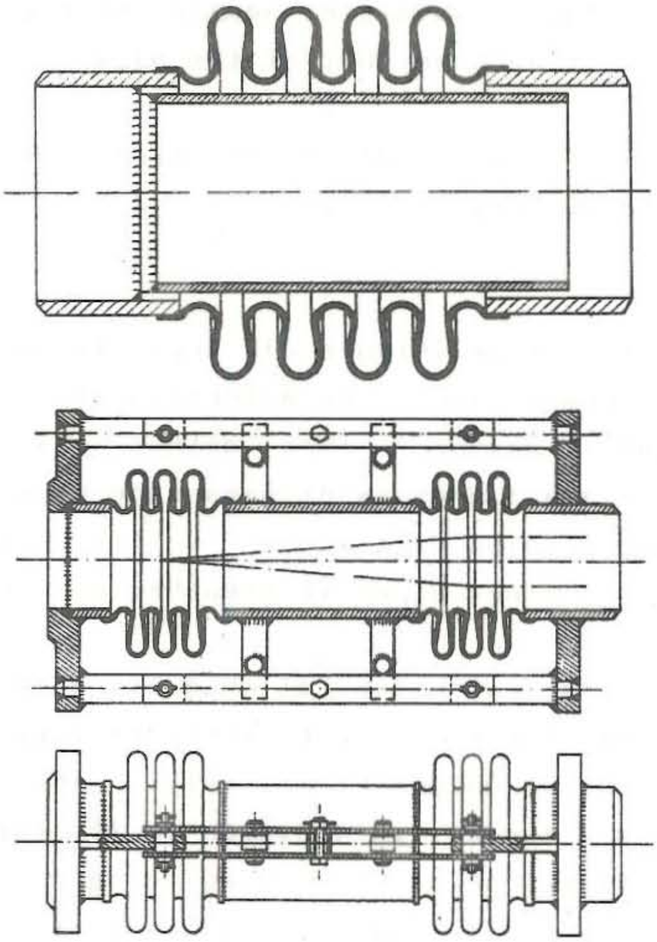


Figure 21. Top: Axial compensator with welding ends and internal sleeve.
Below: Swing compensator, one flanged end, the other weld end.

(2) The angular and swing compensator system works on an entirely different principle from the axial type and its characteristics are entirely different. The bellows is used in bending while restrained lengthwise by a hinge. The angular is a single hinged bellows used in sets of two or three in the same way as a two pin or three pin hinged structure. The swing compensator (Figure 21, center and bottom) is a combined unit comprising two bellows within a hinged framework. The compensator is designed to absorb expansion at right angles to its axis while restrained axially.

Whatever internal pressure applied to such a system tending to open out the bellows lengthwise is held by the tie bars and hinges and is not allowed to propagate as a force on the pipe or pipe anchors. The characteristics of the system may therefore be described as follows:

a. It is truly pressure balanced like the plain pipe loop and is therefore particularly suitable for medium and high pressure applications.

b. As the expansion to be taken up is dependent only on the angularity taken up by the bellows and the distance between them, very large expansion can be absorbed, even up to 50-75 cm at one point.

In view of the hinged arrangement, which gives a pressure balanced system, and the very low thrust required to bend a bellows rather than to compress and extend it, a system using angular and swing units may be treated as if the units did not exist as far as forces are concerned. As long as the pipe is capable of pushing itself along suitable supports and guides, this type of bellows system will operate well.

Bellows compensators of the axial type have been used extensively in the main supply lines of Icelandic district heating systems as well as in the larger diameter parts ( > 200 mm nominal diameter) of the distribution systems. U-shaped expansion systems seem to be more common for the smaller pipe. Experience from Sweden indicates that U-shaped expansion joints are more economical for the smaller pipe and the axial bellows type compensators for the larger pipe with the break even point lying somewhere between 100 and 200 mm nominal diameter (Olufsen, 1963). This seems to coincide with the Icelandic practice as previously discussed. Collection lines in the geothermal field, from boreholes to a collection tank or to a separation station in high temperature areas, however, are usually built with U-shaped expansion compensators or expansion bends, even though their sizes are often considerably larger than 200 mm.

## 6.5. Heat losses from pipelines

The pipelines, above ground or buried underground, constitute one of the main sources of heat loss in a district heating system. It is of the utmost importance that the insulation does not become damp, since this reduces greatly its thermal resistance and may increase heat losses to such levels as to render the district heating system capacity altogether insufficient. Besides, a damp layer of insulation on a steel pipe will induce rapid corrosion.

When computing the heat losses from a typical pipeline the following equation applies in general

$$Q = L(T_p-T_o)/(R_1+R_p+R_i+R_2) = L(T_p-T_o)/R, \qquad (45)$$

where $L$ = length of pipe, m

$T_p$ = temperature of fluid in pipe, °C

$T_o$ = ambient air temperature, °C

$R_1$ = thermal resistance on internal pipe surface, m·°C/W

$R_p$ = thermal resistance of pipe wall, m·°C/W

$R_i$ = thermal resistance of insulation and other material covering pipe, m·°C/W

$R_2$ = thermal resistance of external surface, m·°C/W

$R$ = overall thermal resistance, m·°C/W

For an insulated steel pipe the pipe wall thermal resistance is much smaller than the insulation thermal resistance and therefore it is usually neglected. The same is true for the internal and external surface thermal resistance terms. For asbestos-cement pipe which usually is not too well insulated the pipe wall resistance is included. The approximate form for the overall thermal resistance is then

$$R \approx R_p + R_i, \tag{46}$$

where $R_p$ may or may not be included. For a concentric insulation layer or for the pipe wall itself the thermal resistance is given by the equation

$$R_m = \ln((r_m+h)/r_m)/2\pi\lambda_m, \tag{47}$$

where $r_m$ = inside radius of layer, m

$h$ = layer thickness, m

$\lambda_m$ = thermal conductivity of layer, W/m·°C,

and the subscript m indicates any kind of concentric layer, pipe, insulation, etc.

In a recent report, Björnsson (1980) made a study of heat losses from the types of pipelines commonly used in Icelandic geothermal district heating systems. A summary of his results follows.

The thermal resistance term $R_i$ includes the thermal resistance of all material covering the pipe: insulation, protective cover, earth cover etc., as the case may be. The values of the resistance for the various pipe arrangements is presented below.

6.5.1. Regular above ground insulated pipe. In this case the thermal resistance of the insulation is determined by Equation (47). The thermal resistance of a sheet cover is neglected, but the thermal resistance of a polyethylene protective cover may be included. The expression for $R_i$ is then

$$R_i = (\ln((r_i + h_i)/r_i)/\lambda_m + \ln((r_c+h_c)/r_c)/\lambda_c)/2\pi \ , \qquad (48)$$

where subscript i denotes insulation and c is the protective cover.

6.5.2. Buried steel pipe insulated with polyurethane foam in polyethylene protective cover. As shown in Figure 11b there is a sand fill around the pipe in the pipe ditch making a sand layer of approximately 15 cm thickness around the pipe. The ditch is then filled with compacted earth. The thermal resistance for this pipe is then assumed to be the sum of four parts:

a. The thermal resistance of the insulation.
b. The thermal resistance of the protective pipe.
c. The thermal resistance of the sand layer taken as a concentric layer of 15 cm thickness.
d. The thermal resistance of the compacted earth surrounding the sand layer.

The thermal resistance of the soil surrounding a buried pipeline is given by the equation (Eckert and Drake, 1959)

$$R_e = \ln((2(h/r)^2-1) + 2(h/r)\sqrt{(h/r)^2-1})/4\pi \ \lambda_e, \qquad (49)$$

where r = radius of pipeline (in this case the sand cylinder), m

h = depth of the pipe center, m

$\lambda_e$ = thermal conductivity of soil, W/m·°C.

The expression for the overall thermal resistance is then approximated by

$$R = (\ln(r_i/r_p)/\lambda_p + \ln(r_c/r_i)/\lambda_i + \ln(r_s/r_c)/\lambda_c$$
$$+ \ln(r_e/r_s)/\lambda_s + \ln((2(h/r_e)^2 - 1)$$
$$+ 2(h/r_e)\sqrt{(h/r_e)^2 - 1})/2\lambda_e)/2\pi \,, \qquad (50)$$

where subscripts p,i,c,s,e denote the pipe, insulation, cover, sand, and earth, respectively, and r is the inner radius for a concentric layer. Equation (50) is only an approximation since (49) is based on the assumption that the surface of the buried pipeline (in this case the sand cylinder) represents an isotherm. This is not true for the present case, but in view of the assumption that the sand layer around the pipe has an annular shape, Equation (50) is considered a fairly good approximation for the thermal resistance.

