TC 6.02 District Energy – Sub-Committee Meeting
Sunday, 1:00 pm – 3:00 pm, GWCC, Room B207

Voting Committee Members Present:

Mr. Alan Neely	6/30/2019	Mr. Michael Calabrese	6/30/2020
Mr. Geoffrey Bares	6/30/2020	Mr. Daniel Richard Pyewell	6/30/2020
Ms. Jessica Mangler	6/30/2020	Mr. Tim Anderson	6/30/2022
Mr. Steve Tredinnick	6/30/2021		

New Incoming Voting Committee Members:
None

1. The Chair Called to Order

Technical Committee 6.02 is concerned with district energy technology and integrated systems that provide one or more forms of thermal energy or a combination of thermal energy and electric power from a central plant(s) to meet the heating, cooling, or combined thermal energy and power needs of end-users in two or more structures. The TC collects and disseminates information on regional resource planning including the design, performance, economic analysis, operation and maintenance of central plants, distribution networks, and consumer limited to materials, construction methods, heat transfer, fluid flow, and measurement. The scope of the TC includes the development and assessment of associated technologies for energy use sections.

2. Handouts Distributed

A. 2019 Winter Meeting (Atlanta, GA) Agenda
B. 2020 Systems and Equipment Handbook, Chapter 12 & Chapter 15, with revisions noted.

3. Sub-Committee Discussion:

A. Handbook: Steve Tredinnick (Chair)
- i. Discussion about the revised Chapter 12 - "District Heating and Cooling" & Chapter 15 – "Medium-And High-Temperature Water Heating"
  - (1) Updated the Chapters language
  - (2) Updated the Chapters LCC calculation
  - (3) Discussion of Chapter 15 – "Medium-And High-Temperature Water Heating", not to be combined with other chapter(s)
  - (4) Text is due to society on 4/25/19
  - (5) Discussion about the updates within the document. Updated pipe technology, future worth, etc.
  - (6) Changes made every 4 years
  - (7) Need a volunteer to take over for next book revision. Betsy Goll

B. Program: Tim Anderson (Chair)
- i. Advertise to membership for interesting projects, etc.
- ii. Mohamed Ibrahim reviewed overseas projects. Thermal Storage with conparisons to battery storage
- iii. A potential program was discussed, Balancing a grid with district energy
  - (1) Seminar presentation for Orlando 2020.
  - (2) Workshop or forum need a TC co-sponsor, such as TC 6.9, Anderson will review with TC 6.9. Need an abstract by Kansas City 2019 mtg.

C. Research: Jay Eldridge (Chair)
   i. None

4. Meeting Adjourned at 2:30 pm.

Meeting Minutes: TC 6.02 District Energy
Sunday, 1/13/2019, Atlanta, GA

TC 6.02 District Energy
Sunday, 3:00 pm – 5:00 pm, GWCC, Room B207

Voting Committee Members:

Mr. Alan Neely	6/30/2020	Mr. Michael Calabrese	6/30/2020
Mr. Geoffrey Bares	6/30/2020	Mr. Daniel Richard Pyewell	6/30/2020
Ms. Jessica Mangler	6/30/2020	Mr. Tim Anderson	6/30/2022
Mr. Steve Tredinnick	6/30/2021		
Mr. Scott Murray	7/1/2021		

New Incoming Voting Committee Members:
None

1. The Chair Called to Order and discussed -

Technical Committee 6.02 is concerned with district energy technology and integrated systems that provide one or more forms of thermal energy or a combination of thermal energy and electric power from a central plant(s) to meet the heating, cooling, or combined thermal energy and power needs of end-users in two or more structures. The TC collects and disseminates information on regional resource planning including the design, performance, economic analysis, operation and maintenance of central plants, distribution networks, and consumer limited to materials, construction methods, heat transfer, fluid flow, and measurement. The scope of the TC includes the development and assessment of associated technologies for energy use sections.

2. General introductions
A. Quorum exists.

3. ASHRAE Code of Ethics Commitment – Announced by TC Chair – Alan Neely

In this an all other ASHRAE meetings, we will act with honesty, fairness, courtesy, competence, integrity and respect for others, and we shall avoid all real or perceived conflicts of interest. (see full code of ethics: https://www.ashrae.org/about-ashrae/ashrae-code-of-ethics.)

4. Membership (TC Chair Alan Neely)
A. Members & Guests Introductions, roundtable introductions
B. Review of Current Voting Members and TC officers
i. Ref. Manual of Procedures – 3.33, 2.13, 2.14 & 2.15
C. Recruiting of new members and officers for this technical committee (TC 6.02)
i. Elizabeth Goll
ii. Mohamed Ibrahim
D. TC Chair Alan Neely will be completing his term in 2019

5. Voting for Approval of Minutes
A. January 2018 Winter Meeting in Chicago, IL, (by TC 6.02 secretary, Michael Calabrese)
i. Chicago Minutes were not voted on in Houston, No Quorum
ii. Minor revision was reviewed.
iii. Motion to approve by Tim Anderson, all other voting members approved, none declined.
B. 24th June 2019 meeting in Houston, TX (by TC 6.02 secretary, Michael Calabrese)

i. Minor revision was reviewed.
ii. Motion to approve, Tim Anderson, all other voting members approved, none declined.

6. Announcements by Chair:
A. CEC call for reviewers: 2019 Annual Meeting and others. Contact Tiffany Cox (tcox@ashrae.org) and copy TC Chair and Vice Chair.

