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The Optimisation of District Heating Operating Temperatures and an Appraisal of the Benefits of Low Temperature District Heating

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Summary

One of the most fundamental decisions to be made by a designer of a new district heating system is the selection of design operating temperatures. Lower operating temperatures will reduce the cost of heat production from CHP plant but to achieve lower temperatures requires additional investment in the heating systems within the buildings and there is therefore a need to establish an optimum temperature selection to achieve the most cost-effective scheme.

In order to identify the optimum temperatures, a series of case studies based on notional groups of buildings were developed and comparative economic analyses produced. The case studies comprised combinations of:

- three types of built form: apartment blocks, row houses and commercial buildings;
- two climate types: London and Toronto;
- three types of CHP plant: steam turbine, combined cycle gas turbine and spark-ignition gas-engine.

The analysis of the heating systems within the buildings was carried out by CANMET who developed simulation models using the Simulink software. This enabled hour by hour heat demands and return temperatures to be calculated for each case. This heat demand pattern was then used as the basis for a spreadsheet model developed by Merz Orchard to simulate the operation of a CHP plant over the year and hence calculate the cost of heat production. Designs for the district heating network were produced using System RORNET by Merz Orchard and cost estimates made.

Finally, costs were obtained from manufacturers for district heating substations, radiators and air heating coils required within the buildings. All of these cost elements were calculated for a range of design operating temperatures, ranging from 90°C flow, 70°C return to 70°C flow 30°C return. The results are presented in a series of graphs of the cost of heat against design temperature difference for different flow temperature assumptions for each case study analysed.
It was found that in general it was not worthwhile to reduce the maximum flow temperature below 90°C although a reduction to 70°C during the summer period would be worthwhile for schemes supplied from steam turbine CHP plant. The only exception was the case where commercial buildings were to be supplied and where the higher network heat density and relatively low cost of air heating coils within the buildings' air handling units meant that a maximum flow temperature of 70°C could be more cost-effective.

For a peak flow temperature of 90°C the optimum temperature difference was found to be about 45°C in all cases, although the cost curves are relatively flat and a variation in return temperature of +/- 10°C about the optimum resulted in a cost variation of less than 1%. There are many other potential benefits for using lower temperatures, in particular the ability of the district heating network to utilise low grade heat sources available from industry, solar heat and heat pumps. The cost penalty from selecting a 70°C temperature instead of 90°C was found to result in an overall increase in the cost of heat of between 4% and 6% of a typical heat selling price. It is possible that in some circumstances this relatively small increase is justifiable given the potential environmental benefits in the longer term of maximising the use of waste heat and renewable energy by means of district heating.

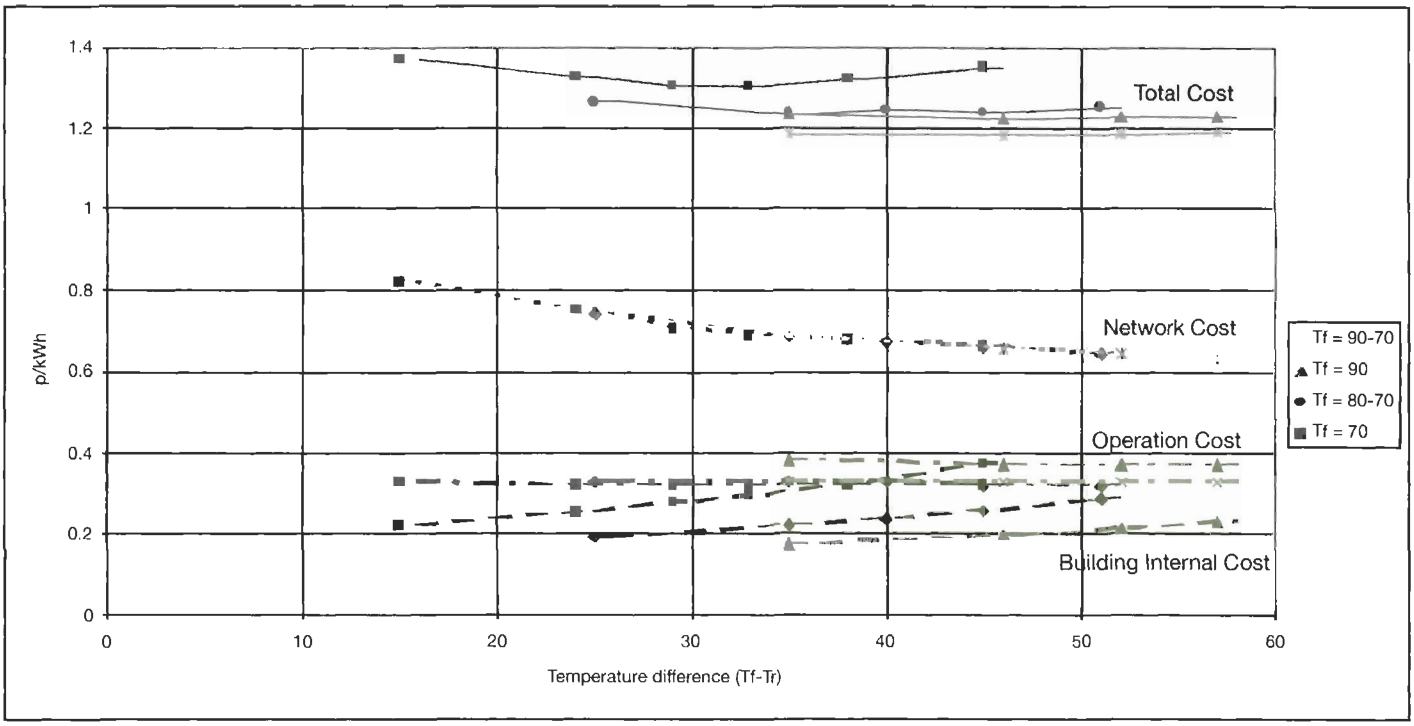

Figure 1 - The Cost of Heat for Apartment Blocks in London supplied from Steam Turbine CHP

Note: T_f = 90-70 means peak flow temperature is 90°C reducing to 70°C in summer, similarly T_f = 80-70 means peak flow temperature is 80°C reducing to 70°C in summer.

Combined Heating and Cooling - Balancing the Production and Demand
Executive Summary (Draft)

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1. Scope and Purpose

Under Annex IV a report with the title "Integrating District Cooling with Combined Heat and Power" was compiled and published in 1996 [1]. Its purpose was to provide guidance to system designers to identify the best options for integrating district cooling with combined heat and power (CHP) by describing the energy efficiency, economics and environmental implications of different alternatives. This work at hand is a follow-up project under the Annex V programme of the IEA. Its objective was to collect systematized information and operation experiences of existing district cooling systems connected to CHP plants, to investigate new promising cooling techniques and to produce optimal design and operating parameters for CHP connected cooling systems. Investigation of balancing the production and demand is concentrated on the production side. The key question in this report is whether more cooling output capacity and annual cooling energy can be drawn from CHP plants by using heat driven cooling techniques. If heat load can be increased in the summer, this also balances the annual production profile of CHP plants. On the demand side the electric air-conditioning peak is lowered, if electric cooling machines are replaced with heat driven units.

In this report district cooling is defined as any means of providing cooling energy for a larger area from a centralized production plant. Thus, cooling achieved by distributing hot water or steam for primary energy of heat-driven cooling machines located in individual buildings is also considered district cooling. In particular, the design and optimization problems are concentrated on district heat (hot water) driven absorption cooling technology, as this is seen as the fastest growing potential for CHP generated cooling - if proven competitive with the electric vapor compression technology.

The feasibility of heat driven cooling is in many cases a question investment cost and thermal performance of the chiller. High performance or energy efficiency saves costs related to primary energy and justifies the use of the technology by reducing emissions. Therefore, a great part of the report concentrates on characteristics and possibilities of heat driven cooling techniques.

The main topics in the report can be summarized as follows:

- Survey of techniques in existing district cooling systems in Germany
- Environmental impacts of cooling
- Performance study of a single-effect water-LiBr absorption chiller
- Energy efficiency of hot water driven absorption cooling
- Design considerations for district cooling systems
- Optimization of district heat driven absorption cooling systems
- Operating experiences of real district cooling systems
- Costs of different district cooling concepts
- Cooling techniques on the market and potential future techniques
- Prediction of cooling demand and market in the future

In the following the contents of the report and some findings are introduced.

2. Survey of Technologies

The operating principles and performance characteristics of different heat driven refrigerating processes are introduced.
The performance of compression chillers is discussed to provide a basis for comparison between processes. Also the cost factors are presented, if available.

A single-effect water-LiBr absorption chiller is discussed with a large number of details, because it is the most usual heat driven application on the present market. It is also the example application used in the optimization part of this report.

Several modifications of the simple single-effect absorption chiller are also presented. These advanced cycles are double-effect cycle, half-effect cycle, and single-effect/double-lift cycle. Also, such modifications as spray-type generator absorption chillers and so-called "mix-match" absorption chillers are introduced. The main goal in modifications is typically the utilization of lower driving heat temperatures and increasing the heating water temperature difference.

Other applications are water-ammonia absorption chillers, desiccative and evaporative cooling, water-silica gel adsorption chillers and steam-jet cycle. Some of the applications are in pilot or experimental stages.

2.1 Estimation of Potential Future Techniques
It is estimated that the ongoing tendency in development of water-LiBr absorption chillers to utilize lower supply temperatures and to achieve greater hot water temperature differences is likely to continue increasingly. The future cooling market is strongly dependent on the new refrigerants for conventional chillers. The lack of substitute refrigerants for compression chillers and the uncertain situation on the market makes absorption chillers even more attractive. The chiller technology for utilizing district heating hot water will be of interest.

In this light, adsorption chillers also have potential. The main advantage is their ability to use low supply temperatures, even as low as 55°C. The high investment costs and lack of operating experience slow down their becoming more common.

3. Environmental Impacts of Cooling

The present state of environmental protection development regarding cooling and refrigeration systems is reviewed. This matter is closely related to refrigerant phase-out schedules. CFCs (R11, R12, etc.) greatly depleting the ozone layer and contributing to global warming have practically disappeared from the market. HCFCs (R22, R123, etc.), which also contribute to the ozone depletion and global warming, are scheduled for phase-out in most of the countries by the year 2030. Several substitutes for all these ozone depleting refrigerants have been developed. The most promising ones are HFCs (R134a, R407c, etc.), which do not deplete the ozone layer but are still greenhouse gases. There is also newly awakened interest towards so-called natural refrigerants as ammonia and water for absorption as well as vapor compression applications.

As the refrigerants become harmless to the ozone layer in the future, the Total Equivalent Warming Impact (TEWI) is going to be more important in deciding between cooling system options. The only global environmental impact of cooling will be the contribution to global warming. The TEWI takes into account both over the life cycle of the chiller: the global warming potential of the refrigerant discharge and the emissions caused by the primary energy consumption for cooling duty. Research on TEWI has shown that, for most applications, a greater quantity of greenhouse gases is released into the atmosphere from energy consumption than from refrigerant discharge. Therefore, in calculating TEWI values energy efficiency analysis is going to play a major role in the future. Of course, other environmental and economic aspects of the cooling options and refrigerants must not be forgotten. Briefly, these aspects related to refrigerants are thermal properties and others as toxicity and flammability.

4. Performance of Single-Effect Water/ Lithium Bromide Absorption Chiller

The performance of a single-effect water/lithium bromide absorption chiller is investigated thoroughly. The basis of this investigation is a simulation model, the results of which are first compared to a real chiller. The simulation model is also used throughout the research for dimensioning the absorption chiller unit and for optimizing the mutual performance in connection with the cooling tower.

The results of this section show that there is a theoretical potential to improve the coefficient of performance (COP) by optimizing the internal flow rate of the water/lithium bromide solution. It does not help to solve the capacity related problems at low driving temperatures nor it does improve the COP at full load. According to the simulations, at part-load and/or with low driving temperatures the COP can even exceed that of the full load. The heat exchanger area distribution of the chiller could also be optimized for a specific application in a certain temperature range, but the sensitivity to the distribution is not crucial as long as the effective area is somewhat "balanced", as shown in the report.

Controlling of the conventional absorption chiller at part load is demonstrated with an example. It shows that the COP does not drop very much below 0.7 at part load if relief in the form of decreasing cooling water temperature is provided. The example also shows some of the complexity behind the absorption chiller part-load control. This provides understanding for optimization possibilities in mutual controlling of a chiller, cooling tower and an air-conditioning device.

5. Energy Efficiency of Hot Water Driven Absorption Cooling

It was concluded in [1] that heat driven absorption cooling is not energy-efficient compared to electrical compression technology if the driving heat is high-temperature heat coming from a boiler. If, however, heat is degraded for a suitable temperature level, for instance in a CHP facility steam turbine, the energy efficiency of absorption systems could even exceed that of electrical compression systems.