6.5.3. Pipeline covered by earth and grass. It is seen that for $h/r \gg 1$ the Equation (49) reduces to

$$R_e \approx \ln(2h/r)/2\pi\lambda_e, \qquad (51)$$

i.e. the thermal resistance term is the same as if the pipe was covered by a concentric earth layer of outer radius 2h. The thermal resistance for the pipeline covered by earth and grass cover is now approximated by dividing the pipe into two halves, an upper half covered by a concentric earth cylinder of outer radius h and a lower half covered by a concentric earth cylinder of outer radius 2h. The thermal resistance for the earth around the pipe is then approximated by

$$R_e \approx \ln(h/r)\cdot\ln(2h/r)/\pi\lambda_e\ln(2(h/r)^2), \qquad (52)$$

where h is the distance from the earth and grass cover surface to the pipe center (see Figure 12b).

With the thermal resistance terms for the various types of pipeline determined by the Equations (48), (50), and (52) the heat losses from the pipes may now be evaluated. For this purpose the following conditions are assumed

$T_o$ = external air temperature = -15°C
$T_p$ = fluid temperature = 80°C and 60°C.

The conductivity values used are shown in Table III, and the quantity of water flowing in the pipes was such as to create a pressure drop of 0.5 and 1.0 bar/km. The mineral wool insulation thickness for the above ground steel pipe is 50 mm and the polyurethane foam insulation thickness for the buried steel pipe varies from 26 mm for the smallest pipe (20 mm nom.dia.) to 57 mm for the largest pipe (250 mm nom.dia.). For the earth and grass covered asbestos-cement pipe the earth layer thickness was assumed to be 600 mm for all pipe sizes. The conductivity value of earth for the buried pipeline and for the earth and grass covered asbestos-cement pipe was assumed $\lambda_e$ =1.0 W/m·K.

The results of the evaluation of cooling of fluid in the various types of pipelines in accordance with the above listed assumptions have been reproduced from Björnsson (1980) and are shown in Figure 22.

Table III

Thermal conductivity values used for evaluating heat losses from pipelines (Bjornsson, 1980).

| | Density, $\rho$, $kg/m^3$ | Conductivity, $\lambda$, $W/m \cdot K$ |
|---|---|---|
| Polyurethane foam | - | 0.035 |
| Rockwool | - | 0.040 |
| Glasswool | - | 0.042 |
| Polyethylene plastic pipe | - | 0.40 |
| Asbestos-cement pipe | - | 0.465 |
| Steel pipe | - | 46.0 |
| Dry sand | - | 0.31 |
| Dry gravel | 1700 | 0.70 |
| Sandy moraine | - | 1.15 |
| Earth and clay | 1500 | 1.45 |
| Earth and clay | 2000 | 2.00 |

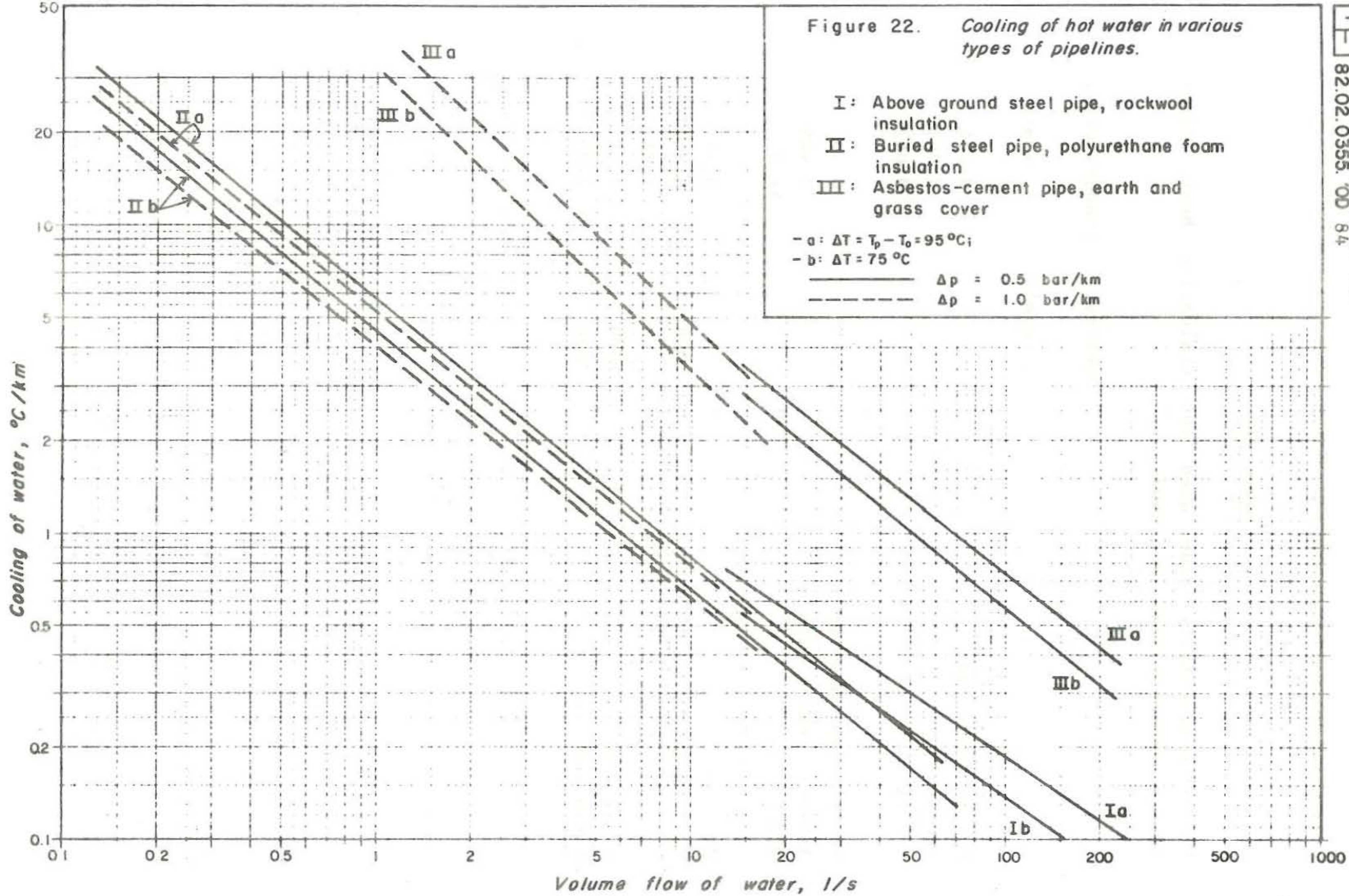



Figure 22. *Cooling of hot water in various types of pipelines.*

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## 7. PUMPING AND PUMPING STATIONS

The pumping jobs needed in a geothermal district heating system may be divided into three categories:

1. Pumping of the geothermal fluid from the boreholes to a deaeration and storage tank.

2. Pumping of fluid from the geothermal area to the district to be heated through the main transmission pipeline.

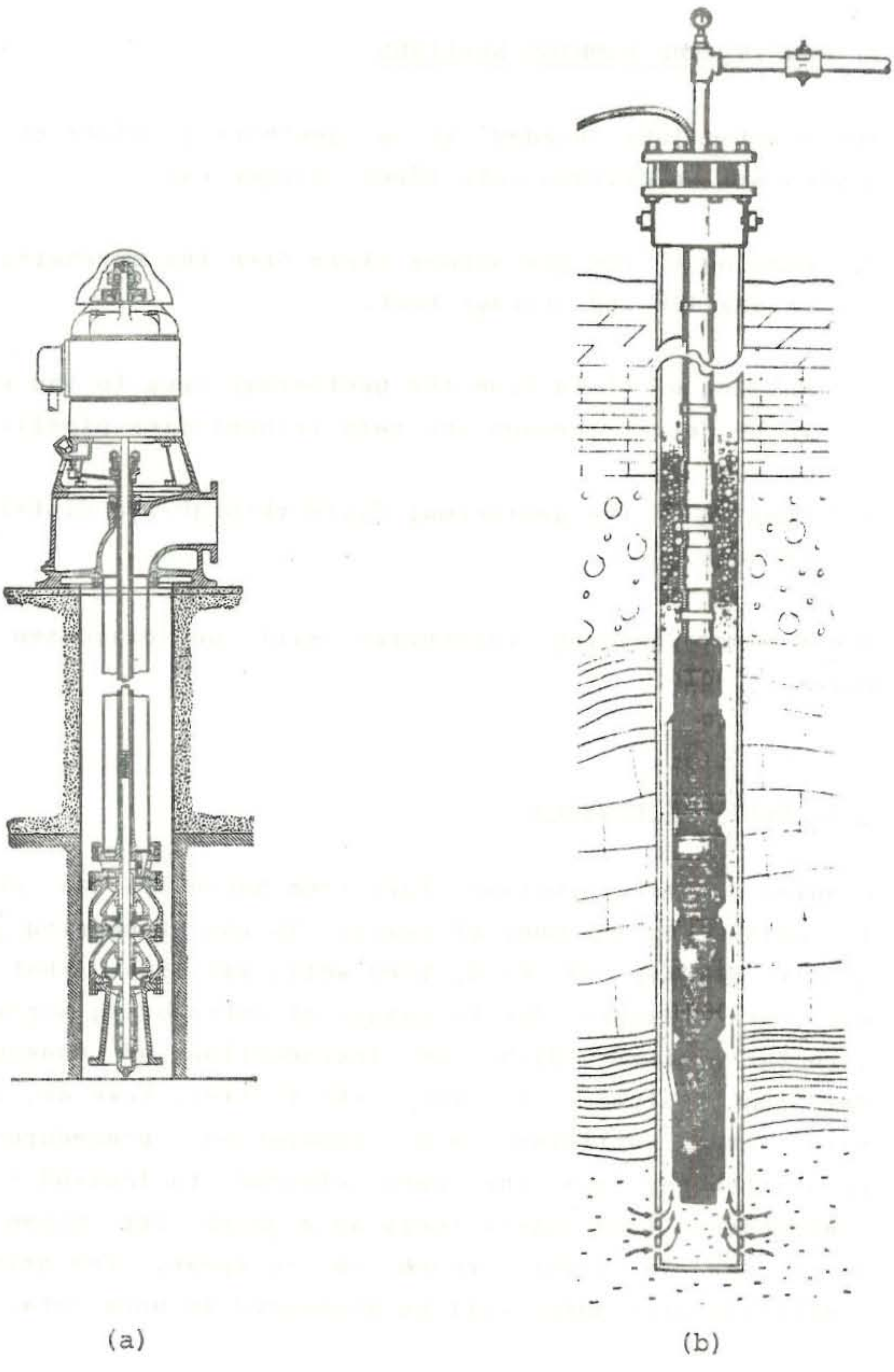
3. Pumping of the geothermal fluid through the distribution network.

These three pumping categories will be discussed briefly below.

### 7.1. Borehole pumping

Pumping of the geothermal fluid from boreholes has been done in Iceland for a number of years. In the pioneering years the quality and size of the drilled wells was such that pumping was not feasible due to danger of introducing surface water into the wells. With the introduction of modern rotary drilling equipment in the late fifties, however, it may be said that drilling and completion procedures were revolutionized and the path cleared to Iceland´s complete independence from fossil fuels as a means for house heating which now is within reach so to speak. The drilling and completion procedures will be discussed in more detail later.

The type of borehole pumps most widely used for geothermal operations in Iceland has been the multistage, mixed flow impeller type turbine pump, driven by an electric motor on top of the well through a line shaft going down through the well to the submerged pump assembly (Figure 23a). The depth at which the pump is set varies with conditions in the geothermal



(a) (b)

Figure 23. Deep well turbine pumps. (a) Line shaft pump.
(b) Submersible pump.

reservoir. In Reykjavik, for example, a depth of 100-150 m is common for these pumps. The use of multiple stages permits pumping against very high heads.

The first deep well pump to be used in a geothermal well in Iceland was installed at the Laugardalur geothermal area in Reykjavik in 1960 (Ragnars 1978). This pump was of a standard type with a semiopen impeller and the line shaft was supported by open rubber bearings lubricated by the hot water. The temperature of the geothermal water being pumped was 130°C and it was soon apparent that the rubber bearings could not stand up to the high temperature.

Different types of rubber bearings were tested, and the temperature limitation was generally found to be around 110-120°C. Additional wear by sand carried by the water contributed also to a short bearing life. Differential thermal expansion between the pipe column and the shaft added to the problem by causing the impellers to stick due to a limited end play between the impeller and pump housing.

Various methods and designs were tested in order to overcome these problems. Bronze and babbitt bearings were tested, but wear by sand continued to be a problem. In 1965 a new type of pump was tested with closed impeller (double-shrouded) and an enclosed, oil lubricated line shaft. The first pump of this type had bronze bearings lubricated by oil introduced at the top of the cover pipe. This method of lubrication, however, was abandoned because of limited bearing life (four months) and because of contamination of the geothermal fluid by the oil.

A new design using teflon bearings was tested in 1966. These bearings were lubricated by filtered geothermal water pumped down the shaft-enclosing pipe. This design has since been used for all deep well pumps of this type, and bearing maintenance is no longer a problem, as evidenced by the teflon bearing life of three to four years in comparison with a bearing life of three to four months with other types of bearings and methods of lubrication previously used.

The long drive shaft of these pumps, however, remains a problem, especially when the well is not fairly straight or if the pump has to be set at great depths. The noise from the pump motor at the surface may also be a nuisance, especially when the well is located in a populated area. In order to avoid these problems, the deep well pumps are also built with a closely coupled driving motor. The motor is completely sealed and of a small diameter so that it can be submerged in the well (Figure 23b). This type of pump is therefore called a submersible pump. With this arrangement the long drive shaft is eliminated and the shaft friction losses and total thrust are minimized.

The submersible pump has recently been taken into use in at least one geothermal area in Iceland (Sverrisson, 1980). The pump has been operating for almost two years at a depth of 369 m, pumping 50 l/s of water at 80°C against a head of 12-14 bar measured at the wellhead, with water level in the well at a depth of 220 m. The casing in which the pump operates has an O.D. of 298.5 mm (11-3/4").

The maximum temperature at which deep well turbine pumps in Iceland have been used so far is about 130°C (Ragnars, 1978). This may not be the temperature limit for the line shaft pumps, but the present pump design does not allow for more differential expansion between the pipe column and line shaft than experienced at this temperature at a depth of 130 m. According to the manufacturer of submersible pumps, such pumps have operated successfully in the oil industry running continuously at 150°C (Sverrisson, 1980).

As the output of geothermal wells may vary considerably the minimum size of the top part of the drilled well must be chosen in accordance with the desired output and pump size to be used. The length of this top part of wider casing must be

150-400 m to enable the installation of the pump to the desired depth. Table IV gives approximate capacities of turbine pumps in well casing of different size.

Table IV

Approximate capacities of deep-well turbine pumps (Kent,1936).

| Nominal well casing dia. | | Turbine pump output, litres/sec. | |
|---|---|---|---|
| mm | in | Medium capacity | High capacity |
| 100 | 4 | 3 | 5 |
| 150 | 6 | 16 | 20 |
| 200 | 8 | 30 | 40 |
| 250 | 10 | 65 | 75 |
| 300 | 12 | 95 | 125 |
| 350 | 14 | 115 | 190 |
| 400 | 16 | 150 | 250 |
| 450 | 18 | 190 | 380 |
| 500 | 20 | 220 | 500 |
| 600 | 24 | 315 | 760 |

## 7.2. Pumping for main transmission line

The water from the boreholes is commonly pumped into an elevated storage tank for deaeration. The water level in this tank should be the highest point in the hot water collection system. From the deaeration storage tank the hot water flows by gravity to the main transmission line pumping station. The necessary elevation difference between the tank and the pump intake will depend on the water temperature, but care must be taken to make it sufficient to avoid cavitation at the entrance to the pump impeller.