7. Visitors and Liaison Reports/Information from other Committee Representatives
A. DOE – Richard Sweetser
i. $10M available for advanced manufacturing, see IDEA liaison, John Andrepont.
ii. CHP, HQ for DOE is moving towards a package systems in a "Catalog". NYSERDA currently has a "Catalog"
B. IDEA – John Andrepont (non-present, provided report to Chair)
i. See the report attached to the agenda handout and available on the TC webpage.

8. Sub-Committee Reports:
A. Handbook: Steve Tredinnick (Chair)
i. Chair reported to the group about the revisions to Chapter 12 - "District Heating and Cooling" & Chapter 15 – "Medium-And High-Temperature Water Heating"
(1) Updated the Chapters language
(2) Updated the Chapters LCC calculation
(3) Chapter 15, not to be combined with other chapter(s)
(4) Text is due to society on 4/25/19
(5) Discussion about the updates within the document.
ii. Motion from Jessica Mangler to approve, all other voting members approved, none declined
B. Membership: John Andrepont (Chair)
i. Advertise to membership for interesting projects, etc.
ii. Steve Tredinnick may not be attending. Alec Cusick, Chair (Alan Neely)
C. Program: Tim Anderson (Chair)
i. Orlando, Winter 2020 was reviewed.
(1) A potential program was discussed, balancing a grid with district energy or an innovative approach to electric
(2) Need a seminar presentation
(3) Review workshop or forum. An abstract will be required.
(a) This TC may need to get a co-sponsor, such as TC 6.9.
Tim Anderson will review with TC 6.9.
(4) Maybe operation and HVACR
(5) A need for a good title and a good track was reviewed. A track with unique cutting-edge approaches. Reference the Orlando / Kansas City tracks #3.
(a) Vikrant Aute is the contact
(6) Cross-reference IDEA for date conflicts.
ii. Thermal energy – pumping issues – tertiary pumping, etc.- hydraulic bridges due 2/8/19
(1) District connections, booster pumps
(2) District energy pump optimization, coupled or de-coupled, workshop under track 3
(a) Vote for Kansas City, Annual 2019. Motion to approve by Tim Anderson, all other voting members approved, none declined, the chair Alan Neely abstained.
iii. Today there was a TC 7.3 and 6.2 co-sponsored a presentation but representation from TC 6.2 was not present.
D. Standards: Open
i. NONE
E. Webmaster: Dan Pyewell (Chair)
i. Reviewed the status of the TC 6.2 website. Documentation will be uploaded for this meeting.
F. Research: Jay Eldridge (Chair)
i. District cooling guide was discussed.

ii. RP 1762 requested an extension until June 2019.
iii. Vote: Motion to approve, Tim Anderson, seconded by Dan Pyewell all other voting members approved, none declined, chair abstained.

9. Old Business-
A. None

10. New Business-
A. Discussion of TCs potentially combining and synergized. A survey form was provided at the breakfast meeting.
i. There are currently 100+ TCs
ii. The conference schedule could be adjusted to have less overlap in the TCs sessions and allow for more conference attendance of the members.
(1) This would require the chairs to be consolidated and smaller working groups formed.
iii. The revised TC format would ideally be in place for the Orlando, Winter 2020 conference.
iv. There was a discuss about which TCs would synergize with TC 6.2
(a) Potentially TCs 8.3, 1.10, 6.9, 6.1 may be appropriate.
(2) Open discussion about the pros and cons of this change
(3) The aggressive schedule change was discussed. It may be unrealistic due to the challenges faced with this consolidation.
v. Jessica Mangler read aloud the survey letter as completed during the earlier breakfast meeting.
B. Tim Anderson reviewed the arrangement of the CHP/DE program track at this conference.
i. It was arranged by teleconference and a short meeting.

11. Meeting Adjourned at 5:00 pm.
A. Announcement by chair Alan to close this session, motion by Jessica Mengler, 2nd by Michael Calabrese, all vote 5-0-1, the chair Alan Neely abstained.

Subcommittee Meetings - Day, Date, Times, & Location Prior to TC 6.2 Meeting:

TC 6.02 Handbook - Sunday, 13 January 2019, 1:00 - 3:00 pm, GWCC, 2nd Floor, Building B, B207

TC 6.02 Program - Sunday, 13 January 2019, 1:00 - 3:00 pm, GWCC, 2nd Floor, Building B, B207

TC 6.02 Research - Sunday, 13 January 2019, 1:00 - 3:00 pm, GWCC, 2nd Floor, Building B, B207

ASHRAE Winter Conference
TC 6.02 District Energy
Sunday, 2019-01-13, Atlanta, GA

Attendance List

ASHRAE

1791 Tullie Circle, N.E./Atlanta, GA 30329
404-636-8400

DRAFT

TC/TG/MTG/TRG MINUTES COVER SHEET

(Minutes of all Meetings are to be distributed to all persons listed below within 60 days following the meeting.)