This question was investigated with a realistic example by simulating a steam extraction CHP plant and taking account of electricity lost to heat generation for absorption chillers.
The CHP facility was assumed to provide normal heat customers with district heat at temperatures of 90/60°C in the cooling season, and then to increase heat output to serve the chillers. Heat losses, electric power transmission losses and auxiliary power need of the whole absorption chiller plant compared to a compression chiller plant were taken into account. With this kind of a procedure the loss of electricity due to absorption cooling can be thought as the electric power consumption of absorption technology. The procedure results in a COP (W_{ch}/W_e) for the absorption chiller that is fully comparable to the COP of the compression chiller.

A thermal COP of 0.70 and auxiliary excess power consumption compared to a compression system of 0.037 W_{ch}/W_e were assumed. A condenser water inlet temperature of 25°C was chosen for the CHP plant.
The example with a highly efficient steam extraction CHP plant represents a case in which the loss of electricity is very high, and hence more underestimating than overestimating the efficiency of the absorption technology. The steam extraction plant is also a relevant example to think of the electricity that could be obtained by installing a condensing tail in a back-pressure plant.
In that case, however, using heat for absorption cooling saves the costs related to installing the condensing tail.

The results show a COP of 0.3...6.9 for the absorption chiller depending on the case.
The lowest COP values were obtained when the district heat supply temperature was raised to 110°C instead of the normal 90°C, in order to provide the conditions of high cooling output. The highest values of the COP (5.6...6.9) resulted from a case in which the supply temperature was the normal 90°C and the absorption chiller was thought to run at part load and capable of $\Delta T = 20°C$ in the district heating network. The variations in the COP within the same case are due to variations in the amount of heat extracted, depending on the cooling load.

The average COP of the compression chiller over the cooling season can be estimated to be roughly 5.2, or 6.0 at its maximum.
The results show that the energy efficiency of the district heat driven absorption chiller can be better than that, but the difference in the example was not great. Therefore, the idea that hot water driven absorption cooling saves considerable amounts of primary energy is not necessarily true in every case. The energy efficiency of such a system is highly case specific and depends on the following parameters:
CHP plant design, heat extraction pressure, CHP plant condenser pressure (if there is a condensing tail) and the state of reference for electricity production. Also, if district heat supply temperature has to be raised at the time of cooling peak, the power production capacity is reduced and, consequently, the balancing effect is also reduced.

6. Design Considerations for Cooling Systems

The problems in the field district cooling are discussed. Design considerations for cooling systems are reviewed by presenting several performance curves for compression and absorption chillers and cooling towers. Also some cost curves related to these machines and auxiliaries are shown. This section provides information that is essential for understanding the questions dealt with later on in the report and gives general information.

One important design consideration for district heat driven absorption cooling arises from the small ΔT of the driving hot water. The absorption chiller heat load in the network raises the return temperature to the CHP plant and therefore reduces the electricity production. If the transmission pipeline is designed for winter time operation temperatures of 120/60°C and the normal 10% heat load in the summer uses temperatures of 90/60°C, the connected absorption cooling load is restricted below 15% of the normal connected heat customers' load. The restriction is due to small ΔT in the driving hot water of the absorption chiller, COP of roughly 0.7 and increased district heat water pumping requirement if the dimensioning water flow is exceeded. In any case, the house connection pipeline has to be enlarged for the water demand of the absorption chiller, and perhaps also some of the branch lines.

It has been suggested that the chiller could be connected in series to the supply line, referred to as supply-supply connection in the report. Also, this connection method sets some restrictions to the system and/or connected cooling load. It is advantageous for the supply-supply connection method if the chillers are located at the first in the network. In this way, the chillers could enjoy higher supply temperature than the rest of the customers. If the supply-supply connected chiller load is too high in relation to the normal heat load in the network after it, the return temperature to the CHP plant increases from the normal as it does in the supply-return connection method.

7. Optimization of District Heat Driven Absorption Systems

The optimization in this report concentrates on district heat driven absorption systems located in customers' buildings. To cover the make-up water costs and the most probably higher being investment costs of the absorption system compared to the compression system, savings must be earned in the costs of primary energy and perhaps maintenance. The conditions in which the hot water driven absorption technology can be feasible are those with a high ratio of electricity price to heat price. This is the case if the heat is practically waste heat, the energy efficiency of heat driven absorption cooling is very high or the heat comes from a back-pressure CHP plant, which is able to make the more electricity, the higher the heat load. The latter case is particularly interesting, if the electricity peak load coincides with the peak air-conditioning cooling demand.

The concept with a back-pressure plant is investigated to find out the potential of generating more electricity with the heat load of absorption machines. Such a co-generation system is optimized from an economic perspective. The main parameters in question are the sizing of the chiller and the cooling tower, and whether the district heat supply temperature should be raised above normal at high cooling load. Namely, when dimensioning the absorption chiller at high hot water supply temperature, the capital costs can be significantly reduced. This part of the project is in a draft phase at the time of writing this summary.

8. Operating Experiences

8.1 Measurements on Low-temperature District Heat Driven Absorption Chiller

Three series of measurements with a district heat driven single-effect water/lithium bromide absorption chiller were performed in Seoul, Korea, at the end of summer 1997. The purpose of the measurements was to record normal operation and to experiment the effects of applying relatively low driving hot water temperatures. The design temperatures of the system are 95/80°C, 31/36.5°C and 12/7°C for driving hot, cooling and chilled water inlet/outlet, respectively. The cooling capacity of the machine is 100 kW_{ch}. The cooling and chilled water flows are fixed at 28.8 m^3/h and 46.1 m^3/h, respectively.

Unfortunately, the measurements contained a systematic error, the source of which could not be identified afterwards. Therefore, exact conclusions of the COP, for instance, cannot be drawn. However, the tests with a step change in the hot water inlet temperature show that the machine finds a new quasi-steady-state in about 60 minutes after a 12°C change. Lowering the hot water inlet temperature caused the hot water ΔT to be smaller and chilled water temperatures to be higher than normal. In general, it proved to be difficult to perform steady-state measurements on a real chiller that is serving a real, variable cooling load.

8.2 Operating Experiences in Germany
In Germany, 170 MW of absorption chiller cooling capacity connected to CHP was installed by January 1996. The markets were investigated with questionnaires sent to 20 cooling plants with a total capacity of 61 MW.

Some 90% of CHP-connected absorption chillers are water-LiBr absorption chillers. About 70% of existing cooling plant capacity is used for air conditioning, the rest for combined use (process cooling and air conditioning). Only 8% of the capacity is provided with two-stage absorption chillers. One-stage absorption chiller unit sizes vary in the range of 0.25…5 MW_{ch}.

The major heat source is the engine-CHP, which covers nearly 50% of the cooling capacity (hot water and steam). The combined cycle and the gas turbine both separately cover 23% of the capacity. Absorption chillers connected to district heating systems have been established since the early 1990s, and their share is increasing. About 65% of the capacity is provided in plants with an output higher than 5 MW. The driving heat temperatures vary widely. Thermal storing in cold water tanks or ice storage has not been used in Germany.

The operating experience in Germany indicates that

- The best operating temperature is found to be over 100°C. Despite this, many absorption chillers are operated with hot water of 85-90°C.
- Absorption chiller part-load performance is stable and high-efficient even at 10% of full output.
- There were only few malfunctions during operation, and most of the failures were due to untrained staff. When the staff is well trained, long maintenance intervals are possible for the absorption chillers.
- Generally, the importance of documentation of operation and maintenance and training the staff were underestimated by many operators, which led to problems.
- The operation of cooling towers is sensitive to lime accumulation and algae growth, which leads to failures and shut-downs. Thus, the water treatment should not be underestimated.

8.3 Costs of Cold Generation
The cost factors for different cooling unit types are presented in the technical presentations of the report. They also include investment and operation cost calculation examples.

The investment costs of single effect water-LiBr absorption chiller plants are found approximately 14% higher than those of compression chiller plants. Respectively, the investment costs of two-stage water-LiBr absorption chiller plants are 54% higher. When comparing one-stage and two-stage absorption chiller plants, the investment in a two-stage chiller plant is 35% higher than that in a one-stage absorption chiller plant. All the costs depend strongly on the unit size, as can be seen from Figure 1.

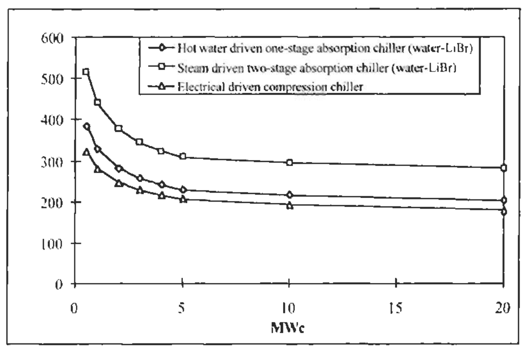

Figure 1. Capital costs of cooling plants.

The annuity of the investment costs is typically 20-35% of the total annual costs of cold generation.

The temperature of the driving heat has a great effect on investment costs. For example, in one case an increase of 10°C (80 → 90°C) in heat supply temperature reduced the investment costs 30%, and the second increase (90 → 100°C) another 20%. The effect on operating costs is much smaller, roughly 5%.

Some notes about costs in general:

- The water treatment costs are an essential cost factor
- The piping need varies strongly from case to

case and has a quite significant effect on capital costs

The report also includes four different scenarios for generating 5000 kW cooling energy. Four means of cold generation were presented and the total costs were compared to those of conventional individual compression chillers. The means are:

1. Local water-LiBr-chillers supplied by existing district heating network
2. Individual chillers (single effect/double lift absorption chiller and mix-match chiller) supplied by the existing DH network
3. Individual chillers (water-silica-gel and DEC-process) supplied by the existing DH network
4. Water-ammonia absorption chiller, ice storage and distribution by ice slurry

Single-effect/double lift absorption chillers, mix-match absorption chillers and adsorption chillers are not competitive compared to individual compression chillers. The DEC-process represented to be the most feasible refrigerating process. However, it can be considered only in context of air conditioning of new buildings, because it generates only cold air, not water.

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1 Mark Spurr, Ingvar Larsson, *Integrating District Cooling with Combined Heat and Power*, International Energy Agency (IEA), Programme of Research, Development and Demonstration on District Heating and Cooling, Novem, Netherlands, 1996.

Cost Effective DH&C Networks

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1. Introduction

In today's construction of district heating networks the preinsulated pipes are the standard material. The quality of the components are assured by CEN standards and also the design methods are well proven. Since this technique seems to be mature and well developed, the high costs for building networks still demand new developments for cheaper installation. In the ongoing project three topics are dealt with which promise economical improvements. These attempts refer on piping technology and construction of the pipe trenches as well. It is the aim of the project to analyse new developments in pipeline construction. The project has been started in September 1997.

The following three tasks will be treated:
1. Cold installation of pre-insulated pipes
2. New ways of installing pipes
3. Reuse of excavated material

Work is carried out in international cooperation with a Finnish consultant, EKONO on task 2 and Chalmers University, Sweden on task 3.

2. Cold Installation of bonded preinsulated pipes

Of all possible options cold pipe installation is the simplest kind of laying technology which in its advanced form opens new and effective possibilities of construction site organization. With appropriate preparation installation can be performed in a single day construction manner.

With today's practice of cold pipe installation the design stress parameters are not any longer limited by the material's 0.2 % yield strain. The design stresses calculated for cold pipe installation exeed the yield strength. Static design is governed by a fatigue analysis based on the appropriate number of load cycles for district heating. Hence, the material's mechanical stress resistance is used to an extensively high degree.

With cold pipe installation resulting stresses can reach the stress of actual yield strain. As a consequence, the pipe endings which are generally located in the expansion zone undergo considerable displacements. Solutions have been worked out to control the high stresses and strains. These are described in the report on the project. There is shown how conventional methods of compensation could be used in an adjusted technique and what new ways of compensating were developed. In general, operating the system with moderate temperatures cuts back some of the disadvantages and restrictions of the cold pipe installation method.

The method of cold pipe installation is definitely applicable today. The decisive questions concerning material stress are solved both theoretically and experimentally. An increasing number of companies does cold pipe installation using several years of experience particularly those made in Denmark, Sweden and Germany. Nevertheless there are companies who have concerns on this extended design method as they hesitate to reduce their hidden safety margins. But from the point of statics the new design is safe and fully justified.

From point of view for future research the experiments of measuring the real movements of pipes in the soil should be mentioned. It is known from theory and operation that movement of a pipe decreases with time. But there are very little experimental data about the real displacements available. Therefore displacements of pipes were measured at pipes under operation for several years. As very interesting result it turned out that motion appears only during the first heating period, even during the first few months of operation at reasonable temperatures. That means after some months of operation pipes don't move any more, or only very little. The consequence for later connections to a line could be that there isn't any (or only reduced) need for means of compensation. This looks promising referring to economy. Unfortunately we are not sure up to now that our measurements are valid at any case. Further investigations would be very interesting and could be helpful to achieve further savings.