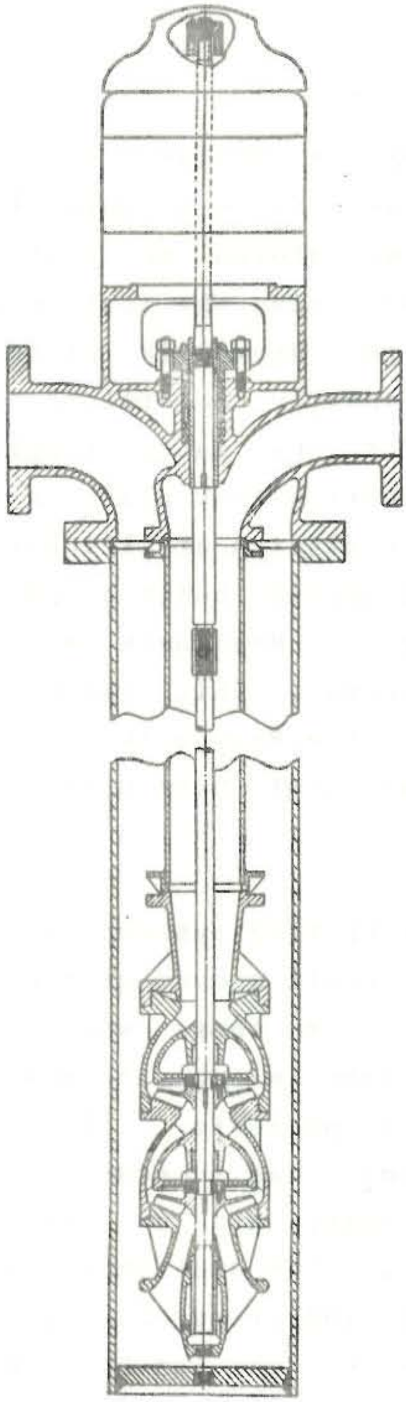
The types of pumps used for the main transmission line will vary in accordance with the required volume and discharge head. Vertical multistage turbine pumps, not unlike the deep well pumps described in the previous section, are common where relatively high pressures are needed. Figure 24a shows an example of the construction of such a pump and the arrangement in the pumping station (with the pump intake below) is shown in Figure 24b. For smaller systems the simpler, single stage, centrifugal pumps may suffice.

For added safety the pumping stations are usually provided with two or more pumps feeding the line in parallel. This way the pumping will continue uninterrupted even though a pump is being overhauled or repaired.

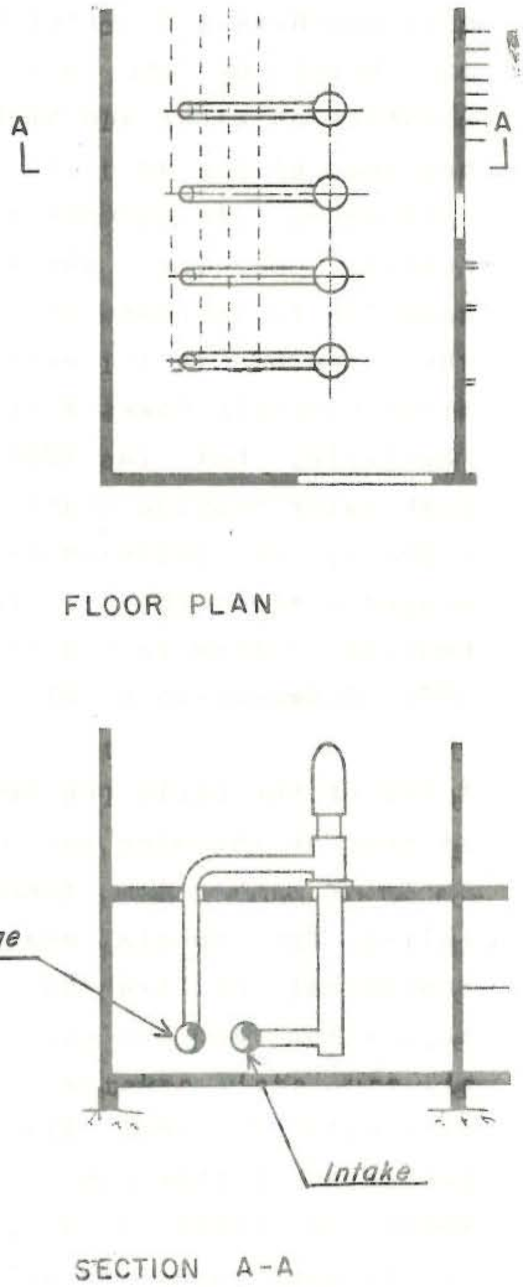
## 7.3. Pumping for distribution network

The hot water being pumped to the district to be heated, is led into a storage tank usually located at a high spot near the district. If the tank sits high enough and if the district is low lying and not too large, no pump may be needed for the distribution system. This case, however, may be considered an exception and it will be assumed that pumping is needed to move the water through the distribution system.

The pumping station is located at a level sufficiently low to prevent cavitation at the pump intake. The type of pumps will vary with the size and extent of the distribution system. For large systems the vertical, multistage turbine pumps, similar to those used for the supply line (see Figure 24), may be used. In smaller systems, the simpler single stage centrifugal pumps are common.



(a)




(b)

Figure 24. Vertical turbine pump (a), and possible layout in a pumping station (b).

## 8. HOT WATER HEATING PLANTS

Most geothermal district heating systems in Iceland designed and built by the early seventies were based on sufficient quantity of water and high enough temperatures so that no heating of the water is necessary even during the most severe cold waves. An exception to this is the Reykjavik District Heating Service which has traditionally maintained the capacity to increase the temperature of the water supplied when needed. In the early days this heating took place in the State Electric Power Works' steam power station at Ellidaar in Reykjavik, but in 1967 the Heating Service built a special peak water heating station at Arbaer in Reykjavik with a capacity of approximately 35 MW-thermal. This station has played a minor role in the operation of the Reykjavik district heating system in recent years and has hardly been used since 1978 (Gudmundsson et al., 1980).

A few of the early geothermal district heating systems as well as some of the more recent systems are built around geothermal water of rather low temperature (60°C or less) which has called for special measures in the homes in order to make an economical utilization of the water possible. As the population and thereby the heating demand has increased, more drilling for hot water has become necessary and by now the prospects in some places for finding the additional water needed are rather dim. Other ways will therefore have to be found in these areas in order to satisfy the heating demand and in many cases the only choice left is to built heating plants for the purpose of raising the water temperature when needed.

With the oil crisis, starting in 1973, the economics of geothermal district heating systems was completely changed. In recent years attention has been directed towards geothermal areas which earlier were out of the question for district heating due to distance from the consumer market, limited water quantity or low water temperature. Both limited water quantity and low water temperature are encountered in some places, and the only choice in such cases is to heat the water in order to bring the capacity of a district heating system up to a point where the heating demand is met. It appears therefore that hot water heating plants will to an increasing degree be operating in Icelandic district heating systems in the future.

Two geothermal district heating systems in Iceland where water heating is required will be discussed briefly below. One of the systems is the Akureyri district heating system which already is in operation (Verkfraedistofa Nordurlands, Ltd., and Verkfraedistofa Sigurdar Thoroddsen Ltd., 1976, Sverrisson, 1979) where the water quantity is limited but water temperature from boreholes is ideal for district heating. The other system is in Talknafjordur in the northwest peninsula where sufficient volume of geothermal water is available but its temperature is only 45°C (Bjarnason, 1980). This system is still in the planning stage.

## 8.1. Insufficient water quantity

The Akureyri district heating system was taken into use in 1978 (Sverrisson, 1979) and at this time (June 1981) over 90% of all homes in the town have been connected. The water for the system comes from boreholes located at a distance of approximately 13 km from the town. The projected hot water requirement for the system is 300 l/s of 90°C water (Verkfraedistofa Nordurlands,Ltd., and Verkfraedistofa Sigurdar Thoroddsen, Ltd., 1976), but in spite of an extensive drilling program in the geothermal area, only 150 l/s of water are available today. In order to cope with this problem a part of the distribution system (30% today) has been made double with the return line bringing the water back to a heavy fuel fired water heating plant where its temperature is raised from about 35-40°C to about 90°C in a water heating boiler of

12 MW-thermal capacity. This present system is not sufficient to fulfill the heating requirements of the town during the most severe cold waves and while geophysical exploration is continued in the vicinity in the hope of finding more geothermal water, the plan is to install another 12 MW-thermal fuel fired boiler in the water heating plant. Eventually, when the country's electrification plans are sufficiently far along, it is projected that these fuel fired boilers will be replaced by electrically heated boilers.

## 8.2. Low water temperature

Geothermal water of temperature too low to be utilized effectively for a district heating system (under 60°C) is found in numerous places in Iceland. Most often such water is of no practical interest due to its location in remote areas too far from populated districts. Occasionally, however, the water is in the vicinity of inhabited areas, and the question arises if or how such water may be utilized efficiently for district heating.