TC/TG/MTG/TRG No. 6.2 DATE

TC/TG/MTG/TRG TITLE DISTRICT ENERGY

DATE OF MEETING 2019-01-13 LOCATION ATLANTA, GA; GWCC RM B207

MEMBERS PRESENT	YEAR APPTD	MEMBERS ABSENT	YEAR APPTD	EX-OFFICIO MEMBERS AND ADDITIONAL ATTENDANCE
ALAN NEELY	2017			
MICHAEL CALABRESE	2018			
JESSICA MANGLOR	2017			
DAN PYEWELL	2018			
STEVE TREDINNICK	2017			
GEOFFREY BARES	2016			
TIM ANDERSON	2018			

DISTRIBUTION: *All Members of TC/TG/MTG/TRG plus the following:*	
TAC Section Head:	SHx@ashrae.net Where x is the section number
All Committee Liaisons As Shown On TC/TG/MTG/TRG Rosters (Research, Standards, ALI, etc.)	See ASHRAE email alias list for needed addresses.
Mike Vaughn, Manager Of Research & Technical Services	MORTS@ashrae.net

ASHRAE Shaping Tomorrow's Built Environment Today

TC Sign-in Sheet

Meeting Info: Tc 6.2 Date: 1.13.19

Name	Affiliation	E-mail	Member (Voting, Corresponding, or Guest?)	YEA Member? (Yes/No)
MICHAEL CALABRESE		mcalabrese@burns-group.com	V	NO
Alan Neely	OC	Alan.NEELY@[illegible].com	Voting	No
Timothy Anderson	Applied Engr	tanderson@applied-es.com	Programs	NO
Jessica Mangler	AEI	jmangler@aeieng.com	VM - Vice chair	Y
John Molnar	CoEng Advisors	john.molnar@coengadvisors.com	CM	Y
Mohamed Ibrahim	DcPro	AQRAA@hotmail.com	corresponding	No
Steve Tredinnick	Burns & McDonnell	stredinnick@burnsmcd.com	VM	N
Gary Phetteplace	GWA Research	garyp@gwaresearch.com	CM	N
DAVID WADE	RDA Engineering	dww@rdaeng.com	CM	N.
STEPHEN KAZIM	P2S INC	Stephen.Kazim@p2sinc.com	CM	No

TC4.2 19/01/13

Name	Affiliation	E-mail	Member (Voting, Corresponding, or Guest?)	YEA Membe (Yes/No
Charlotte Dean	P2S Inc	Charlotte.Dean@p2sinc.com	C M	Yes
Margaret Phillips	p2S Inc	margaret.phillips@p2sinc.com	CM	yes
Matt Bickett	ASHRAE	mebickett@gmail.com	CM	NO
Baki Cvijetinovic	SNC Lavalin	branislav.cvijetinovic@snclavalin.com	Guest	Yes
Daniel Pyewell	Tanco Engineering	dpyewell@tancoeng.com	VM	No
Jim Young	ITW	jyoung@itwinsulation.com	CM	N
Heather Sharif	ITW	hsharif@ itwinsulation.com	guest → need to become CM	Ye
JAY ELDRIDGE	DAIKIN	JAY.ELDRIDGE@DAIKINAPPLIED.COM	CM	[illegible]
DHARAM PUNWANI	AVALON CONSULTING	dpunwani@avalonconsulting.com	CM	N
Geoff Bares	Enwave Chicago	geoff.bares@enwave.com	VM	N

TC 6.2 — 19/01/13

Name	Affiliation	E-mail	Member (Voting, Corresponding, or Guest?)	YEA Member? (Yes/No)
Charlotte Dean	P2S Inc	Charlotte.Dean@p2sinc.com	C M	Yes
Margaret Phillips	P2S Inc	margaret.phillips@p2sinc.com	CM	Yes
Matt Bickett	ASHRAE	mebickett@gmail.com	CM	NO
Baki Cvijetinovic	SNC Lavalin	branislav.cvijetinovic@snclavalin.com	Guest	Yes
Daniel Pyewell	Tanco Engineering	dpyewell@tancoeng.com	VM	No
Jim Young	ITW	jyoung@itwinsulation.com	CM	No
Heather Sharif	ITW	hsharif@itwinsulation.com	guest → need to become CM	Yes
Jay ELDRIDGE	DAIKIN	JAY.ELDRIDGE@DAIKINAPPLIED.COM	CM	NO
DHARAM PUNWANI	AVALON CONSULTING	dpunwani@avalonconsulting.com	CM	NO
Geoff Bares	Enwave Chicago	geoff.bares@enwave.com	VM	No

TC 6.2 19/01/13

Name	Affiliation	E-mail	Member (Voting, Corresponding, or Guest?)	YEA Member? (Yes/No)
Rich Sweetser	Exergy Partners	rsweetser@ exergypartners.com	CM	No
Art Giesler	Permalert	art.giesler@art.net	CM	N
Elizabeth (Betsy) Goll	FPL Energy Services	elizabeth.goll@FPL.com	CM	N
DAVID DVORAK	UNIVERSITY OF MAINE	dvorak@maine.edu	CM	N
Kevin Chisholm	MAEC, LLC	kevchis@aol.com	guest	N
SAMEER KESHAVAN	SOUTH DAKOTA STATE UNI. STUDENT	sameer.keshavan@gmail.com	Guest (CM)	Y
Dawen Lu	Lu+Smith Engineers	dlu6838@gmail.com	Guest	N
Sayani Hari Babu	Gemini Cooling System Pvt Ltd	geminicoolingsystem@gmail.com sayanihari@yahoo.com	Corresponding Member	N

ASHRAE Winter Conference
TC 6.02 District Energy
Sunday, 2019-01-13, Atlanta, GA

<u>Revised Meeting Minutes</u>
ASHRAE Winter Conference
Sunday, 2018-01-21, Chicago, IL