The predominant advantage of cold pipe installation is the reduction of construction time and costs. Extensive investigations of the time span of construction have shown that construction time decreases to about 70% and costs to about 80% when compared to pipe laying by ways of thermal pre-stressing.

The results of this investigation are put together in the final report which will be published within some months. A draft version of the report is available and can be looked at on the poster presentation.

3. New ways of installing pipes

The proven installation of the outgoing and return pipes side by side is supplemented by two new arrangements today. Both are introduced for lowering installation costs. One is the twin pipe system which is mainly used in Finland and only little in other countries up to now and the second is the laying of the two pipes on top to one another, a German development. Both systems need smaller trenches. They have parallels with respect to compensation.The twin pipes use the return pipe by fixing it to the outgoing line for compensation; this might also be done with vertically arranged pipes.

In Finland the twin pipe system plays an important role in the lower diameter range. It is produced in dimensions up to DN 200. For DN 20 the share is about 70% of the total installed length, for DN 125 it is still about 25 %, but it is used only little above this diameter. The twinpipes are installed with preheating and don't need any compensation. With one jacket pipe for both pipes they have only one half of the jacket pipe connections; they need less material for the PE casing and insulation and they have lower thermal losses. The supposed increased needs for mounting seem to be not high and can be neglected as experts can prove.

In Finland for special purposes experiments were carried out with prestressing of twin pipes during production process (1993/1994). The prestressing is secured by temporary anchors which are cut out after installation. The advantage is that no preheating is needed on the building site and trenches can be refilled immediately after installation of the pipes. So the construction time is reduced and the disturbances to the traffic and to the public are minimized. Nevertheless this method is used only on a smaller scale in comparison to preheating.

The piggy back laying of pipes is used more and more in Germany. This installation method was developed within the last years and is incorporated into the German design rules for DH lines meanwhile [AGFW FW 401].The advantages are first that a smaller trench is needed and less road surface has to be restored, second that branches can be installed laterally to the main line so that both can be layed with the same minimum coverage and third that standard single pipes can be used.

An interesting part of the work was to calculate cost effects of both installation methods so that they can be compared objectively. EKONO Energy made calculations for the Finnish way of laying and MVV calculated the costs for Germany where construction costs are known to be high.

The main results are:

- In Finland the twin pipes are cheapest. (Savings about 25 %) Piggy back laying is only slightly cheaper than conventional laying; savings of 5 to 10 % can be achieved.
- In Germany with twin pipes and with piggy back laying savings are of a similar order of about 20 %. It seems to be necessary to excavate trenches with vertical walls and to support them as long as they have to be kept open.

The results of the calculation with the exact values will be shown in the poster presentation. There can also be seen under which consumptions the calculations were carried out: In Finland pipelines are installed in trenches with inclined walls and in Germany with vertical walls. In the Finnish comparison the twin pipe was calculated with a thinner sand protecting layer than the vertical arrangement. The calculation was done for special lengths for variant diameters. The German comparison was carried out for a medium diameter (DN 80) up to now and different cross sections of the trenches.

It can be summed up that with both laying methods new DH lines can be built considerably cheaper than by installing them conventionally side by side.

4. Reuse of excavated material

The most desirable material for backfilling pipeline trenches is

- excavated soil
- recycled material or overburden
- excavated material or overburden after processing

Of course it should be preferred to avoid processing at all, but as long as long term material properties are unknown it can be advantageous to use processed material instead of virgin.

Two questions will be treated in the study:
1. material properties (mainly long term friction behaviour) and their changes with time
2. damages of PE casing by stones.

The benefits of reusing material is obvious. Cost savings result from avoided transportation, no or less new material is needed and also no or less dumping. The positive environmental effects are less use of natural resources and less harm to environment. - Processed material may have the additional advantage of lower masses to be excavated and less road surface to be restored. The trenches can be smaller as processed backfill can be poured into the ditch like a liquid. Thus laying personal hasn't to enter the trench. Processing means normally sifting out big stones and mixing up additives. In few cases just washing in the backfill with water can be sufficient.

From Sweden investigations on the damage of the jacket pipe by coarse material will be contributed. There will be reported the impacts of stones on the PE casing which was tested under heavy traffic loads. The precautions which should be taken using coarser material than usual will be described. With Chalmers University it is not yet definitly fixed what alltogether will be contributed. At the moment it is checked if some special experiments on processed material (measurement of compaction and friction) can be added.
Work on this task is in good progress but it is not yet in the final stage.

5. Acknowledgements

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- Chalmers University/SP provning Forskning, Gothenburg, Sweden.

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- Veili-Pekka Sirola, Finland.

Mannheim
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Dr.-Ing. Frieder Schmitt

Figure 1: Graph of the measurements for the 1. and 2. heating-season.

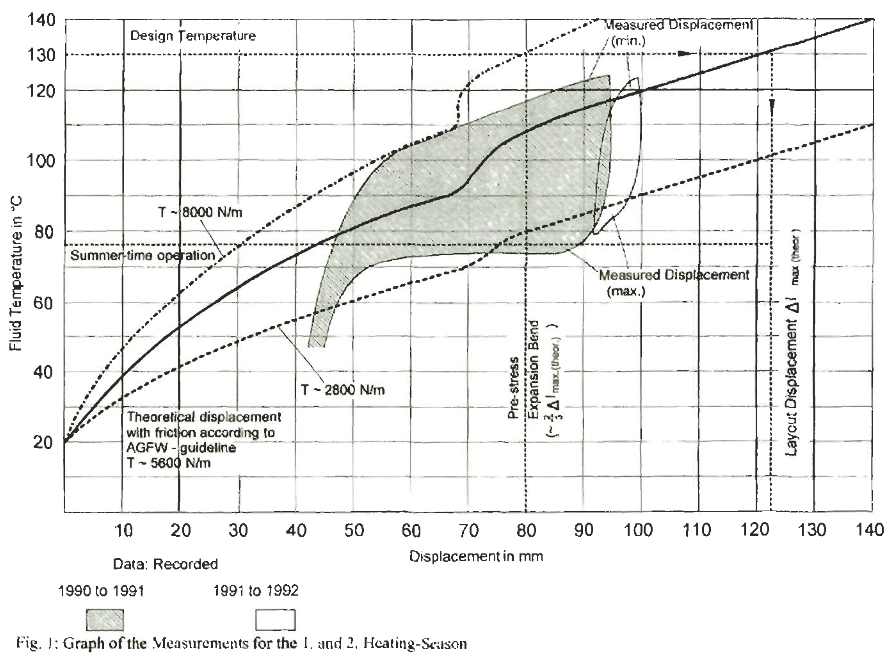

Fig. 1: Graph of the Measurements for the 1. and 2. Heating-Season

Fatigue Analyses of District Heating Systems
Executive Summary (draft)

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Nils Olsson

Introduction

This project is divided in two parts:
1. A practical part with temperature measurements
2. A theoretical part dealing with design model and calculations.

Practical part

The practical part was a continuation of a project under IEA District Heating and Cooling, Annex IV, named: Temperature Variations in Preinsulated DH Pipes, Low Cycle Fatigue, ref. [2]. In that temperature variations were measured at 17 district heating sites in Denmark, Germany, Korea, The Netherlands and Sweden.

In this project measurements were made by the Korean District Heating Corporation with the equipment used in the Annex IV project (4 units) at locations chosen by the Korean District Heating Corporation. The measurements at the four new locations lasted for one full year.

The data was sorted by the rain-flow method and matrixes of temperature variation and graphs are produced in accordance with the data processing done in the Annex IV project.

The data processing was done by Lund's Institute of Technology, Sweden.

Theoretical part

The theoretical part is based on the design model in ref. [1]: Draft standard, which is based on the hot-spot method.

Based on this method a limited number of details of preinsulated bonded pipe systems are analysed. The details include:
- A house connection, where the tee piece is the critical part
- Some 90° bends
- Bevel welds (small changes of direction up to e.g. 5°)

Background

The development of preinsulated pipe systems for district heating has for quite some time been characterised by simplification of laying methods, thus employing cold-laying or pre-stressed systems instead of using expansion facilities like compensators and U-bends, giving more robust and cost-effective systems. The simplified laying methods on the other hand give rise to higher stress and strain in the system, and therefore suitable calculation methods have been developed in order that the full potential of the systems can be utilised. This development has, e.g., taken place in an technical committee, TC 107 under the European Standardisation Organisation, CEN. The result, a Draft Standard for the Design and Installation of Preinsulated Bonded Pipes for District Heating, is presently being prepared for enquiry [1].

When the stress range is larger than twice the yield stress, the system is said to be in the low cycle fatigue range. During the work with the draft standard, it became clear that the most important limit state for preinsulated bonded pipes is low cycle fatigue. In this limit state, the temperature variations are the most decisive action.

On this background, the measuring project in Annex IV was implemented with the purpose to register the number of temperature variations (at 17 sites). In this project the measuring program was extended with 4 new sites in Korea. Furthermore, this project deals with the whole concept for calculation in the low cycle fatigue limit state to give a general view of the method and to give user-friendly examples for calculations in this limit state.

Results

Temperature measurements
In this project temperature measurements were made on 4 points numbered 18 to 21 in continuation of the 17 points in Annex IV. Point 19 is a commercial building with district cooling, the other sites are blocks with apartments as follows:

- Number 18: 208 apartments
- Number 20: 408 apartments
- Number 21: 690 apartments

All measuring points are placed in substations on the primary side, which means on the district heating side of the installation and on top and underside of the pipes.

The temperature variations are transformed to full equivalent temperature cycles with a full temperature variation at 110°C. The results for the 4 sites are shown in enclosure 1.

The results for point 19 and 21 are in the same range as the results in Annex IV while the results for point 18 and 20 for the return pipe gives much higher values than we have seen in Annex IV. The maximum value in Annex IV was about 1,000 cycles by b=3 (b is the slope of the SN-curve) for the return pipes by the consumer. For point 18 and 20 the corresponding values are about 3,000 cycles.

With respect to new results, a new table for all the results in Annexes IV and V have been worked out. The table is shown below. The results for production (main pipes) are on changes, while all the new points are placed at consumers (service pipe).

Table 1
Numbers of full temperature cycles for ΔT_{ref} = 110°C and b=3, 4 and 5 based on results in this project and in the project in Annex IV, ref. [2].

Supply Production	Min.	Average	Max.
b=3	17	136	365
b=4	4	42	102
b=5	1	18	37

Return Production	Min.	Average	Max.
b=3	2	7	14
b=4	0	1	1
b=5	0	0	1

Supply Consumer	Min.	Average	Max.
b=3	7	130	578
b=4	2	51	308
b=5	1	28	197

Return Consumer	Min.	Average	Max.
b=3	35	667	3262
b=4	4	174	819
b=5	1	55	245

By analysing the results from measuring on top and bottom of the pipes, it can be concluded, that there is only small differences on the supply pipe. On the return pipe there are differences which go up to 25°C and with some single values of 40°C. Especially one consumer has big differences in the summertime. The differences mean very little for the fatigue analyses, but the differences may have effect on temperature measuring in connection with operation and energy measuring systems.

The commercial building with district cooling has an average number of temperature cycles, but the level for the return temperature is in the range from 60°C to 100°C with an average of about 80°C. This level is 10-20°C higher than the other consumers.

Maximum lines for temperature history
In this project there has been developed curves which show the "maximum line" for the temperature historic. The lines are shown in enclosure 2 and 3. The formulas for the curves are as follows:

- Supply pipes:
  $n = 2x10^{6} (\frac{1}{\Delta T})^{2.6}$
- Return pipes:
  $n = e^{(12-0.123 \Delta T)}$

N is number of cycles during 30 years for ΔT = 1, 2. 3 ... °C.

where
n (ΔT = 1) means all cycles for $0 < \Delta T \leq 1$°C
n (ΔT = 2) means all cycles for 1°C $< \Delta T \leq$ 2°C
etc.

Example:
If we set in 50 in the formula for return pipes, we get n = 347, this means that over 30 years we can expect 347 temperature variations in the range from 50-51°C. If we use the formula for supply pipes, instead we get 77 expected temperature variations.

Design model
The design model is based on suggestion by CEN/TC/JWG1. The design model is discussed in this project.