Two areas of the nature discribed above have been under consideration in Iceland in recent years. One is at Holar in Hjaltadal, an educational center in Northern Iceland, where a borehole at a distance of 9 km has yielded a flow of 20 l/s of water at 56°C. The heating requirement at Holar has been estimated at 420 kW-thermal (Gudmundsson et al., 1979). The other area is at Talknafjodur, a village on the north-west peninsula of Iceland, where a borehole at a distance of 4 km from the village delivers 25 l/s of water at 45°C. The heating requirement of Talknafjordur has been estimated to be 978 kW-thermal.

The above two areas have two things in common: abundant flow of water compared with the prospective heating market and low water temperature. Admittedly the Talknafjordur area is considerably worse off with its 45°C water whereas the Holar temperature is at or near the lower limit of what has been considered usable for district heating systems in Iceland. For both of these areas the use of a heat pump bas been considered for the purpose of raising the water temperature for district heating. For the Holar area, however, the heat pump idea has been abandoned and the decision made to use the water as is for district heating and fresh fish farming. The Talknafjordur scheme, on the other hand, is still in the planning stage, and it will be discussed briefly below.

The use of heat pumps has been growing rapidly in recent years, but as yet it has not been introduced in Iceland. In most cases the heat pumps come in small units designed for individual homes or buildings. Lately, however, larger units have become available, which seem to be adaptable for geothermal use in Iceland.

In the Talknafjordur district heating plan (Bjarnason, 1980) the water temperature is to be raised to 70°C and transmitted to the users in a single pipe distribution network. The use of a heat pump for water heating will be very favourable for geothermal water due to the relatively high temperature of the heat source (geothermal water). The coefficient of performance (COP) for the heat pump raising the water temperature to 70°C is estimated to be 4.15. An even better COP is obtained if the heat pump is required to heat the water to a lower temperature. Thus an estimated COP of 5.0 is obtained if the heat pump delivers the water at 60°C.

Weather data from Talknafjordur indicate that water temperature of 60°C would be sufficient 90% of the time based on a temperature requirement of 70°C during the most severe cold waves. Due to the high initial cost of heat pumps, the analysis for Talknafjordur has shown (Bjarnason, 1980), that it will be more economical to use a heat pump to deliver two thirds of the thermal power needed (to 60°C) and a heavy fuel oil burning hot water boiler for the remaining one third of the thermal power (to 70°C) than to have a heat pump with the capacity needed to deliver the full power. Both arrangements will result in a heating cost for the consumer in Talknafjordur considerably below the present fuel oil burning cost (50% or better), but the heat pump-boiler combination will deliver the heat energy at a price of about 15% less than will the heat pump alone.

Due to uncertainty in electric power supply as well as uncertainty in the pricing policy of the electricity supplier for heat pump uses, the Talknafjordur scheme, although very promising, has not yet been carried out. As mentioned earlier the heat pump has as of yet not found use in Iceland. The very favourable reports on heat pump operations abroad and the high coefficients of performance attainable in geothermal water heating should, however, make it an extremely attractive option as an aid in utilizing the lower temperature geothermal areas in Iceland.

# 9. AUTOMATIC CONTROL

The degree of sophistication of automatic control equipment for geothermal district heating systems in Iceland varies greatly from an almost complete automation of the large Reykjavik Hot Water Supply Service system to the absolute minimum that can be justified for some of the small and less complicated district heating systems. The 60 MW-thermal Akureyri district heating system in northern Iceland probably falls somewhere in between these two extremes and its control system will be described briefly below (from Verkfraedistofa Nordurlands Ltd., 1976, and Verkfraedistofa Sigurdar Thoroddsen, Ltd., 1976).

## 9.1. Automatic control center

The Akureyri district heating system consists of most of the various components which have been described up to this point in this report. These include the following:

1. Boreholes with deep well pumps located about 13 km away from the town.

2. Deaeration and storage tank where the hot geothermal water from the boreholes is collected.

3. Main transmission line pumping station.

4. Two storage tanks in town, each serving its section of the distribution network, which is in two parts due to the large differences in elevation in the town. The lower storage tank receives the hot water from the main transmission line.

5. Double pipe distribution network in part of the upper distribution network.

6. Pumping station, located near the lower storage tank, for pumping water from the lower storage tank to the upper storage tank and to a part of the upper distribution network. The lower distribution network is fed by gravity from the lower storage tank. The part of the upper distribution network not fed by the pumping station directly receives water by gravity from the upper storage tank.

7. Heavy fuel fired hot water heating boiler plant located near the lower storage tank.

The automatic control center is located in the pumping station in town (no. 6 above). It is so designed that it gives an overview of the operation of the system and enables the control of its equipment by remote sensing and remote control from one central location. The equipment runs automatically for the most part, but manual operation and control is possible if and when needed.

Figure 25 shows a line diagram for the control system. Remote sensing and remote commands are indicated as follows:

| | |
|---|---|
| Motor starting/stopping ..................... | push button |
| Motor running/cut off ....................... | indicator light |
| Deaerator tank water level .................. | gage |
| Position of control valves ..... ............ | gage |
| Pressure upstream/downstream of master valve. | gage |
| Rate of flow in main supply line ............ | gage |
| Storage tanks water levels .................. | gage |
| Rate of flow to lower distribution network .. | gage |
| Rate of flow to upper distribution network .. | gage |

## 9.2. Deaerator tank

The deaerator tank water level controls a valve on the main supply line on the discharge side of the pumps. This control valve is an electrically operated butterfly valve with less

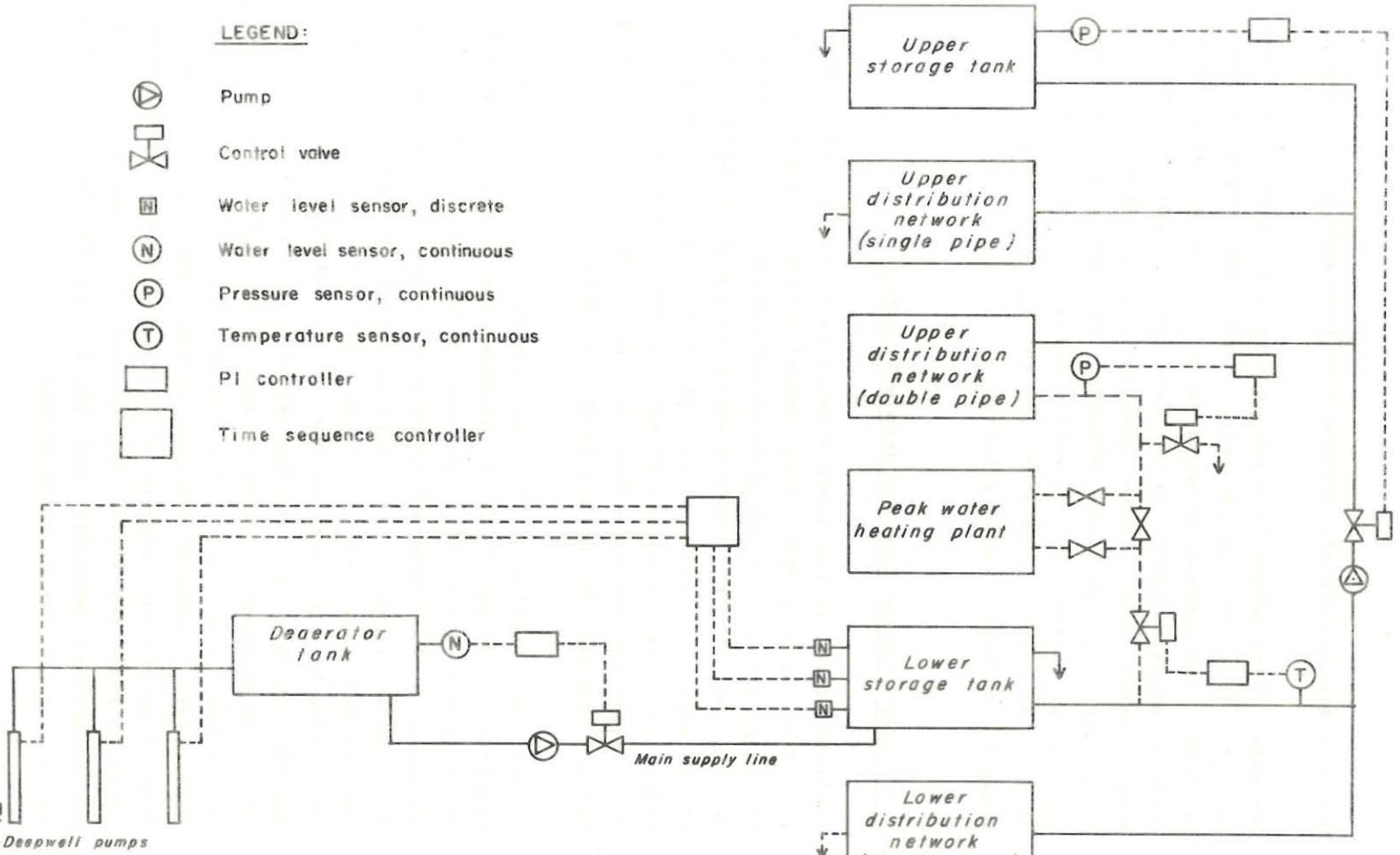



Fig. 25 Akureyri district heating automatic control system

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than one minute complete closure time from wide open position.