Meeting Minutes: TC 6.2 District Energy
Sunday, 2018-01-21, Chicago, IL

TC 6.2 District Energy
Sunday, 1:00 pm – 3:00 pm, Palmer House

Voting Committee Members

Mr. Alan Neely	6/30/2020		
Mr. Geoffrey Bares	6/30/2020	Mr. John Andrepont	6/30/2017
Ms. Jessica Mangler	6/30/2020	Mr. Drew Overmiller	6/30/2017
Mr. Lawrence Markel	6/30/2019	Mr. Daniel Richard Pyewell	6/30/2017
Mr. Jay Eldridge	6/30/2018	Mr. Tim Anderson	6/30/2022
Mr. Scott Murray	7/1/2021	Mr. Steve Tredinnick	6/30/2021

New Incoming Voting Committee Members

Mr. Scott Murray 7/1/2021

1. The Chair Called to Order, and discussed the following.

Technical Committee 6.2 is concerned with district energy technology and integrated systems that provide one or more forms of thermal energy or a combination of thermal energy and electric power from a central plant(s) to meet the heating, cooling, or combined thermal energy and power needs of end-users in two or more structures. The TC collects and disseminates information on regional resource planning including the design, performance, economic analysis, operation and maintenance of central plants, distribution networks, and consumer limited to materials, construction methods, heat transfer, fluid flow, and measurement. The scope of the TC includes the development and assessment of associated technologies for energy use sections.

A. Agenda & Attendance Handouts
B. ASHRAE Code of Ethics Commitment – TC Chairman – Alan Neely
C. Members & Guests Introductions

2. Voting for Approval of Minutes from 21th January, 2018 meeting in Chicago, IL.
A. Not completed. Need quorum.

3. Announcements by Chair:
i. Section 6 Breakfast: Conferences & Expositions Committee (CEC)
(1) 2018 Chicago, IL
- Track 1: Fundamentals and Applications
  Track Chair: Frank Schambach (frankschambach@mindspring.com)
- Track 2: HVAC&R Systems and Equipment
  Track Chair: Jennifer E. Leach (pennst8jen@yahoo.com)
- Track 3: Refrigeration
  Track Chair: Vikrant Aute (vikrant@umd.edu)
- Track 4: Building Life Safety Systems
  Track Chair: Robert Alan Neely (alan_neely@pghcorning.com)
- Track 5: Controls – Smart Building Systems and the Security Concerns as Technology Emerges
  Track Chair: Melanie Derby (derbym@ksu.edu)
- Track 6: Commissioning – Optimizing New and Existing Buildings and their Operation
  Track Chair: Dennis Alejandro (denzjac@yahoo.com)
- Track 7: Zero Net Energy Buildings – The International Race to 2030
  Track Chair: Jason DeGraw (jason.degraw@nrel.gov)

- Track 8: Residential Buildings – Standards Guidelines and Codes
  Track Chair: Kimberly Pierson (kdpwildcat@gmail.com)
- Track 9: Research Summit
  Track Chair: Ann Peratt (ann.peratt@gmail.com)

(2) 2018 Summer Houston, TX
- (a) Thursday, 2018-01-11: Website Opens for Seminar, Workshop, Forum, Debate and Panel Proposals
- (b) Monday, 2018-01-15: Conference Paper Accept/Revise/Reject Notifications
- (c) Friday, 2018-02-09: Program (Seminar, Forum, Workshop, Debate and Panel) Proposals Due
- (d) Friday, 2018-02-09: Revised Conference Papers/Final Technical Papers Due
- (e) Monday, 2018-02-19: Conference and Technical Paper Final Accept/Reject Notifications
- (f) Tuesday, 2018-03-01: Registration Opens
- (g) Monday, 2018-03-19: Seminar, Forum, Workshop Accept/Reject Notifications
- (h) Monday, 2018-04-30: Upload of PPTs Begin
- (i) Friday, 2018-06-01: All PPTs Due Online
- (j) Wednesday, 2018-06-20: Final Day for Commercialism Revision Upload prior to on-site
- (k) Saturday, 2018-06-23: Speaker's Lounge Opens
- (l) **Conference begins (2018-06-27)**

(3) Program Statistics for Long Beach: total of 107 available slots
- (a) Conference Papers
  - (i) 151 abstracts submitted, 129 approved
  - (ii) 71 papers received
  - (iii) 20 conference papers presented
- (b) Technical Papers
  - (i) 27 received
  - (ii) 19 technical papers presented
  - (iii) 6 technical paper presentations
- (c) Seminars
  - (i) 121 submitted
  - (ii) 61 presented
- (d) Workshops
  - (i) 13 submitted
  - (ii) 9 presented
- (e) Forums
  - (i) 11 submitted
  - (ii) 4 presented
- (f) Debates (New)
  - (i) 4 submitted
  - (ii) 3 presented
- (g) Panel (New)
  - (i) 5 submitted
  - (ii) 2 presented

ii. Programs: Review of the upcoming annual conference, 2018 Houston programs

(1) Blake Ellis attempted to get all of the energy chairs together for a coordination of the sessions. These included TCs 1.10, 6.2, 6.9

(2) Open discussion for upcoming Houston sessions
- (a) Ideas discussed were
  - (i) Creation of a morning session to have all energy sessions concurrent.
  - (ii) District energy track. Topics discussed include, Natural Disasters, hurricanes, Weathering the storm, etc.

(3) January 2018 a general synopsis, four proposed sessions are to be reviewed.