A conservative conclusion based on the results of the present project would be:
1. The present design method as suggested by CEN/TC/JWG1 [1] is maintained:

- N_0 is calculated from temp. history
- The Palmgren-Miner rule applies
- $\Delta\sigma = c\,\Delta T$
- Von Mises or Tresca for multiaxial stress/strain state
- SN-curve is used for lifetime estimation

2. Improved design methods must be based on fracture mechanics and stress history.
3. Fatigue life must be characterised by temperature history, not by full temperature cycles.
4. Small temperature cycles e.g. $\Delta T < 20^\circ C$ can be ignored.
5. Modelling of pipe-soil interaction must be improved, specially p-y diagrams in areas with road surface.
6. Hot-spot stresses based on FEM-methods versus experimental stresses based on the experiments made by Markl
7. Is the calculation of reference stresses for multi-axial stress-strain states correct with respect to the low cycle fatigue range?
8. A "flat" SN-curve ($b \geq 4$) is more likely to represent the true conditions rather than a "steep" SN-curve ($b \leq 3$)
9. Should a curve for un-welded material be included? The calculation examples might suggest that a higher SN-curve can be applied for un-welded material.

   However, insufficient modelling of soil reactions, the transformation from multi-axial stress state to reference stress and the lack of reduction factor for electro-chemical environmental actions indicate that the designer should be cautious applying a higher limit.

The uncertainties in the present design methods are large concerning:

1. Actions (temperature history and p-y diagrams)
2. Modelling
3. Stress concentration factors
4. Choice of SN-curves including the effect of electro-chemical environment.

Alternative design approaches could be:

1. Temperature variations must be monitored (specially at the consumers) and controlled at a low level (N_0 < ??). If N_0 is chosen sufficiently low cycle fatigue design will be unnecessary. This approach will be possible in systems equipped with "intelligent" heat meters presently under development.

Or

2. For T_{design} < e.g. 90°C design is done according to company standards and low cycle fatigue design is unnecessary.
   For T_{design} > e.g. 90°C cavities are established at all expansions and all tees are chosen according to DIN 2615 Reihe 4. With this approach it might also be possible to develop standardised solutions in order that low cycle fatigue design becomes unnecessary.

Calculation examples

In the report calculations are made on bends, tees and bevel welds. the calculations are based on draft standard CEN TC107/TC267/JWG1, ref. [1]. The calculations on bevel welds shown that there are no fatigue problems with respect to the calculations in the report. But bevel welds can give problems with buckling by cold laid systems where the second order effect can give an important action.

There has been calculated examples with bends and tees. The analyses for bends show that we may have problems with the bend while a normal bend in dimension DN 150 only can allow 196 load cycles ($\Delta T = 110^\circ C$) without safety factor. The possibility to get a stronger bend is to choose a thicker steel wall, a foam cushion or concrete duct. This gives the following load cycles for a DN 150 pipe: 710 cycles for steel wall thickness from 4 to 7.1 mm, 704 cycles for foam cushions and 3587 cycles for concrete duct. The results are shown in enclosure 4 and 5. It is very clear from the calculations that a concrete duct is effective for lowering the stress level and as a consequence of this, it can allow more load cycles.

The calculations on tees show that the right type of a tee can give very high numbers of cycles. For example a normal extruded T-piece DN 200/DN 80 with normal wall thickness can allow 58 load cycles without safety factor. But if we instead choose a DIN weld-in T-piece, Reihe 4, the cycles will be added to 7469 cycle ($\Delta T = 110^\circ C$).

The calculations for tees are shown in enclosure 6 and 7.

Further studies

1. At measuring sites R12, R18 and R20 fatigue failure have been recorded. However, none of these sites have had a life time of 30 years in spite of the fact that the number of full temperature cycles have been calculated to be in range normally assumed by design. For these sites it might be interesting to establish the actual stress history. Alternatively it could be considered if the rather large number of temperature cycles can have caused micro cracking, which due to the electro-chemical environment (the pH-value of the water) causes stress crack corrosion (SCC) earlier than expected by the usual design approach.
2. Calculating the reference stress at a multi-axial stress state by using von Mises or Tresca. The possible error thus introduced might explain the difference in the SN-curves used and the lower experimental curve established by Markl.
3. Improved modelling of the pipe-soil interaction under road surfaces.

Authors

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References

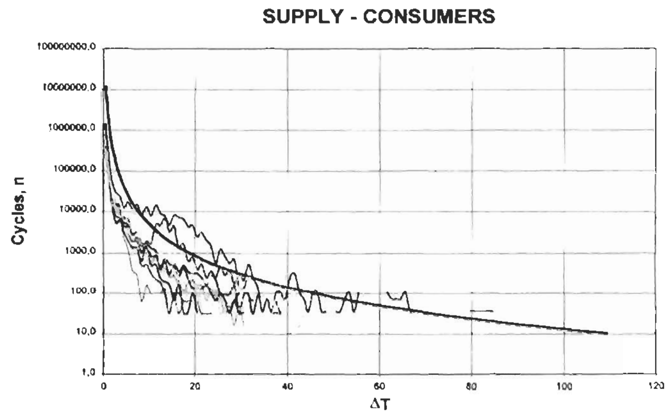
[1] Draft standard CEN TC107/TC267/JWG1, Design and Installation of Preinsulated bonded Pipes for District Heating, September 1997

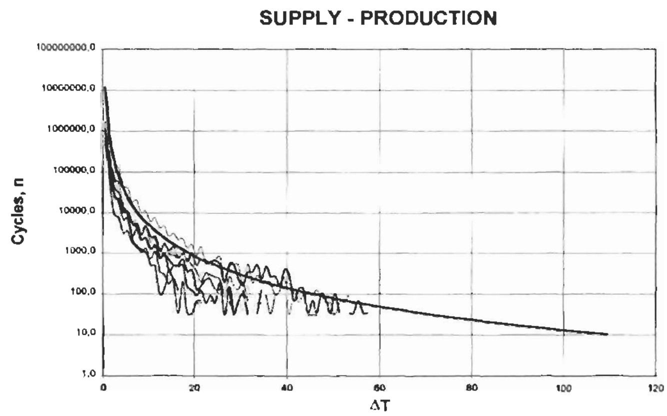
[2] IEA District Heating and Cooling. Temperature Variations in Preinsulated DH Pipes, Low Cycle Fatigue. Peter Randløv, Karl Erik Hansen, RAMBØLL Marie Penderos, Lunds Institute of Technology NOVEM July 1996. ISBN 90-72130-97-9

Enclosure 1

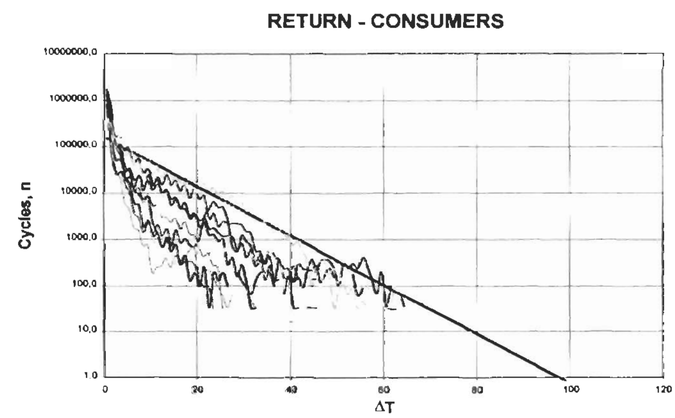
Location:	b	Measuring period days	Nr. of full cycles. in 30 years DT_{ref} = 110°C
S18 Top	3,0	354,6	43,4
S18 Top	4,0	354,6	20,5
S18 Top	5,0	354,6	14,1
S18 Under	3,0	354,6	43,1
S18 Under	4,0	354,6	20,5
S18 Under	5,0	354,6	14,1
R18 Top	3,0	346,7	2952,9
R18 Top	4,0	346,7	747,2
R18 Top	5,0	346,7	216,8
R18 Under	3,0	354,6	2815,2
R18 Under	4,0	354,6	706,1
R18 Under	5,0	354,6	203,2
S19 Top	3,0	346,2	180,9
S19 Top	4,0	346,2	59,7
S19 Top	5,0	346,2	25,2
S19 Under	3,0	327,4	189,0
S19 Under	4,0	327,4	63,0
S19 Under	5,0	327,4	26,7
R19 Top	3,0	346,2	474,2
R19 Top	4,0	346,2	143,4
R19 Top	5,0	346,2	53,3
R19 Under	3,0	346,2	494,6
R19 Under	4,0	346,2	149,1
R19 Under	5,0	346,2	56,0
S20 Top	3,0	370,8	69,1
S20 Top	4,0	370,8	25,3
S20 Top	5,0	370,8	13,0
S20 Under	3,0	370,8	70,4
S20 Under	4,0	370,8	26,0
S20 Under	5,0	370,8	13,4
R20 Top	3,0	370,8	2564,3
R20 Top	4,0	370,8	678,2
R20 Top	5,0	370,8	212,2
R20 Under	3,0	370,8	2865,3
R20 Under	4,0	370,8	790,3
R20 Under	5,0	370,8	257,1
S21 Top	3,0	345,9	37,2
S21 Top	4,0	345,9	13,2
S21 Top	5,0	345,9	6,8
S21 Under	3,0	345,9	38,5
S21 Under	4,0	345,9	13,8
S21 Under	5,0	345,9	7,1
R21 Top	3,0	345,9	30,4
R21 Top	4,0	345,9	5,5
R21 Top	5,0	345,9	1,2
R21 Under	3,0	345,9	30,9
R21 Under	4,0	345,9	5,6
R21 Under	5,0	345,9	1,2

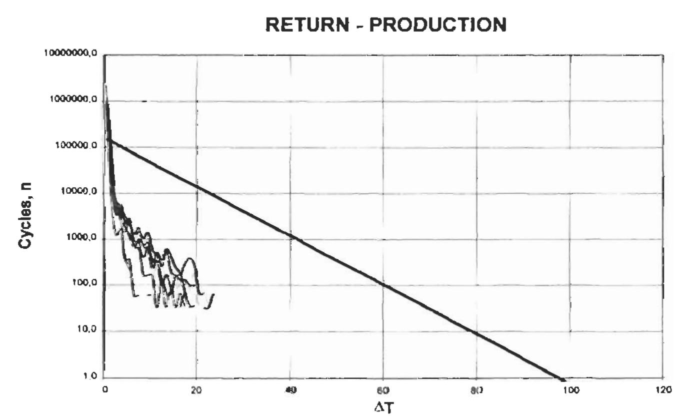
Enclosure 2



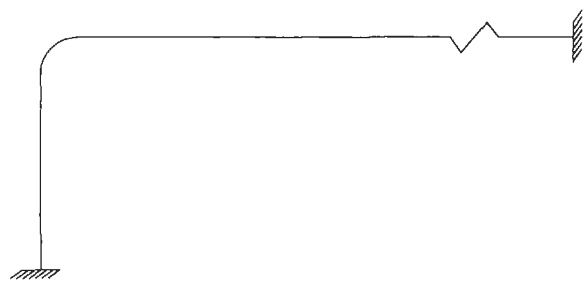
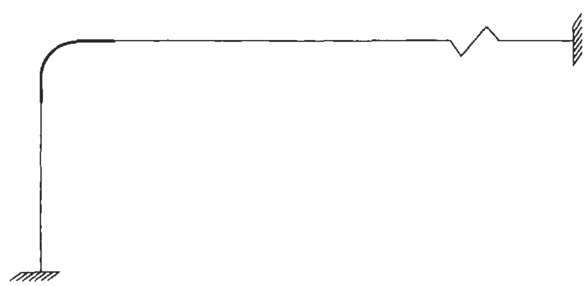
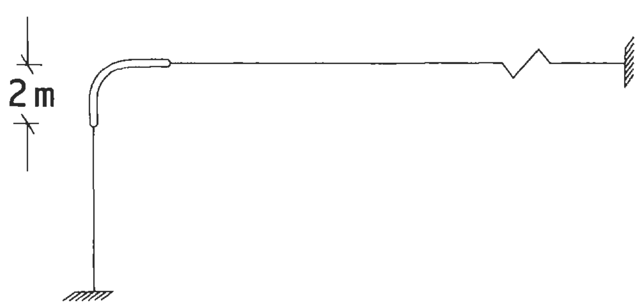