The water level sensor is a pressure sensor which transforms the water level into an electric current signal ranging from 0 to 20 mA. The controller operates in accordance with these current signals which also serve as an indication of the actual water level, both locally and at Akureyri. The controller is of the PID-type (proportional -integral -derivative control action) with continuous output from 0 to 20 mA. A remote indicator at the Akureyri control center shows the butterfly valve position.

## 9.3. Lower storage tank and deep well pumps

The water level in the lower storage tank at Akureyri controls the deep well pumps. This control action is discrete, i.e. the pumps are started and stopped when needed. The following description of the control action is based on three control stages with one pump acting at each stage as shown in Figure 25. As the number of wells in the system increases it is possible to have more than one pump at each control stage without changing the principle of the control action.

At low load conditions (in the summer) only one pump is required, controlled by a maximum water level in the tank. The sole function of this control stage, therefore, is to maintain a full tank and to prevent it from overflowing. As the load is increased, two pumps or all three must be used. The three pumps act in turns, one after the other, each controlled by a pressure switch at a different setting of 40-50%, 70-75%, and maximum water level, respectively. The first two of these pressure switches have small on-off level difference and are also controlled by time sequence controllers to prevent starting at times when not needed as well as to prevent stopping if it takes too long for the water level to reach the proper control level. The pumps may also be controlled manually and their position in the order of control stages can be interchanged.

## 9.4. Upper storage tank

The water level in the upper storage tank controls a valve on the upper distribution network trunk line on the discharge side of the pumps. The function of the valve is to maintain a maximum water level in the tank and to prevent overflowing. The maximum opening of the valve is adjusted so as to permit a rate of flow slightly higher than the average daily flow rate under maximum demand conditions as determined by experience.

The water level sensor is a pressure sensor which transforms the pressure into an electric current signal in the interval from 0 to 20 mA. In addition to acting as a control agent, the current may also be used to indicate the actual water level on a gage or on a continous strip recorder. The controller is a PI-type (proportional- integral control action) step controller and the control valve is an electrically driven butterfly valve.

## 9.5. Double distribution network

The backflow pressure in the double distribution network is controlled by a 0 to 20 mA output pressure sensor placed at a location in the backflow section of the distribution network where it is considered advisable to maintain a constant pressure. A PI-controller receives this output current signal and controls thereby a discharge control valve which discharges water to the town sewer system. The control valve is a step controlled, electrically driven butterfly valve.

The supply water temperature is controlled by proper mixture with the backflow water. The control valve is on the backflow pipe and controlled by a temperature sensor on the supply line. The controller is a PI-type step-controller.

## 10. DRILLING IN GEOTHERMAL AREAS

The first geothermal well in Iceland was drilled in 1928. The location was at the Thvottalaugar, an old hot spring used for decades for washing clothes. The drilling equipment was of a German make, an old rig brought to the country a few years earlier for mining exploration. The drilling was done by the rotation of drilling rods with a soft metal bit at the lower end. The grinding action of the drill bit was brought about by pumping brittle metal shots down through the drill rods with the circulating fluid. On reaching the well bottom the brittle metal shots were embedded in the soft metal crown of the rotating drill bit which thus became like a coarse grindstone working on the formation.

As the first small scale geothermal district heating system in Reykjavík proved its value, drilling was continued with this rig as well as with two other rigs of the same type purchased by the city of Reykjavík in the pursuing years (in 1937 and 1949). The penetration rate of these early rigs was extremely low, evidenced by the fact that in the years from 1933 to 1955, 72 wells were drilled of a total depth of almost 24,000 m, the deepest being 650 m. This corresponds to an average penetration rate of approximately 2 m/day.

These early wells were successful only to the extent that they were free-flowing or artesian. Pumping from the wells was out of the question due to the small wellbore (10-15 cm) and the surface casing was not cemented in. Stimulation of the water flow was sometimes attempted by airlift pumping, but this procedure contaminated the water with oxygen.

The Drilling Division of the State Electricity Authority (now National Energy Authority) was started in the late forties and today all drilling in Iceland, geothermal and otherwise, is carried out by the Division. All geothermal drilling is now performed by five rotary drilling rigs which the Division owns and operates, ranging in depth capacity from 400 to 3600 m.

The Drilling Division also owns and operates a few cable tool drilling rigs which are mainly used for cold water well drilling around the country. As these drilling rigs get involved in geothermal drilling in Iceland, their operation as well as the rotary drilling techniques will be described briefly below.

## 10.1 Cable tool drilling

The cable tool rig is not a true drill in the strict sense of the word since it does not rotate, but employs a heavy hammer bit that pounds and crushes the rock (Anderson and Lund, 1979, see Figure 26). The rock fragments mix with a water slurry in the well which is then bailed out. This drilling method is common for cold water drilling, and before rotary equipment was taken into use in Iceland (1958) it was extensively used for geothermal drilling. The cable tool drilling rigs have their advantages such as being comparatively inexpensive to buy and requiring only two men to operate. Among the disadvantages of this drilling method are the slow penetration rate as compared with rotary drilling, especially at depths below 200-300 m, and in geothermal drilling blowout prevention equipment cannot be adapted. The cable tool rig is therefore unsuitable for drilling in areas where the water temperature exceeds 100°C, and lower temperature wells, if free-flowing, may be dangerous if drilled by this method.

The main role of the cable tool rig in geothermal drilling in Iceland today is to pave the way so to speak for the rotary drilling equipment. This means that the rig is brought to the drill site to start the hole, to drill the first 25 to 50 m, and to set the conductor pipe to keep the loose surface layers from falling into the hole. The rotary drilling rig is very ineffective at shallow depths as sufficient weight cannot be applied to the drilling bit, so it is necessary to get the hole started by another drilling method.