  - iii. Concept for the Houston Conference would be to have a few local hospitals, such as MD Anderson, Methodist, etc. to discuss the issues around this topic. higher education could be another item. Example used was Juan at UTA from IDEA.
  - iv. Tours for Houston discussion by Michael Langton
    - (1) 45MW@Teco, NRG plant which serves the stadium, and a few others customers such as Texas Children's Hospital
    - (2) Local Airports.
    - (3) Potentially focus on natural disaster concepts
    - (4) Expressed was the need for tour volunteers
    - (5) Blake Ellis, proposed to make all tours available for the district energy track attendees.
- B. CEC Call for Reviewers: 2018 Annual Meeting and others. Contact Tiffany Cox (tcox@ashrae.org) and copy TC Chair & Vice Chair
- C. Seminar, workshop or forum
  - i. Open discussion of which all discussed which one is better and what the definitions of these are.
  - ii. The Chair discussed that at the TC breakfast this morning, it was explained that the presenters do not get CEUs for their presentation. Chair to review the NY PE rules.
  - iii. TC is in need of speakers for this, and decision on which format to propose to follow. should submit about two or three presentations. Chair discussed the need for someone to complete an abstract. A question posed, does the TC want other TCs to co-sponsor discussion.
  - iv. Terrance Raulins gave a workshop this morning on some case studies, interactive This was more focused on the O&M. TC 7.3, could help in the future with this.

4. Sub-Committee Reports:

- A. Handbook: Steve Tredinnick (Chair)
  - i. Chair reported to the group that he is using an authoring portal. Chapter 12 District Heating and Cooling and Chapter 13 for High Temperature Hot Water from TC 6.1 who are no longer completing updates.
  - ii. Steve will discuss with Gary ? and get formulas integrated. One year from now we should be voting on it. Topic should be resume in Houston meeting. Request was made for reviewers of the Chapters.
  - iii. If members need to access the authoring portal, they must be on the roster. The user can't use Chrome, a newer MS Word version, and the user can't use Edge. Follow up on how to get in- Authoring.ashrae.org use your ASHRAE password, must be on roster for the TC, corresponding can use it too.
- B. Membership: John Andrepont (Chair)
- C. Program: Tim Anderson (Chair)
- D. Standards: Open
- E. Webmaster: Jessica Mangler
- F. Research: Jay Eldridge (Chair)
  - i. This committee is member of TC. Discussed were topic of District Cooling for Combustion. Chair of TC 1.10. This was approximately $75,000 for the project cost with $5,000 is from combustion contributors.

5. New Business-

- A. Jeff Sloan discussed ASHRAE's Standard 90.4 committee, a new Energy Standard for Data Centers.
- B. Discussion of district energy plant scope inclusion opportunities in the 90.1 standard.
  - i. There is a coordination meeting tomorrow, 2018-01-22 at 7:30am between 90.1 and TC 6.2. This incudes Gary Ash, Tim (MD Anderson), Representative from Arkansas State.
  - ii. It was discussed but generally, the TC is not in favor of a plant efficiency standard. It's inherent by using the other standards, except for pipe sizing. TC representative needs to attend that meeting.

6. Meeting Adjourned, to be resumed at 3:00 pm.

7. The Chair Called to Order, and discussed the following.

A. Visitors announced
B. Attendance sign in list
C. ASHRAE Code of Ethics Commitment by TC Chairman – Alan Neely
D. Voting Quorum discussed. TC needs two additional Voting Members.
   i. To be Voted on: Long Beach Meeting Minutes and New Members.
E. Chicago Sub-Committee Meeting
   i. Three to four presentations under TCs District Energy topic.
   ii. Abstracts are due 2018-02-19.
F. Programs: Review of the upcoming annual conference, 2018 Houston programs
   i. Creation of a morning session to have all energy sessions concurrent.
   ii. District energy track. Topics discussed include, Natural Disasters, hurricanes, Weathering the storm, etc.
G. Papers: Motion from Jessica Mangler to approve the TC 1.10 paper. TC 6.2 to co-sponsor three papers from other TCs. Motion seconded by Geoffrey Bares, all other (5) voting members approved, none declined.

8. Liaison Discussion:
A. Section Head of TC 90.4
B. Earl Williams, past Chair of TC 6.2
C. DOE-
   i. The Technical Assistance partnership, will be funded next year, they are waiting for the budget. Government is currently shutdown. They want resiliency in various jurisdictions. They are cresting a web based e catalog which will be piloted this year. For anyone who wants to provide a package, take standard data and make a performance system, it reduces risk, and reduces cost and time.
D. IDEA-
   i. He described what idea is and about the Campus energy coming up. There is no conflict with ASHRAE conference
E. Section head of TC 6.
   i. Will help with submissions and new items. Authors will be allowed another submission by 2018-03-15. Innovated research grants, 41 submissions, they awarded to TC 6.7
   ii. They need the full TC approval for processing, it's stated to best discuss in-person.

9. Sub-Committee Reports:
A. Research:
   i. Discussed were topic of District Cooling for Combustion. Updating the District Cooling Design Guide. Guide to add in Middle East design specifics. The second update is intended for the building owner of District Cooling. Concern was expressed that this guide will not include reviewers from this TC. The ASHRAE contact person is Steve Comstock.
B. Handbook: Steve Tredinnick (Chair)
   i. Chair reported to the group Chapter 12 and Chapter 15 updates. Revised equations need updating from the Design Guide with Handbook. Chapter 15 requires some grammatical updates. The Sub-Committee needs a projector to use at the Houston meeting.
C. Membership: John Andrepont (Chair)
   i. The minimum number of voting members required is eight. Post this Winter meeting, the Chair is to update the TC roster. Submit emergency paperwork to update non-voting members to become voting membesr. Use the website to request TC membership.
D. Webmaster: Jessica Mangler
   i. The website has recently been updated. Please review.