Enclosure 3

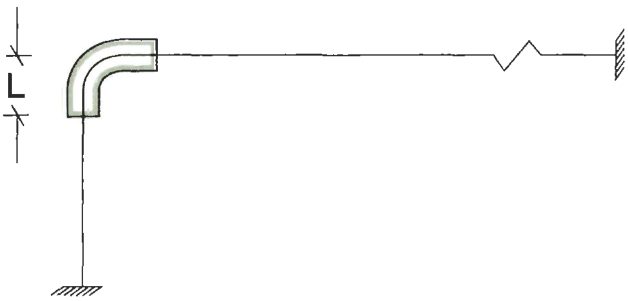




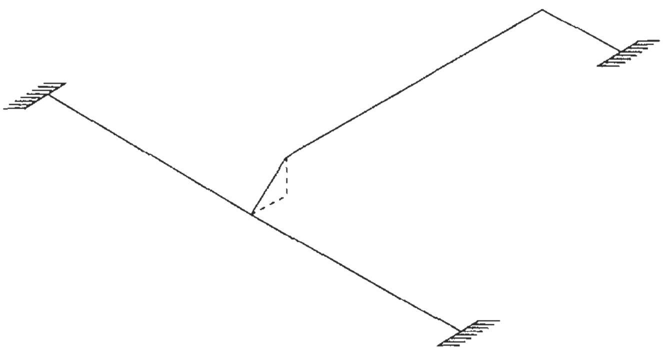
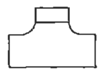
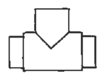
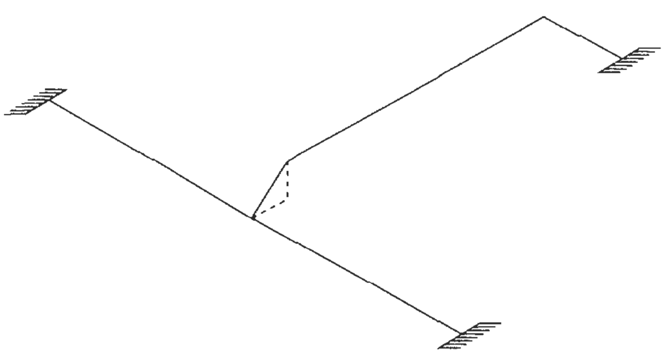
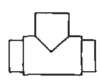
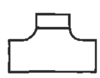
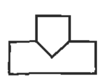
Enclosure 4

	Number of loadcycles to induce fatigue failure, without safety factor, $\Delta T_{ref} = 110$ °C		
Normal wall thickness in bend	DN80	DN150	DN600
	**140**	**196**	**1961**
	-	-	-
	-	-	-
Increased wall thickness in bend			
	**925**	**710**	**3440**
	-	-	-
	-	-	-
With foam cushions (Normal wall thickness in bend)			
	**393**	**704**	**2755**
	-	-	-
	-	-	-

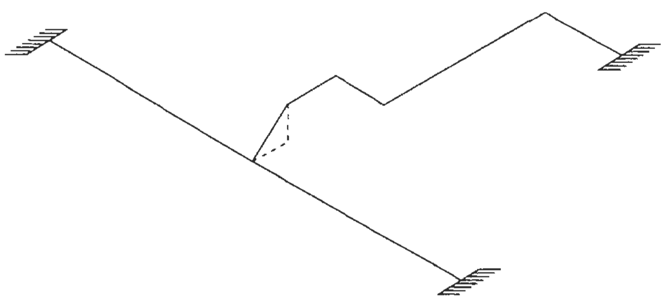
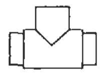
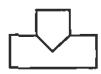
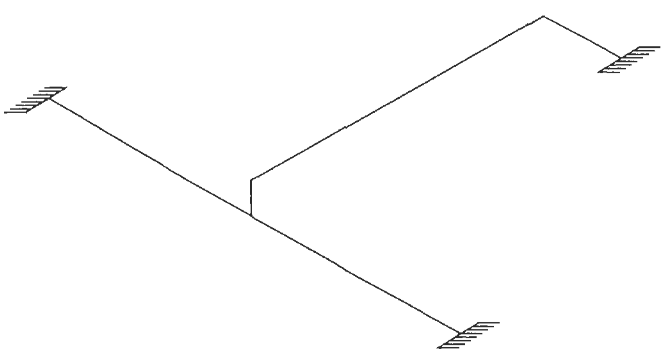
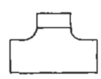
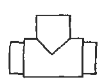
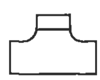
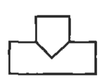
Enclosure 5

With concrete duct (Normal wall thickness in bend)	Number of loadcycles to induce fatigue failure, without safety factor, ΔT_{ref} = 110 °C		
	DN80	DN150	DN600
	**3956**	**3587**	**4534**
	L = 2 m.	**L = 3 m.**	**L = 6 m.**
	-	-	-

Enclosure 6

Axial stress in main pipe: 150 N/mm²		Number of loadcycles to induce fatigue failure, without safety factor, $\Delta T_{ref} = 110\ °C$			
		DN80/ DN20	DN200/ DN80	DN100/ DN100	DN450/ DN450
		**165**	**58**	**88**	**23**
		**306**	**141**	**673**	**110**
		**14356**	**7469**	**5183**	**6782**
		**2832**	**1719**	**2394**	**3043**
		**646**	**392**	**529**	**674**
Axial stress in main pipe: 200 N/mm²					
		**122**	**43**	**55**	**15**
		-	-	-	-
		**9344**	**4992**	**2983**	**3939**
		-	-	-	-
		-	-	-	-

Enclosure 7

		Number of loadcycles to induce fatigue failure, without safety factor, $\Delta T_{ref} = 110$ °C			
Axial stress in main pipe: 150 N/mm²		DN80/ DN20	DN200/ DN80	DN100/ DN100	DN450/ DN450
		**3062**	**6510**	-	-
		**6988**	**24898**	-	-
		**154773**	**348926**	-	-
		-	-	-	-
		-	-	-	-
Axial stress in main pipe: 150 N/mm²					
		**58**	**10**	-	-
		-	-	-	-
		**6023**	**1776**	-	-
		-	-	-	-
		-	-	-	-

District Heating and Cooling in Future Buildings

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Background

Efficient use of energy and energy saving in buildings are expected to be very focused areas in the future due to resource policy, environmental reasons, and the fact that the cost of energy is expected to increase relatively more than most other products. One of the consequences of this evolution is a constant development of better insulation of buildings.

This evolution has already - to some extent - changed the problem of controlling indoor temperatures in buildings from being mainly a heating problem to be an integrated heating and cooling problem. This evolution may have some effects on the competitiveness of District Heating and District Cooling in the future.

Objective of the IEA-project

The main objective of the project is to sort out the new conditions for the future expansion of District Heating (DH) and District Cooling (DC) that will be caused by the changes we can foresee in the future building stock. Expected development of the heating and the cooling loads and energy consumption patterns are in focus.

The project

The project is organised as a co-operation project between SINTEF Energy Research (N), VTT Building Technology (F), Fraunhofer UMSICHT (G) and Korea District Heating Corporation (K), with the project leadership at SINTEF, and with project support from the other partners. The project is monitored by an "Experts Group" consisting of appointed members from seven of the member countries of Annex V of the IEA-District Heating and Cooling Project.

A basic goal of the project is to map the heating and cooling loads and the energy consumption in a "typical" office building and a "typical" residential building from 1990, and make comparisons with the same expected "typical" buildings in 2005+. A simulation tool will be used for the study. The simulations for the actual countries are done for the same defined "standard" building configuration.

The defined "standard" office building and the defined "standard" residential building has been discussed and agreed on in the "Experts Group".

The office building has four storeys, with the floor area of 18m • 30m of each storey, and the residential building is a row house building with four flats in each row house with two storeys, with the floor area of 7m • 8m of each storey for one flat. The simulations are performed for a typical climatic situation in the respective four countries; Norway, Finland, Germany and Korea.

The input data for the building structure are based on the national building codes in the respective countries. The input data for the local climate are based on a standardised "reference year". The construction of the "reference year" for the different countries is based on slightly different methods, but this fact has been considered to have no significant effect on the practical result from the simulations.

In this paper we are presenting results from the following places in the four countries; Oslo, Helsinki, Oberhausen and Seoul.

It has to be mentioned that the climate may vary quite a lot within the different countries. Nevertheless, the results should give a good indication on the average situations in the respective countries.

The simulations are performed with the dynamic simulation programme FRES, (Flexible Room climate and Energy Simulator) which is developed at SINTEF/NTNU.
Based on the results from the simulations, the project has also some subsidiary goals:

- Determination of primary energy savings by DH compared to local heating systems
- Determination of the reduction of electric energy consumption and peek power by DC compared to conventional local cooling plants
- Determinations of environmental benefits from DH&DC compared to local heating and cooling plants

Presentation of some preliminary results

It is generally known that the density of the heating and the cooling consumption is one of the most critical factors for the feasibility of DH and DC systems. The density in relation to the maximum load demands is also a very critical factor. The reason why can be seen by the following equation, which basically describes the total cost per unit for the delivered energy:

$$C_s = \frac{I_s \bullet a}{\tau} + e_s \qquad (1)$$

C_s =	total specific energy cost	(money/kWh)
I_s =	specific investment cost	(money/kW)
e_s =	specific cost of energy	(money/kWh)
a =	annuity factor	(1/year)
τ =	equivalent time of max. load	(hours/year)

This equation shows that the findings from the simulated cases could be very fruitfully compared on basis of the values of the figure τ for the respective cases.

In the simulations shown here the room temperatures are set to 21°C when calculating the heat loads. The cooling loads will be very much depending on the maximum allowed room temperature during the high load periods. Here we have simulated the cooling load presupposing that it will be acceptable to let the room temperature slide from 21°C to 25°C before cooling of the room is introduced. If cooling is not introduced, a cooling load indicates that the room temperature will slide above 25°C.

The figures 1 - 8 show results from the simulations for the "standard" office building for the stage of 1990 and 2005+. The figures show to the left the called **duration curve of the *hourly mean values* of the heating load** as long as we have no cooling load. To the right in the figures the **duration curve of the cooling loads** is shown with the corresponding values of the heating loads when we have heating and cooling loads at the same time.

Figure 1.

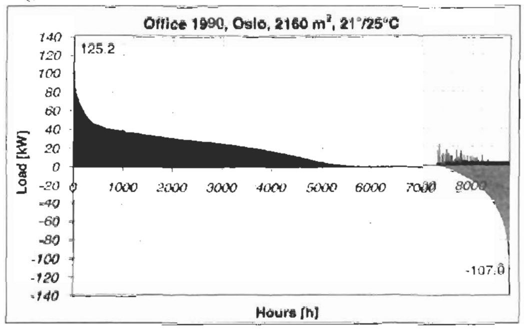
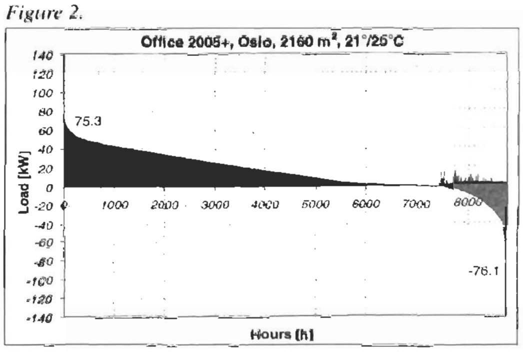


Figure 2.



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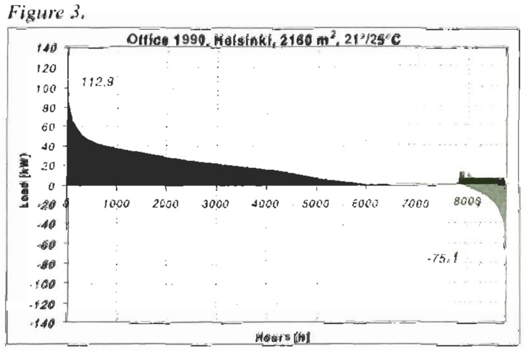

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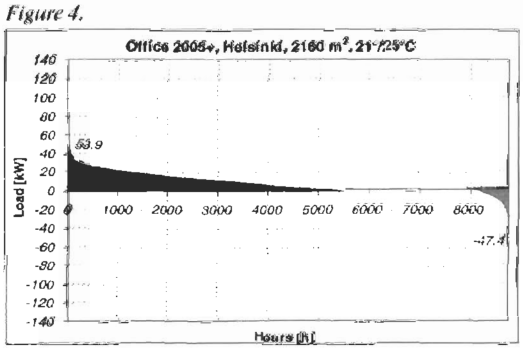

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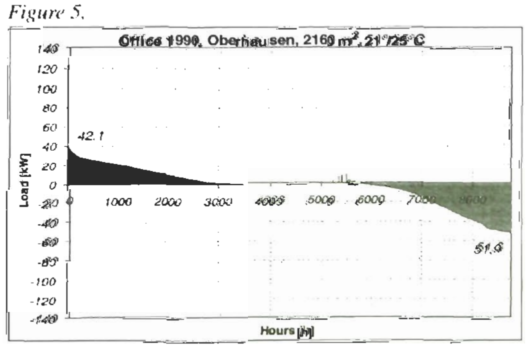

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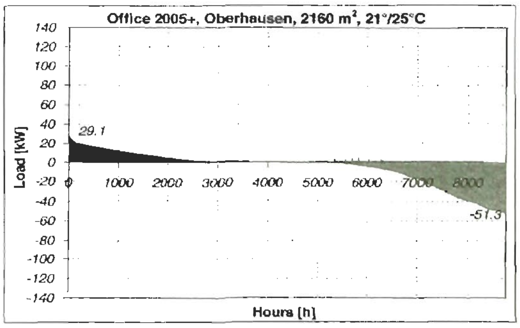

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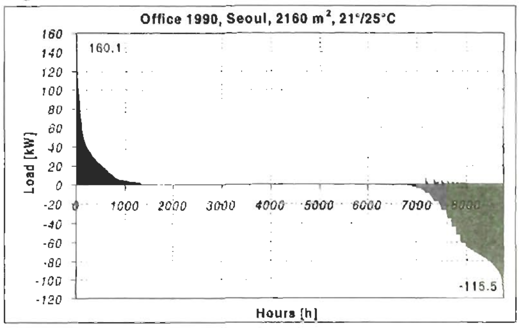


Figure 8.