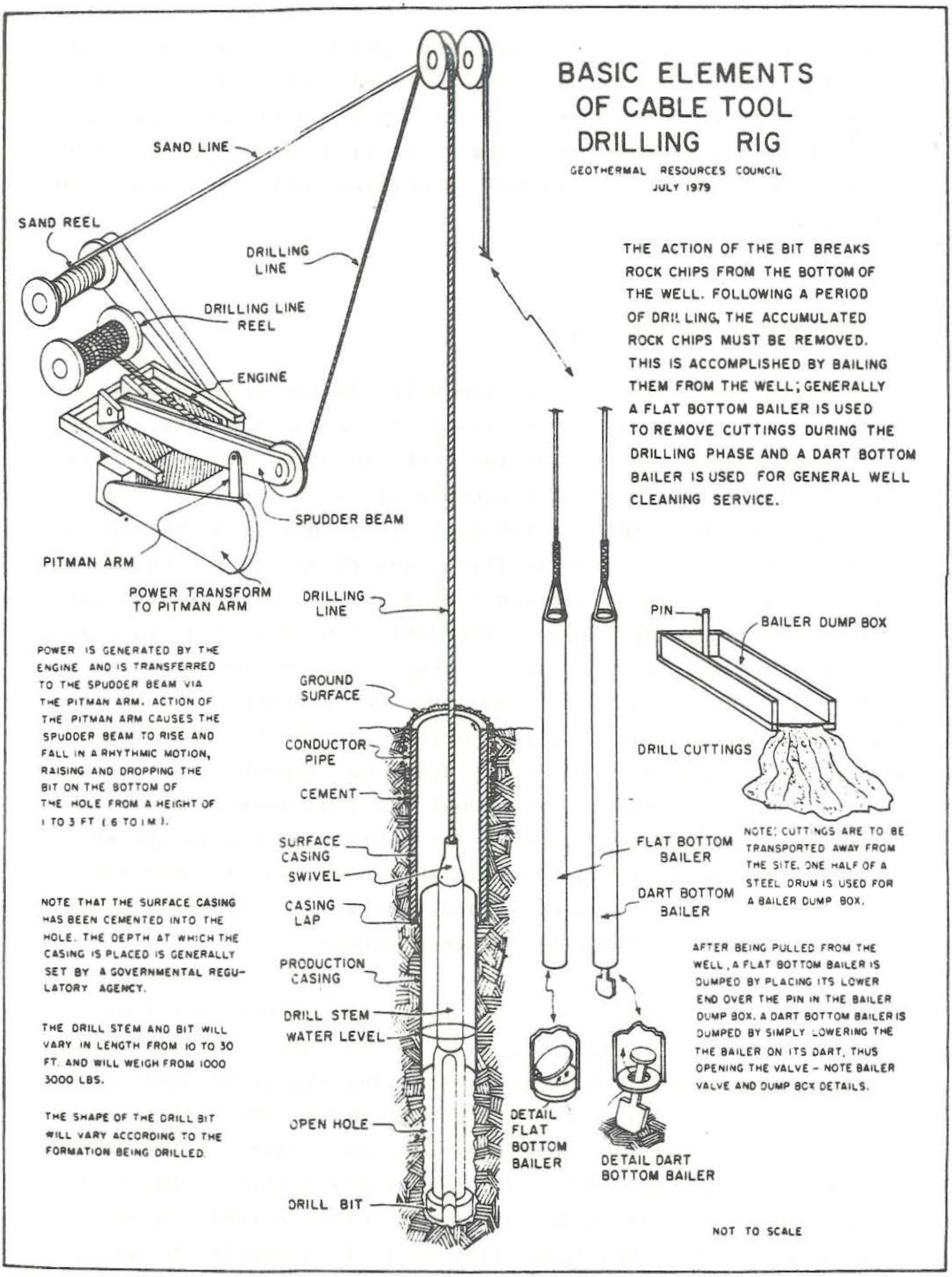



Figure 26. (From Anderson and Lund, 1979).

## 10.2 Rotary drilling

The rotary drilling technique is the most common drilling method today, in oil well drilling as well as in geothermal well drilling (Figure 27). The bit is more like a drill, even though the drilling action is just as much chipping and crushing as it is cutting. The cuttings are removed from the hole by fluid circulated down the drill pipe and up through the annulus between the drill pipe and the hole or between the drill pipe and the casing.

The most common drilling fluid used in rotary drilling is made of water mixed with bentonite clay with other additives as needed to adjust the density and other properties of the fluid. This is commonly called drilling mud and it has many useful properties such as preventing cavein of the wellbore, and the heavier density mud helps to contain high pressures encountered in drilling. In geothermal drilling in Iceland, however, water alone is normally used as drilling fluid, although occasionally bentonite mud is circulated in order to remove cuttings, to prevent cavein of the sidewalls of the wellbore, or to gain control of a blowing well. Other drilling fluids used in rotary drilling in geothermal areas are for example air mixed with water and a foaming agent or air alone. These fluids have not been used in Iceland.

## 10.3 Low temperature geothermal well design

The design of a typical low temperature geothermal well in Iceland is fairly straight forward. The initial drilling as pointed out above is done by a cable tool rig to a depth of 25-50 m and loose surface layers are cased off with the socalled conductor pipe. Rotary drilling then takes over. Drilling depth and well diameter vary from one area to another, and will also depend upon the size of the drilling rig, the expected flow rate of the well, and the size of deep well pump to be used.