10. New Business:

A. Open Discussion:
   i. Guest from the Standard 90.4 committee
      (1) Has a coordination request amongst TCs. Data centers use energy and he would like to correspond with anyone who has ideas on this topic. Standard 90.4 for CHP is a good idea. In Seattle, Amazon has a small microgrid system which has made this topic relevant for discussion.
   ii. For District energy plants Standard 90.1 doesn't apply. There may be opportunities for inclusion of the Standard 90.1 scope in the standard for inclusion of the joint TCs. Equipment efficiency only, not for process. It was mentioned that energy metering be included.
      (1) Gary Ash from OSU discussed his experiences with District Heating flash steam losses. He uses DDC monitoring on all steam traps to monitor real time.
   iii. The scheduled meeting tomorrow morning, 2018-01-22, may be cancelled.
   iv. Chair to schedule a conference call to discuss between the TCs, preferably in March 2018.

11. Motion to Adjourn Meeting
   A. Announcement by chair Alan to close this session, motion by Daniel Pyewell, all vote 5-0-1.

<u>Subcommittee Meetings</u>
Day, Date, Times, & Location Prior to TC 6.2 Meeting:

TC 1.08 Handbook - Sunday, 2018-01-21, 8:00 -10:30 am, Palmer House
TC 1.08 Program - Sunday, 2018-01-21, 10:30 -11:00 am, Palmer House
TC 1.08 Research - Sunday, 2018-01-21, 11:00 - 12:00 pm, Palmer House

ASHRAE Winter Conference
TC 6.02 District Energy
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IDEA Liaison Report

January 9, 2019

IDEA Liaison Report to ASHRAE TC 6.2 and TC 1.10

Prepared by: John Andrepont (+1-630-353-9690 / CoolSolutionsCo@aol.com)

The International District Energy Association (IDEA) is an industry association of >2,300 members. The membership includes: District Energy (Thermal) Utilities; Physical Plant and Utilities Personnel from University, Medical, and other District Energy Facilities; Equipment Manufacturer/Suppliers; Service Providers to the Industry; other District Heating & Cooling Associations; plus Government, Student, and other Personal members.

IDEA publishes a quarterly magazine, entitled *District Energy*.

IDEA has two major conferences each year (one focused on campus District Energy systems and one focused on commercial thermal utility District Energy systems), as well as smaller workshops on subjects such as marketing and thermal distribution, plus occasional regionally-focused or technology-focused conferences. Each major conference also focuses on integrating Combined Heat & Power (CHP) and Thermal Energy Storage (TES) with District Energy.

Upcoming IDEA conferences include:

- 32nd Annual Campus Energy Conference, Workshops & Trade Show
  February 25-March 1, 2019 – New Orleans, Louisiana
- 110th Annual Conference & Trade Show
  June 24-27, 2019 – Pittsburgh, Pennsylvania
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ASHRAE Handbook

CHAPTER 12

DISTRICT HEATING AND COOLING

DISTRICT heating and cooling (DHC) or district energy (DE) distributes thermal energy from a central source to residential, commercial, and/or industrial consumers for use in space heating, cooling, water heating, and/or process heating. The energy is distributed by steam or hot- or chilled-water lines. Thus, thermal energy comes from a distribution medium rather than being generated on site at each facility.[1]

According to Pierce (1994), district heating was proposed in 1613 in London; however, the first known examples were hot-water systems installed in St. Petersburg, Russia, in the 1840s. In the United States, the first district heating system was installed at the Naval Academy in Annapolis, Maryland, in 1853. Birdsell Holly is often credited with inventing district heating in Lockport, New York, in 1877, but he was simply the first to make it a commercial enterprise, ultimately expanding it to other cities. Most major U.S. cities soon were served by steam-based district heating systems; the majority of those systems still survive today, the largest in New York City. Although steam systems were also built in a number of major European cites, hot-water-based district heating saw significant growth in Europe, in part because of the reconstruction after World War II and the construction or expansion of many U.S. military bases, which were an ideal application for district heating. Both steam and high-temperature hot-water systems were built on bases. In Europe, especially Scandinavia, hot-water-based district heating systems adopted low temperatures for supply; that trend continues today, with systems using supply temperatures of 158°F (70°C) or lower.

Early attempts at district cooling date back to the 1880s (Pierce 1994). By the 1930s, commercial systems were being built. Although recent development in district cooling had been confined mostly to the United States, there has been increased international activity, notably in the Middle East and in Europe. The International District Energy Association (IDEA) represents both heating and cooling utilities and reported that approximately 86% of the conditioned building space added in the last six years by its member systems was added outdoor of the United States; the vast majority of that growth was district cooling development in the Middle East (IDEA 2008a).

Whether the system is a public utility or user owned, such as a multibuilding campus, it has economic and environmental benefits depending somewhat on the particular application. Political feasibility must be considered, particularly if a municipality or governmental body is considering a DHC installation. Historically, successful DHC systems have had the political backing and support of the community. IDEA (2008a) has many applicable suggestions in developing and designing district cooling systems.

Applicability

District heating and cooling systems are best used in markets where (1) the thermal load density is high and (2) the annual load factor or operating hours are high. A high load density is needed to cover the capital investment for the transmission and distribution system, which usually constitutes a significant portion of the capital cost for the overall system, often amounting to 50% or more of the total cost for district heating systems (normally lower for district cooling applications).

The annual load factor is important because the total system is capital intensive. These factors make district heating and cooling systems most attractive in serving (1) industrial complexes, (2) densely populated urban areas, and (3) high-density building clusters with high

[1]The preparation of this chapter is assigned to TC 6.2, District Energy.