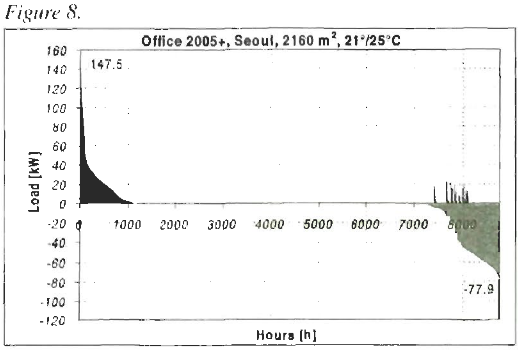


Figure 9 shows the **total yearly heating energy consumption** for the office building in the respective countries, and figure 10 shows the **total yearly cooling energy consumption**.

Figure 9.

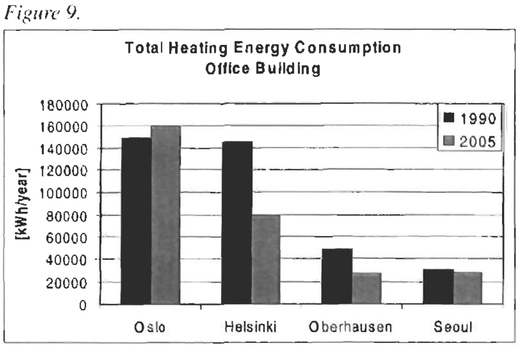
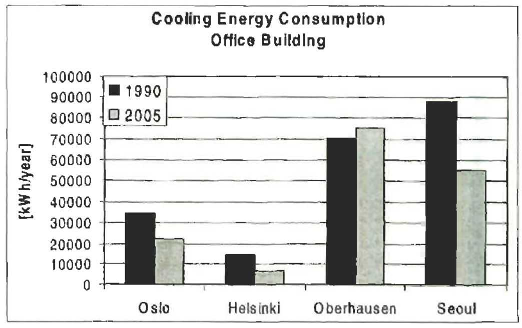


Figure 10.



Figures 11 and 12 show the **specific values for the total yearly heating and cooling energy consumption** for the office buildings in the respective countries.

Figure 11.

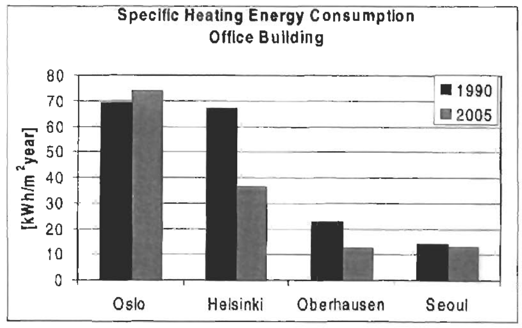
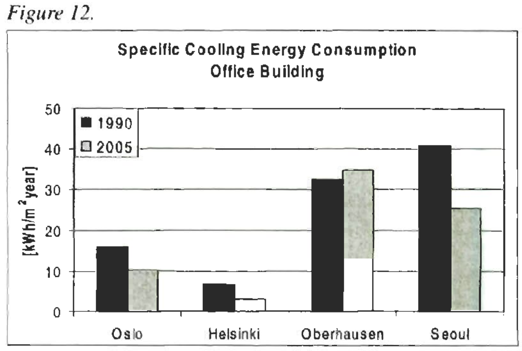


Figure 12.



The figures 13 and 14 show the **specific maximum heating and cooling loads** for the office buildings.

Figure 13.

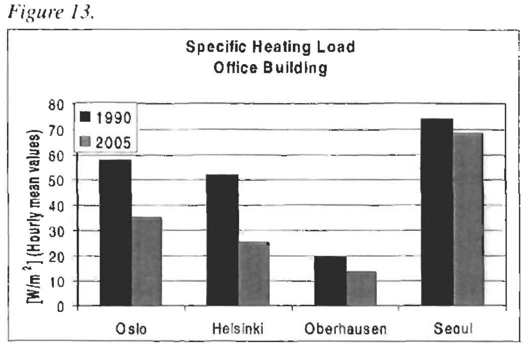
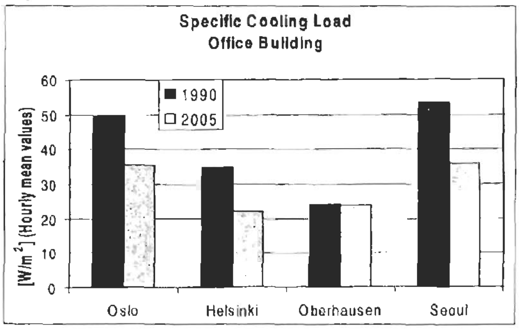


Figure 14.



The figures 15 - 22 show the results from the simulations for the "standard" residential building for the stage of 1990 and 2005+. The figures show so called **duration curves of the *hourly mean values* of the heating load** as long as we have no cooling load to the left.

To the right in the figures the **duration curve of the cooling loads** is shown with the **corresponding values of the heating loads** when we have heating and cooling loads at the same time. The values for the residential buildings are average values per flat for the four flats in the simulated row house.

Figure 15.

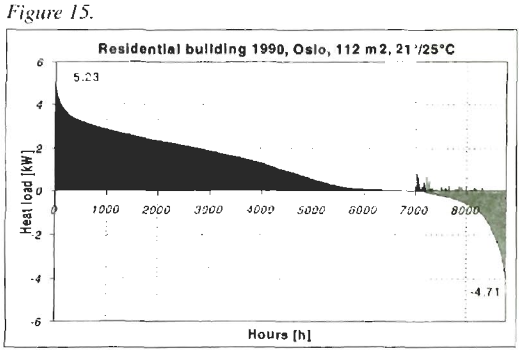


Figure 16.

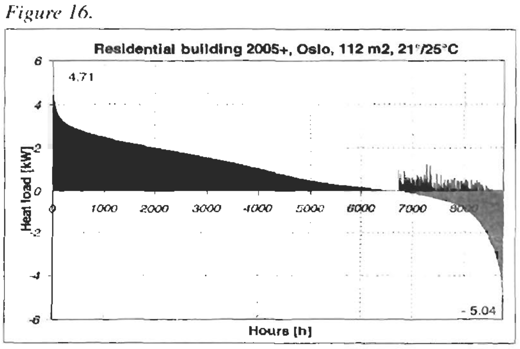
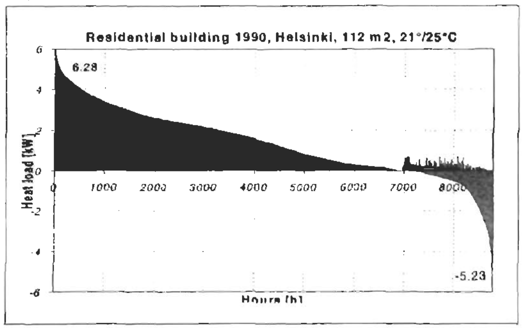


Figure 17.



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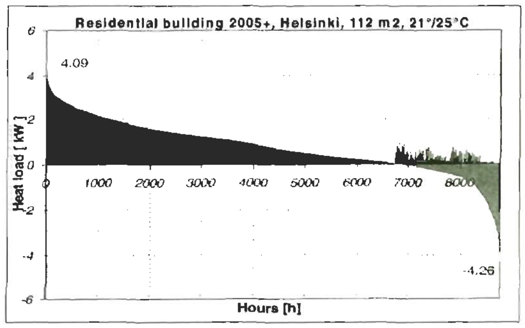


Figure 19.

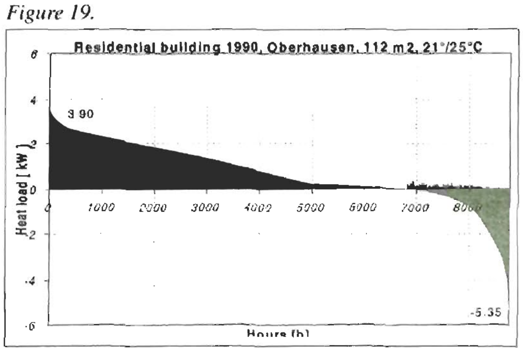


Figure 20.

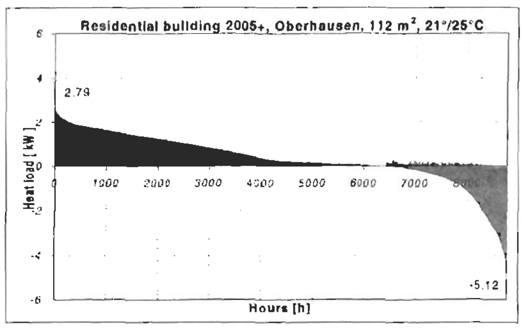


Figure 21.

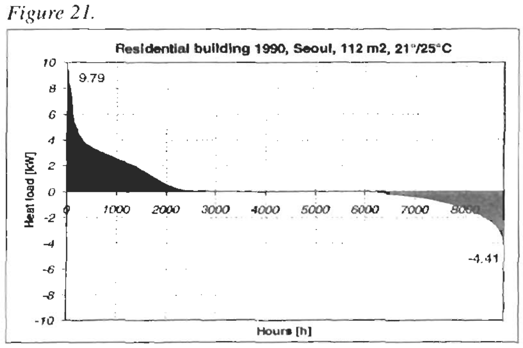


Figure 22.

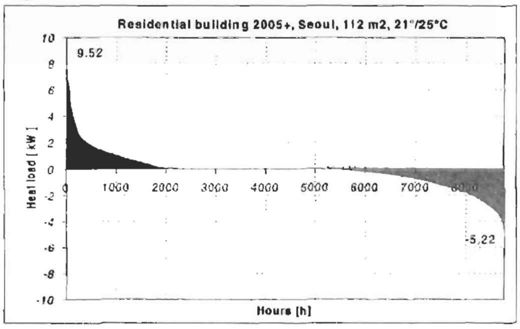


Figure 23 shows the **total yearly heating energy consumption** for the residential building in the respective countries, and the figure 24 shows the **total yearly cooling energy consumption**.

Figure 23.

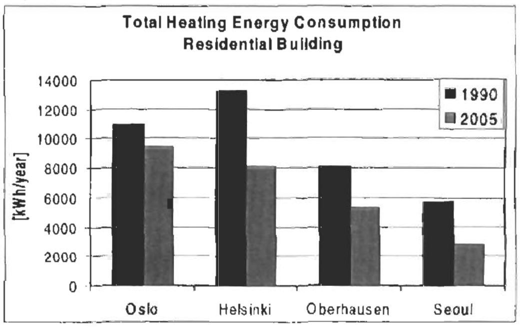
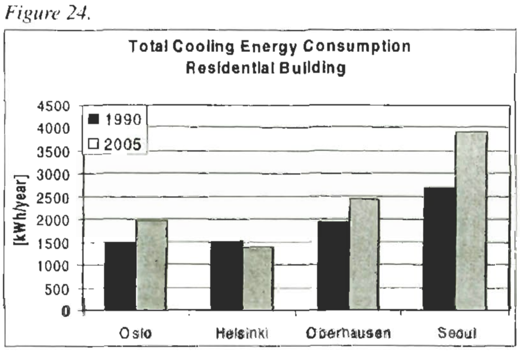


Figure 24.



The figures 25 and 26 show the **specific values for the total yearly energy consumption for heating and cooling** for the residential buildings in the respective countries, and the figures 27 and 28 show the **specific maximum heating and cooling loads**.

Figure 25.

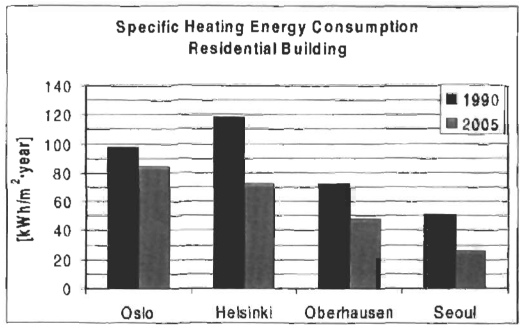


Figure 26.