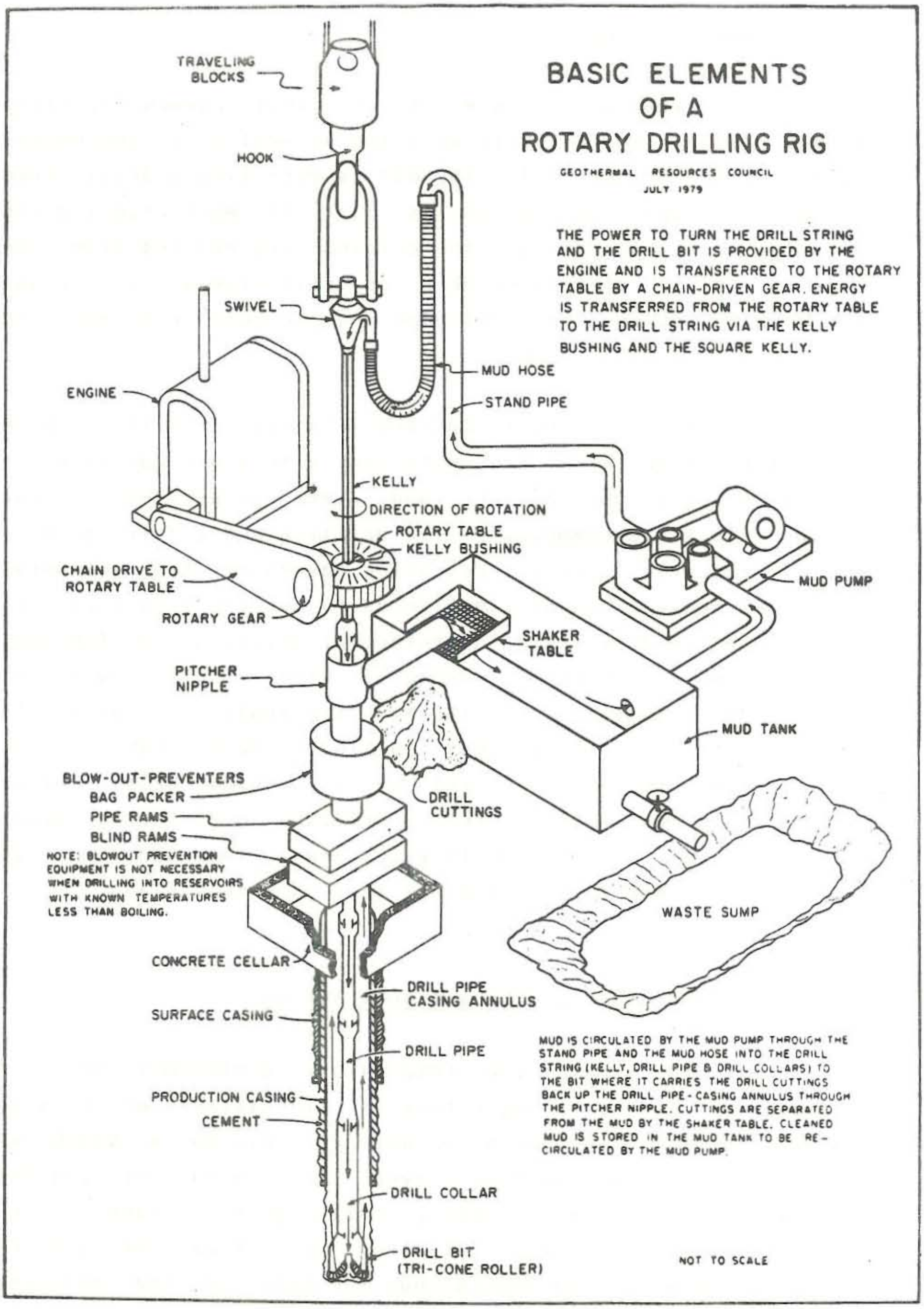


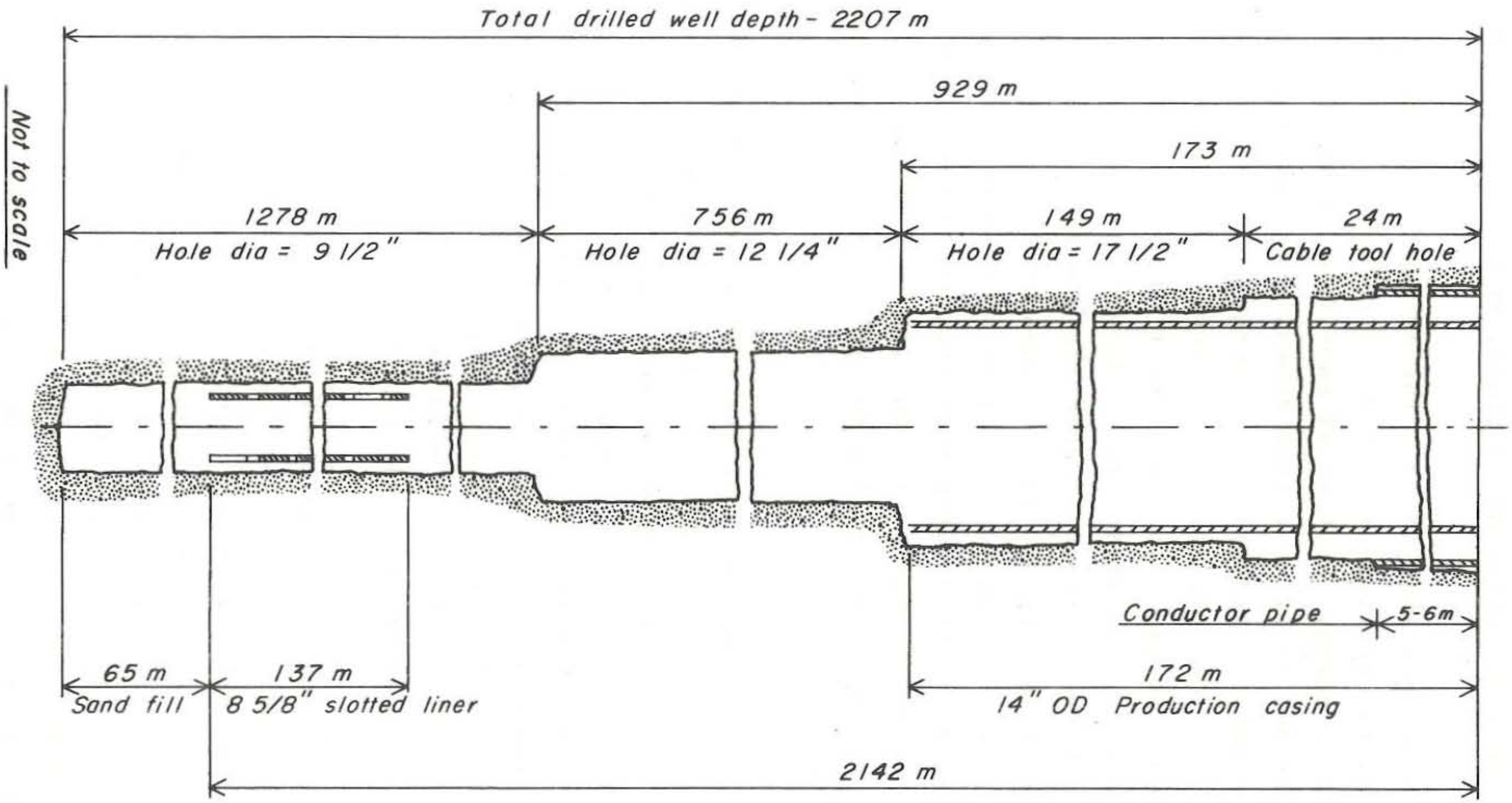

Figure 27. (From Anderson and Lund, 1979).

The production casing is normally set to a depth ranging anywhere from 150 to 400 m depending on the reservoir characteristics and expected drawdown during production. This production casing is cemented in the borehole from bottom to the surface. This protects the steel casing pipe from possible external corrosion and prevents the flow of surface water into the well. Such flow is undesirable as it could contaminate the geothermal water with oxygen and it would lower its temperature.

Below the production casing the wells are usually completed without casing or liners and the well production takes place through the open hole. The final depth of the well is determined by the results obtained during drilling, by the expected depth of the aquifer, and by the depth capacity of the drilling equipment. The final depth may also be affected by difficulties encountered during drilling such as collapse of the borehole and insufficient removal of the cuttings due to lost circulation, when the drilling fluid disappears through cracks and fissures in the formation instead of returning through the annulus up to the surface.

The use of injector packers for flow stimulation of completed wells has been used with good results in Iceland since 1967 (Tómasson and Thorsteinsson, 1975). During stimulation an inflatable packer is set between two or more producing horizons in the well, and water in turn injected beneath and above the packer. The rate of pumping is from 30 to 100 l/s and varies according to the resistance of the producing horizons to flow. Wellface pressures range from a few bars to 60-70 bars at 30 l/s in highly resistive horizons. This procedure has resulted in up to three- to fourfold increases in productivity of the wells. This increase in productivity is predominantly attributed to the reopening of producing horizons clogged by drill cuttings and lost circulation materials, but also to the removal of zeolite and calcite vein deposits thus increasing the permeability of the rock formations in the immediate vicinity of the wells.

Figure 28 shows a low temperature well recently completed in the Seltjarnarnes geothermal area. The injector packing caused caving of the well around a lost circulation zone encountered at 2094 m and a slotted liner was placed in the well around this area in order to prevent complete collapse of the well. The sand fill at the bottom of the well is caused by material from the caving location.




Drilling and casing design for a recently drilled hot water well

Fig. 28

## 11. ECONOMY OF GEOTHERMAL DISTRICT HEATING IN ICELAND

Statistics show that in 1979 the energy consumption in Iceland amounted to approximately 51,000 Terajoule (1 TJ = $10^{12}$ J). Of this total energy about 44.4% came from imported fossil fuel products and 55.6% were from domestic energy sources which was divided between hydroelectric energy, 17.6%, and geothermal energy, 38.0%. The trend in recent years is evidenced by the fact that the imported energy part as a percentage of the total energy consumption was reduced by 3.8% from the previous year with the increase in domestic energy coming exclusively from geothermal energy (percentages in 1978: 48.2, 18.0, 33.8 in same order as above (from Iceland Research Council, 1981)). It may safely be assumed that this trend has continued, since 1980 statistics show a 7% reduction in fuel imports, but the final energy figures for the year are not available at this time (June 1981).

By far the largest part of the energy consumption in Iceland goes to space heating or over 41%. In 1978 the space heating energy was divided between imported fuel, electricity and geothermal in the ratios 22.8%, 10.3%, and 66.9%, respectively. With the current trend the fossil fuel part is being reduced, and it is estimated that in about five years' time, fossil fuel space heating will be eliminated altogether and the space heating needs will then be met by geothermal energy, 80% and by electricity, 20%.

In Iceland there are now about 50 geothermal district heating systems operating in towns, villages and smaller communities utilizing heat from as many low and high temperature geothermal areas (Saemundsson, 1981). These systems vary greatly in size from the large Reykjavik District Heating Service to very small systems serving only a few families. It is estimated that at the end of 1980 over 75% of the Icelandic population enjoyed geothermal heat in their homes, and with district heating systems now under construction and on the drawing board, this percentage is expected to rise to 80%

within a few years.

Space heating in geothermal district heating systems in Iceland is very economical compared with other means of heating as shown by the following average unit energy prices as they were estimated for the year 1978 (from National Research Council, 1981):

| Energy source | Average price, kr./kWh | |
|---|---|---|
| Fuel oil | 6.38 | Average exchange |
| Electricity | 4.83 | rate in 1978: |
| Geothermal district heating | 1.07 | US$ 1.00=kr2.72 |

It is to be noted that the large 1979 oil price increase is not included in the above prices which means that today the geothermal district heating price is even more favourable.

The price ratio between geothermal heating and electric heating has stayed approximately the same, but both of these utilities have in recent years been under strict price control, which has led to prices somewhat lower than needed for normal operation and maintenance as claimed by the owners and operators. Even so it is evident that geothermal space heating is Iceland is on the whole very economical and a cost ratio of 1 to 4 between geothermal and fossil fuel burning is probably a reasonable estimate.

The old and well established Reykjavik Hot Water Supply Service accounts for about 70% of the thermal power of all Icelandic geothermal district heating systems. This Service together with a few of the early district heating systems, altogether serving at least 75% of the geothermal heating market in Iceland, are the most economical and these are responsible for the very favourable price of the geothermal heat. District heating systems built in recent years are less economical, ranging in cost from 30% to 80% of the cost of fuel oil heating.

By now practically all communities in Iceland, located in the vicinity of geothermal areas enjoy geothermal district heating or such systems are being built or planned. At this time geothermal investigations are under way in 10-15 communities with a total population of approximately 6,000, but they all have their problems, such as long supply line distance, low water temperature, and undesirable chemical contents of the geothermal water. Solutions to these problems are being investigated, but beyond that the prospects of further increase in the utlization of low temperature geothermal energy lie mainly in uses other than space heating, such as aquaculture, hay drying, and other light industrial uses.

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