1. SYSTEM MASTER PLANNING

Planning a "green field" district energy system or mapping out growth scenarios for an existing system takes a great deal of effort in gathering information regarding not only the customer base, but also plant sites, distribution routing, fuel sources and sensitivities, levels of redundancy, etc. This planning phase is typically referred to as **master planning**. Master plans take a long-term view of system development and growth, investigating many parameters and potential road blocks to development and growth. For a more complete explanation of the master planning process, see Chapter 2 of ASHRAE's *District Cooling Guide* and *District Heating Guide* (Phetteplace et al. 2013a, 2013b).

At a minimum, a master plan should provide a prioritized program for long-term guidance for building, expanding, and upgrading the district systems, which are typically built incrementally. A good master plan serves as a technically sound marketing tool for the owner's engineers to present needs and solutions to management or to prospective customers. Unfortunately, many owners view utility master plans as an interesting technical exercise with a life of one or two years. When this has been an owner's experience, it is usually because of at least one of the following reasons:

- Failure to involve the owner's staff
- Failure to provide intermediate owner reviews
- Use of an unverified database
- Lack of creativity in developing technically sound system alternatives for screening and final selection by the owner
- Inaccurate cost estimation, often related to overly optimistic estimation using unit costs that do not include all elements of the systems

The process for developing a master plan may be likened to a pyramid (Figure 2) (Bahnfleth 2004). The success of the plan depends on the foundation: a strong, accurate database that includes discovery and verification. All other aspects of a master plan stand on this basis and get their credibility from it. With the database in place, alternatives are identified and the preliminary estimates of cost (screening grade) for each option are used to work with the owner to select the most promising choices. These selections are then subjected to more intense analysis before making the final decisions of how the new plant (or expansion of an existing plant) will be developed, as well as how plans for future projects will be laid out. Thus, the pinnacle of the pyramid is a prioritized, priced list of projects needed to both keep pace with the physical growth of a facility and to provide replacements and upgrades to the existing system, if there is one.

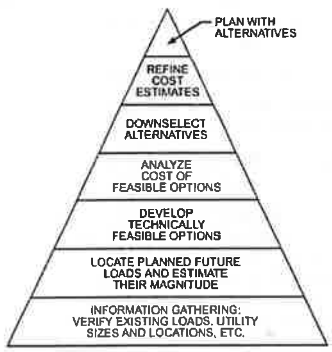

Fig. 2 Master Planning Pyramid
(Bahnfleth 2004)

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Typical technical issues addressed in master plans include the following:

- Code and regulator implications
- Phases of development and construction
- Other uses of thermal energy (i.e., absorption chiller for steam system summer use)
- Planned growth of system and potential of building use change
- Siting of central plant (noise issues, survivability during severe weather event, flooding, aesthetics)
- Topography
- Fuel availability, days of on-site storage, fuel handling, and cost
- Equipment selection (chillers, boilers, cooling towers, etc.)
- Distribution system routing and material selection
- Heating and cooling medium options (system temperatures and pressures)
- Type of consumer interconnection (direct or indirect)
- Metering technology and invoicing scheme

Temperature and pressure can dramatically affect the economic feasibility of a DHC system design. If the temperature and/or pressure level chosen is too low, a potential customer base might be eliminated. On the other hand, if there is no demand for absorption chillers or high-temperature industrial processes, a low-temperature system usually provides the lowest delivered energy cost.

The availability and location of fuel sources must also be considered in optimizing the economic design of a DHC system. For example, a natural gas boiler might not be feasible where abundant sources of natural gas are not available.

Initial Capital Investment. The initial capital investment for a DHC system is usually the major economic driving force in determining whether there is acceptable payback for implementation. Normally, the initial capital investment includes concept planning and design phases as well as the construction costs of the three major system components: (1) thermal energy production plant, (2) distribution system, and (3) consumer interconnections [also known as energy transfer stations (ETSs)].

Concept Planning. In concept planning phase, many areas are generally reviewed, and the technical feasibility of a DHC system is considered. This includes master planning and estimating system thermal loads and load growth potential, prospective plant site locations, piping routing, and interconnection or conversion requirements in the customer's building. The overall system concept design usually requires the services of an experienced power plant or DHC engineering firm.

Financial Feasibility. The overall capital and operating costs of the DHC system determine the energy rates charged to the prospective customers. These rates must be competitive with the customer's alternative HVAC system life-cycle costs, so a detailed analysis of this system tailored to the nature of the district energy provider is required to determine the system's financial feasibility. For example, a municipal or governmental body must consider availability of bond financing; if the entity is a private/for-profit organization, then the appropriate discount rates must be considered for the financing. Review **alternative energy choices** and **fuel flexibility**, because potential consumers are often asked to sign long-term contracts to justify a DHC system. Fuel flexibility offers the DE provider a method to keep generating costs low by being able to react to spikes in fuel costs. The financial analysis must take into account the equipment's life-cycle costs, including initial construction, operating, maintenance, and replacement, over a system life of at least 50 years.

Final Design. This phase is extremely intensive and may take several years to decades to complete the vision of the district energy provider. All technical assumptions for the design parameters must be confirmed (e.g., temperatures, pressures, flows, loads) while preparing the construction documents for the plant, distribution system, and customer interconnections. The importance of the consumer building's HVAC system water temperatures or steam pressures should not be overlooked, because they essentially drive the selection and sizing of the central plant equipment and distribution materials. Multiple building surveys may be required to determine the piping's exact entry point and routing to the appropriate mechanical or pump room.