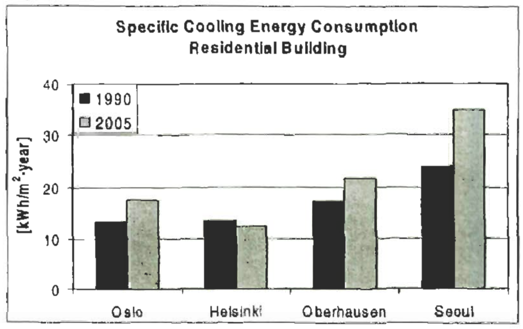


Figure 27.

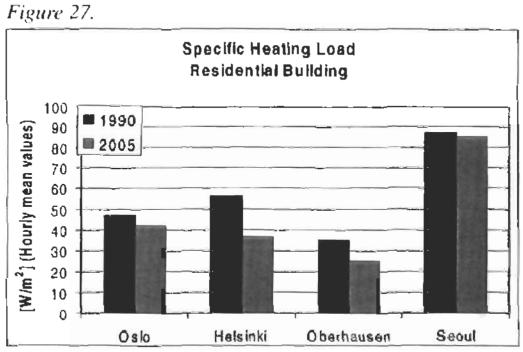
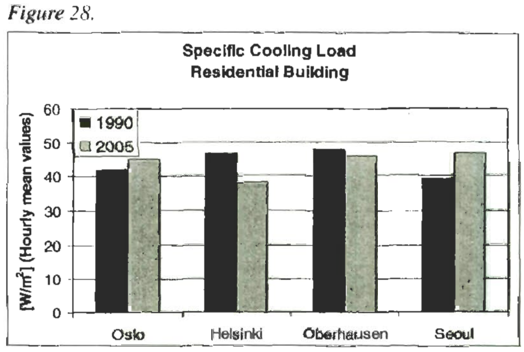


Figure 28.



Equivalent time of maximum load

Table 1 shows the calculated values of the equivalent time of maximum load (τ) for the total heating for the office building.

Office 2160 m²	Oslo	Helsinki	Oberhausen	Seoul
90-21°/25°	1190	1280	960	190
05-21°/25°	2120	1460	1030	190

Table 1. Equivalent time of maximum load for the total heating (hours/year)

Table 2 shows the calculated values of the equivalent time of maximum load (τ) for the total cooling for the office building.

Office 2160 m²	Oslo	Helsinki	Oberhausen	Seoul
90-21°/25°	320	190	1330	760
05-21°/25°	290	140	1490	710

Table 2. Equivalent time of maximum load for the total cooling (hours/year)

Table 3 shows the calculated values of the equivalent time of maximum load (τ) for the total heating for the residential building.

Resident 112 m²	Oslo	Helsinki	Oberhausen	Seoul
90-21°/25°	2090	2100	2070	590
05-21°/25°	2000	1980	1890	300

Table 3. Equivalent time of maximum load for the total heating (hours/year)

Table 4 shows the calculated values of the equivalent time of maximum load (t) for the total cooling for the residential building.

Resident. 112 m²	Oslo	Helsinki	Oberhausen	Seoul
90-21°/25°	320	290	360	610
05-21°/25°	390	320	470	750

Table 4. Equivalent time of maximum load for the total cooling (hours/year)

Some conclusions

- When considering the results, we have to keep in mind that the project at present is not finalised. It means that the work done so far has not passed all the quality checks that has to be done before the presentation of the final report on the findings.

- We have to be aware of the fact that the simulations give only theoretical values. Due to non-optimal construction and operation of the Heating, Ventilating and Air Conditioning systems (HVAC-systems), we will normally observe other values in practice. Normally the heating and cooling consumption might be considerably higher than the theoretical values. Nevertheless, the theoretical results should give a good picture on the relative values.

- The simulation results will, to some extend, be influenced by choices made by opinion by a small group of individuals based on their best judgement. These choices may of cause in some cases be debated. All the details behind the simulation results will be presented in the final report.

- The results show very clearly the typical effects of the different climatic conditions in the different countries. We also see the effects of different strategies of running the heating systems. The high values of the heating loads in the Seoul cases are caused by the fact that night setback of the room temperatures are applied. In the simulations it is assumed that you have heating capacity available to reheat the rooms to the desired value of 21°C during one hour. Reducing the capacity and choosing a longer period to reheat the room might be a better solution.

- The consequence by changes in the building codes is very clearly demonstrated in the office case from Oslo. The new building code from 1997 demands that the ventilation plant in an office building has to be run throughout day and night due to the contaminants from the building materials. It is hard to believe that the building codes for example in Finland and Norway will stay as differently in the future as we see in the office case. The results from all countries very clearly show the effect of the improved insulation of the walls and the windows from 1990 to 2005+.

- The equivalent time of maximum load seems to stay at the same level in the future except for the office building in Oslo due to new demands in the building code from 1997

- According to the plan, this IEA-project will be finished by April 1, 1999.

Plastic Pipe Systems for District Heating - Handbook for Safe and Economic Applications
Executive Summary

Contractor: ZW Energiteknik, Sweden
Heimio Zinko, project leader

Subcontractor: Manfred Klöpsch Ingenieur büro für Fernwärmetechnik, Germany
Manfred Klöpsch

Natural Resources Canada, Canada
Rob Brandon

1. Introduction

Plastic medium pipes have in some countries, especially in Scandinavia, been used for many years in floor heating applications and in smaller local networks. In Denmark these pipes are also quite common for smaller district heating pipe lines. In spite of years of experience, there still exists doubts about the possibilities of using plastic pipes in district heating applications, mostly because of the limitations in pressure and temperatures which must be observed when using plastic medium pipes. The use of plastic medium pipes is also limited to relatively small dimensions, i.e. below 100 mm diameter which makes it necessary to mix steel pipes and plastic medium pipes in many applications, a combination for which experience so far was not systematically documented.

The aim of the project is to compile know-how and installation experience from various countries and to present the results about plastic medium pipe techniques in the shape of a handbook. The handbook will describe the basic properties of plastic materials involved and the conditions for it's applications, as well as recommended laying and installation techniques to be used for receiving both technically and economically favourable results. Important examples from different countries will be included in the book.

The handbook will be devided in two main parts: An ***engineering part*** describing the main aspects of using and applying plastic medium pipes, including also economical system aspects, and a ***material part***, giving more detailed background information about the specific material properties as an Appendix. This summary presents some examples of information which can be found in the handbook.

2. Plastic pipe systems

The plastic medium pipe systems are described for all makes commercially available on the European market (i. e. in more than one country) for the last years. This implies that products under development or just pilot products are not included. This limits the make description to the following systems:

Bonded pipe systems with PEX:
ABB-PEX flex (ABB Isolrohr). Brugg-Calpex. Isoplus-Isopex. Lögstör LR-Pex. Tarco PEX/FLEX.

Non bonded pipe systems with PEX:
Uponor Ecoflex.

Pipe systems with other material than PEX
Flexalen.

A complete list of properties of these systems mentioned above is included in the Table 1. It gets evident from this list, that the PEX medium pipe is the prevailing pipe material and that practically all PEX-pipes to be used for district heating have a diffusion barrier of EVOH (ethylenvinylalcohol) which reduces the risk of oxygen diffusion to a great extent. Polybuthylene (PB) - with or without a diffusion barrier of Aluminium - is only offered by one manufacturer. PEX pipes are available in dimensions up to ND 100 mm whereas the PB pipes are offered also up to ND 125 mm. For pipe joints a variety of joint systems are available, most of them are of type press or screw fittings. An exception is the PB pipe which can be welded.

The advantage of plastic medium pipe systems is their flexibility. This holds not only for the plastic pipe but for the total pipe system including insulation and jacket. Even for the largest diameter the minimum radius of curvature is given to 1.5 m. In all pipe systems, except the Uponor-Ecoflex and Tarco, the insulation is made by PU-foam, covered with an outer jacket of PE. The Ecoflex system uses PEX foam insulation with a PE jacket, and Tarco uses a PU-foam with a jacket of an elastomere ethylenic butylacrylate.

3. Installation practice

The most important difference between plastic pipe systems and preinsulated pipes is their simple and quick assembly. Whereas a typical time for the construction of a section of preinsulated pipes might be six weeks, plastic medium pipe systems can be installed within a few days.

In assembly, only simple tasks have to be carried out which can be completed quickly. For this reason it has been shown that all installation work can be managed by the same contractor. For small pipe systems, the contractor normally uses light and easy manoeuvrable equipment, with which he can work quite efficiently. The pipe ditch is kept quite narrow with a minimum of excavated material since no welding work has to made down in the ditch (except some large holes for joints or service Tees). The sand bed can be immediately filled up, the plastic pipe laid down in it's full length and the ditch can be refilled within hours except may-be the excavation of larger holes made for joints or branches. By that way, roads are usually only blocked for some hours, bridges are not required and traffic interruption and other impact to the public is kept at a minimum.

The supply and return pipes are laid down in the ditch from above, without dragging them in the ditches. Changes in the pipe direction can be easily undertaken and any hinder such as large stones, trees or crossing cables can be circumvented. The pipes can also change from side-by-side to one-on-top-of the-other and back again. There is no need for costly expansion zones around the pipes in the ditch.

The installation of the total pipe system is very quickly completed since only a few connections have to be made and these connecting points can be positioned where an open hole does not disturb. The time consuming installation of a leak detection system is no longer necessary.

Pipe connections are best carried out as press connections. They can be mounted using a special tool far more quickly than a welded connection on a steel pipe. Visual control of the joints is sufficient.

Whereas the pipe laying effort for the main pipe system is considerably lower for plastic pipe systems compared to preinsulated pipes, branches require about the same effort for both systems. In plastic medium pipe systems, branches are produced very often with prefabricated Tees and the joints have to be carefully insulated and tightened.

4. Plastic medium pipe systems in comparison with preinsulated steel pipes

Today preinsulated steel pipes provide a reliable system which fulfils the requirements of district heating operation. On the other hand, cost considerations show advantages for plastic systems, the smaller the dimensions, the more economic. However, the choice of plastic pipe systems means always also a compromise between advantage and disadvantage. Table 2 compares the general properties of both types of systems. In summarising, it can be said that plastic medium pipe systems are limited in applications, in particular concerning temperature and pressure, but also in dimensions. On the other hand, they offer the advantage of quicker and simpler assembly at the building site.

All work considered, the effort spent for laying plastic medium pipe systems is very much less than for preinsulated pipes. This results also in lower total pipe system costs in comparison with the preinsulated pipes. In studies made in Sweden, Germany and Denmark it was shown that the total system costs are well below those for preinsulated pipes up to ND 65, the difference being larger the smaller the dimensions are. That means that the main advantage of plastic medium pipe systems are in applications where the transported energy is below 500 kW.

5. Material aspects

The most dominant questions concerning the plastic medium pipes are the limitations in temperature and pressure at one hand and the oxygen diffusion at the other hand.

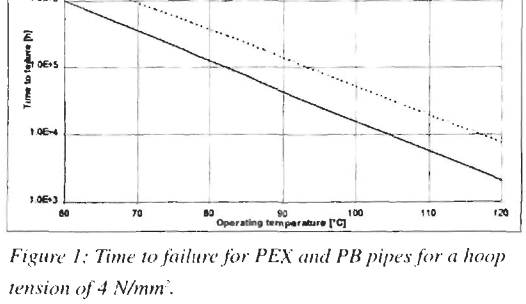


Figure 1: Time to failure for PEX and PB pipes for a hoop tension of 4 N/mm².

Life time dependent on temperature and pressure

This question has been investigated in different laboratories in Europe as well in the USA. However, plastic pipes have developed during the years and early measurements should be taken with caution. Reliable measurements for PEX and PB pipes have been published by Studsvik (Ref 1, 2) and are summarised in Figure 1. These measurements are based on both real time laboratory measurements for more than 10 years and accelerated measurements at elevated temperatures and show that the time to failure is depending on both pressure and temperature. By taking mean values of the time to failure over different measurement series a life time diagram as function of temperature and pressure can be constructed. Figure 1 shows an example for the expected minimum life time of PEX and PB pipes at a network pressure of 500 kPa (corresponding to a Hoop tension of 4 N/mm^2 including a safety factor of 1.8) (Ref 3).

The diagram indicates a life time of 114 years at T = 70 °C for PEX and for PB a life time of 34 years. However, when used in a normal district heating environment the operating temperature is varying during the year according to the ambient air temperature. That means that the highest temperature will prevail only short time. For instance in a typical district heating network operated at moderate temperatures it is expected that the temperature will be about 100 hours at 90 °C whereas the summer temperature might be 65 °C for more than 5000 hours. In such a net the expected life time can be calculated according to the "Miner rule" by adding life times for different temperature intervals. In such a temperature controlled operation a PEX system would last at least 150 years and a PB system 60 years.

Hence based on such calculation examples, different countries have made different recommendations for the use of plastic medium pipe systems. The best way to summarise those before a safe Standard will be introduced is to say that in Germany a network pressure of 500 kPa and a maximum temperature of 90 °C is recommended. In Sweden (and similar in Denmark) the recommendation is 600 kPa and 80 °C with maximum 95 °C for short periods. Inversely, for safe operating in 30 years at constant temperatures, the allowable temperature for PEX would be 83 °C and that of PB 74 °C.

Diffusion of oxygen

An important question is that of oxygen permeation through the plastic pipes. As it is known, untreated PEX and PB material exhibit such a high rate of oxygen permeation that such pipes only can be used in special applications where all metallic materials in contact with the water must fulfil fresh water quality standards.

Therefore plastic pipes are covered with permeation barriers in order to reduce the oxygen permeation. In PEX, all commercial makes for district heating use ethylenvenylalcohol, EVOH, as such a barrier. This barrier is usually applied as a thin film between an inner layer of PEX pipe and an outer protection layer.

Figure 2: Oxygene permeation in four different plastic pipe systems according to Ref /3/.

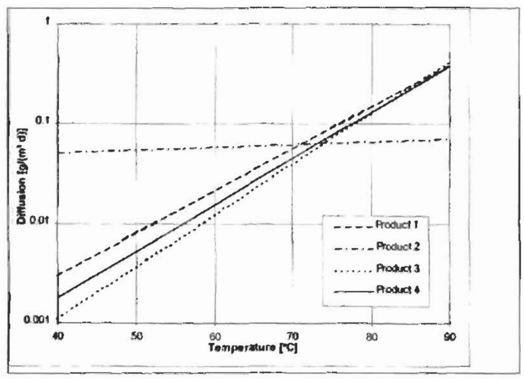
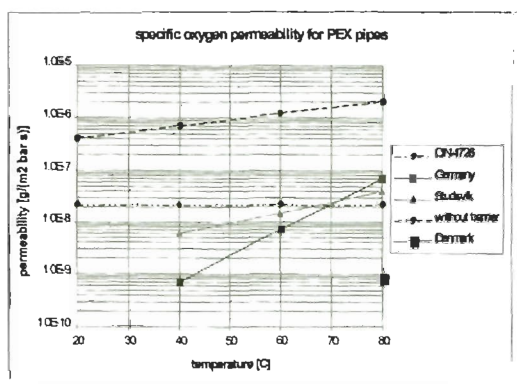


Figure 3: Oxygene permeation from measurements in different countries



Measurements of oxygen permeation through plastic pipes are very difficult to be carried out and can easily be biased by systematic errors. In recent times, such measurements have been performed on PEX pipes by laboratories in Sweden, Denmark and Germany (Ref 3, 4, 5). Usually such measurements are related to a given pipe dimension as shown in Figure 2 with the oxygen diffusion given in g/(m^3*day).

In order to be able to compare measurements from different countries, it is more convenient to convert them to specific values per m^2 medium pipe wall. Figure 3 compares the results from measurements performed at the three laboratories. The results from Studsvik, Sweden, are for sole medium pipes, the results from Denmark and Germany for the complete pipe system. Depending on the temperature the permeation difference between systems with and without barriers is in the orders of 100 - 1000.

The most important question is which impact the remaining oxygen permeation has on the possibility of connecting plastic pipes to steel systems. Field investigation has been performed at Fernwärmeverbund Saar in Germany (3). The expected corrosion impact from oxygen due to plastic pipes in two combined systems with twice as much plastic pipe volume as steel pipe volume was calculated to be less than 0.03 - 0.1 mm loss of steel pipe thickness during 35 years. A study in Denmark resulted in still lower values (4). There is no common recommendation yet for a minimum relation between amount of steel pipes and plastic pipes, but a study in Sweden (Ref 6) led to the conclusion that an amount of 10 % steel pipes in a combined plastic pipe/steel pipe net should be adequate for safe operation. However, the possible impact of local pitting and crevice corrosion has not bee studied.

For PB pipes, oxygen permeation studies are not known. The products of Flexalen are available without or with a permeation barrier of aluminium, which is expected appreciably to reduce the oxygen permeation.

Water vapour penetration

The influence of the diffusion of water vapour through the wall of PEX medium pipes on the pipe system is under investigation in Germany (3). In pipe systems consisting of media pipe, insulation and jacket, accumulation of moisture in the insulation of the pipe-system or also on the innerwall of the jacket can be expected, if the permeability of the jacket is less than that of the medium pipe. Moisture which accumulates in the insulation is expected to reduce the thermal resistance of the insulation. This on the other side will increase the temperature of the jacket and hence increase it's permeability until an equilibrium will be reached. Hence each pipe system is expected to have it's own balance of humidity depending on operation temperature and materials involved in jacket and insulation. On-going measurements at FFI (7) on three different plastic pipe systems have after one year not reached equilibrium concerning it's humidity content. It is expected that the industry will use results of those measurements for further optimising the water diffusion properties of their plastic medium pipe systems.

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Table 1: The Properties of Systems with Plastic Medium Pipes

Property		ABB-Isolpipe	Brugg-CALPEX	isoplus-isopex
1.	**Measurements**			
1.1	Nominal diameter DN			
	Single pipe from/to	16 - 25	20 - 65 (80)	16 - 80 (100)
	Double pipe from/to	not in programme	20 - 40	16 - 50
1.2	Length supplied: max	200 m DN 16 100 m DN 20,25	384 m DN 20 to 88 m DN 65 (80) Double pipe: 224 m - 88 m DN 20 DN 40	360 m DN 16 50 m DN 80 Double pipe: 360 m DN 16 50 m DN 50
1.3	Wall thickness jacket pipe	2 mm	2.0 - 2.8 mm	2.2 - 3.0 mm (EN 253)
2.	**Material**			
2.1	Medium pipe/Product	PEX	PEX DIN 16892/16893	PEX DIN 16892/16893
2.2	Thermal insulation	PUR foam	Pur foam Flexible 57 kg/m3	PUR hard foam 80 kg/m3(EN 253)
2.3	Jacket pipe	PE-LD	PE-LD	PE-HD
3.	**Safe loads (Set by manufacturer)**			
3.1	Max.temperature for variable operation	80 C constant	max. 95 C	max. 95 C
3.2	Pressure (for Tmax, variable)	10 bar DN 16/20) for 80C 6 bar DN 25) const	6 bar (10 bar)	6 bar
3.3	min. Radius of curvature	0.5 to 0.8 m	0.8 to 1.2 m Double p. 0.9 to 1.2 m	0.8 to 1.4 m
3.4	Minimum cover	ca. 40 cm to 50 cm	60 cm (40 cm without traffic)	40 cm
3.5	Required thickness of sand bed layer	5 cm	10 cm	5 cm
4.	**Construction, Design**			
4.1	Compound	yes	yes	yes
4.2	Measures for flexibility	Smooth pipe	Corrugated pipe soft PUR foam	Smooth pipe
4.3	Diffusion barrier available	EVOH	EVOH	EVOH
4.4	Single/Double pipe	Single pipe only	Single pipe; small DN double pipe as well	Single pipe; small DN double pipe as well

Lögstör LR-PEX	Tarco PEX/FLEX	Uponor Ecoflex-Thermo	Fernwärme Systeme Flexalen
16 - 80 (100) not in programme	16 - 40 not in programme	20 - 80 (100) 20 - 40	20 125 20 - 40
200 m DN 16 to 100 m DN 65 (90) then Rods à 12 m (Rods from DN 50)	200 m DN 20 to 50 m DN 40	200 m DN 20 to 50 m DN 80 (100) Double pipe: 200 m - 100 m DN 20 DN 40	100 m DN 65 from DN 80 (10) 12 m
no details	no details	1.1 - 2.2 mm	no details
PEX/Wirsbo DIN 16892/16893	PEX DIN 16892	PEX/Wisrbo	PB/Pipe Life
PUR foam medium hard > 50 kg/m3	PUR foam flexible > 60 kg/m3	PEX foam	PUR-element
PE-LD	EBA-Copolymer	PE-HD	PE
max. 95 C	max. 95 C	max. 95 C	max. 95 C
10 bar DN 16/20 6 bar from DN 25	10 bar DN 16/20 6 bar from DN 25	6 bar	6 or 10 bar
1 to 1.5 m	0.4 m DN 16 0.8 m DN 40	Single pipe 0.25 - 1.2 m Double p. 0.5 - 1 m	1.25 m
40 cm	40 cm	40 cm to 90 cm	50 cm to 80 cm
10 cm	no details - stone-free	10 cm above/below 15 cm sides	10 cm
yes	yes	no	no
Smooth pipes PUR foam medium hard	Smooth pipe elastic (rubber)	Corrugated pipe soft PUR foam	Corrugated pipe, insulating elements
EVOH	EVOH	EVOH	1. without: VS Type 500 2. with Al-Foil: System 1000
Single pipe only	Single pipe only	Single pipe; small DN double pipe as well	Single pipe; small DN double pipe as well with outgoing and return pipes in direct contact

Property		ABB-Isolpipe	Brugg-CALPEX	isoplus-isopex
5.	**Pipe joints**			
5.1	Connecting technique for medium pipe			
5.1.1	Metallic/Welding	Press fittings with O-Ring-seal; DN 16 and 20 screw connections as well	Metal coupling Press fittings (REHAU) and screw fittings (Beulco)	Press fittings; screw connectors in special cases (Beulco)
5.1.2	Transition plastic/metal or connection to steel pipe	metal connecting elements with weldable ends	metal connecting elements with weldable ends and screw connection	metal connecting elements with weldable ends
5.1.3	Branch of plastic pipe	Press fittings	metal tee connector	Press fittings
5.2	Connecting technique for insulation and jacket pipe			
5.2.1	Jacket pipe connection	Metal half shells	GFK-half shell	GFK-half shells
5.2.2	Thermal insulation	PUR-foam on location	Shaped parts of PE-foam and PUR-foam on location	PUR-foam on location
6.	**Particular properties of the system**			
6.1	Longitudinal water-tightness	yes	yes	yes
6.2	Water absorption into thermal insulation	low	low	low
6.3	Special features of the system	Compound construction	Compound construction, jacket pipe slightly corrugated	Compound construction
6.4	Notes	2 thicknesses of insulation available	Laying possible in horizontal borings usually 2 thicknesses of insulation available	Diffusion barrier coloured red

Lögstör LR-PEX	Tarco PEX/FLEX	Uponor Ecoflex-Thermo	Fernwärme Systeme Flexalen
16 - 80 (100)	16 - 40	20 - 80 (100)	20 125
not in programme	not in programme	20 - 40	20 - 40
200 m DN 16 to 100 m DN 65 (90) then Rods à 12 m (Rods from DN 50)	200 m DN 20 to 50 m DN 40	200 m DN 20 to 50 m DN 80 (100) Double pipe: 200 m - 100 m DN 20 DN 40	100 m DN 65 from DN 80 (10) 12 m
no details	no details	1.1 - 2.2 mm	no details
PEX/Wirsbo DIN 16892/16893	PEX DIN 16892	PEX/Wisrbo	PB/Pipe Life
PUR foam medium hard > 50 kg/m3	PUR foam flexible > 60 kg/m3	PEX foam	PUR-element
PE-LD	EBA-Copolymer	PE-HD	PE
max. 95 C	max. 95 C	max. 95 C	max. 95 C
10 bar DN 16/20 6 bar from DN 25	10 bar DN 16/20 6 bar from DN 25	6 bar	6 or 10 bar
1 to 1.5 m	0.4 m DN 16 0.8 m DN 40	Single pipe 0.25 - 1.2 m Double p. 0.5 - 1 m	1.25 m
40 cm	40 cm	40 cm to 90 cm	50 cm to 80 cm
10 cm	*no details - stone-free*	*10 cm above/below* 15 cm sides	*10 cm*

Table 2: Advantages and disadvantages of plastic pipe systems compared with preinsulated pipes

Criterion	Plastic Pipe Systems	Preinsluated Pipes
Safe Loads		
Pressure, Temperature	**Limit**	
	- necessary	+ no problem
Construction, Design		
Available nominal diameters		
compensation	**- small pipes**	+ all
thermal insulation	**+ not necessary**	- requires much effort
jacket pipe		
diffusion tightness	**identical**	identical
(oxygen, water vapour)	**- tight to some extent**	+ tight
Assembly on site		
Building time	**+ short**	- long
Number of Joints	**+ low**	- high
Flexibility	**+ high**	- low
(Adaption to route)		
Weight	**+ low**	- high
Operation safety		
Corrosion in medium pipe	**+ impossible**	- possible
Leak detection	**+ not necessary**	- normal
Building costs	**+ lower as preinsulated pipes**	- higher

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