System Construction and Costs. The accuracy of construction cost estimates for the central plant, distribution system, and FTS designs depends on the quality and detail of the upfront planning and final design efforts. Although the construction cost usually accounts for most of the initial capital investment, neglect or over/underestimation in any of the other three planning phases could mean the difference between economic success and failure of the DHC system. As with any construction project, field changes usually increase the final cost and delay start-up; therefore, identifying all of the design issues and schedule in advance is extremely important. Even a small delay in start-up can adversely affect both economics and consumer confidence. It is also extremely important that the contractors have experience commensurate with the project difficulty.

Although the plant is a major component of the construction costs, the distribution system also accounts for a significant portion of this initial investment. Distribution design depends on the heat transfer medium chosen, its operating temperature and pressure, and the routing. This system component also has the most risk associated with its installation because it can encounter many underground obstructions. Often, utilities are not well documented in streets, and piping alignment conflicts are not identified until excavation begins. Potholing or exploratory excavations can save project time and costs when used appropriately. Failure to consider these key variables results in higher-than-planned installation costs. An analysis is recommended to determine the distribution system's insulation requirements to determine the required insulating properties and thickness. The section on Economical Thickness for Pipe Insulation discusses determining insulation values.

DHC project costs vary greatly and depend on local construction environment and site conditions such as

- Labor rates
- Construction environment (i.e., slow or busy period)
- Distance to ship equipment
- Permits and fees (e.g., franchise fees)
- Local authorities (e.g., traffic control, times of construction in city streets)
- Soil conditions (e.g., clay, bedrock)
- Quality of equipment and controls (e.g., commercial or industrial)
- Availability of materials
- Size of distribution piping system
- Type of insulation or cathodic protection for buried and aboveground piping system
- Type of distribution system installation (e.g., direct buried, tunnel)
- Distribution system depth of burial and restoration of existing conditions (e.g., city streets, green areas)
- Distribution systems below-grade conflict resolutions
- Economies of scale

Sample construction cost unit pricing is as follows. Typically, the lower unit costs are for larger systems. The designer is cautioned that costs can vary widely because of size, complexity, location, and economic conditions:

- Cooling plant (building, chillers, cooling towers, pumps, piping, controls) = $1800 to $3500 per ton ($500 to $1000 per kilowatt)
- Boiler plant (building, boilers, stacks, pumps, piping, controls) = $1500 to $2300 per boiler horsepower ($150 to $230 per kilowatt)
- Distribution systems (includes excavation, backfill, surface restoration, piping, etc.):

central plant activities	time during unplanned outages.
Replacement or refilling of refrigerants	If refrigerant is scheduled for phasing out, chillers must be retrofitted to accept new refrigerant plus any topping off of refrigerants (or replacement).
Spare parts and supplies	Chiller and boiler and auxiliary equipment require replacement of parts (e.g., gears, oil, tubes) for normal maintenance procedures.
Cost of chemical treatment for steam, chilled, condenser, and hot-water systems	Includes scale and corrosion inhibitors, biocides, oxygen scavengers, etc.; these costs could be considerable.
Cost of contracted maintenance	Some owners outsource specific tasks to service companies such as chiller or boiler maintenance and overhauls.
Energy and Resource Usage	
Peak heating and cooling thermal loads	Used to apply energy demand rate and sizing of plant equipment (chillers, boilers, pumps, electrical service, water service, etc.).
Annual heating and cooling usage	Used to apply utilities' energy consumption rate to equipment meeting thermal loads.
Annual water and sewer usage	Quantify makeup water usage and blowdown discharge pertinent to cooling towers and boilers.
Other Costs	
Architectural and engineering design services	Specifically for new or retrofit applications.
Fees and licenses	Air and water permits, high-pressure steam operator licenses, city franchise fees for running piping in street, etc.
Insurance of equipment	Typically a percentage of construction costs.

Because there are benefits to the building owner that cannot be assigned a monetary value, the comparison between the two alternatives should be a value-based decision. In other words, the analysis should not only include quantitative variables, but also qualitative variables and benefits that have intrinsic value to the proposition. For example, some qualitative variables that add value to a building connecting to district energy system could include

- Some of the mechanical and electrical space can be repurposed or rented out for other than storage (e.g., office space, roof garden)
- No plumes from cooling towers or boiler stacks
- Increased thermal energy source reliability
- More stable energy costs
- Other than a possible demand charge, a customer is only billed for energy used (metered)
- Fewer greenhouse gas emissions and smaller carbon footprint
- Not having any equipment idling and using energy in hot standby
- Freeing maintenance staff to perform other duties other than central plant operations

Of course, the building owner must determine whether any or all of these parameters pertain or are valuable in a particular building.

See Chapter 37 in the 2015 *ASHRAE Handbook—HVAC Applications* for more detailed explanation of preparing a life-cycle cost calculation.

Example 1. A building owner is evaluating two different methods of providing chilled water for cooling an office building: purchasing chilled water from a local district cooling provider, or installing a conventional chiller plant. The building load is estimated to be 2400 tons (8440 kW) with an annual cooling load of 6,264,000 ton-hours (22 030 500 kWh). The contract is for 25 years, the discount rate is 5.5%, and all costs but water/sewer will be escalated at 3.5%, with water/sewer at 10% per year (based on historical data from municipality).

BLCC5.3-18 was used as the financial modelling tool for this example. The National Institute of Standards and Technology (NIST) developed the Building Life Cycle Cost (BLCC) programs to provide computational support for the analysis of capital investments in buildings. BLCC5 is a free downloadable program from: https://www.energy.gov/eere/femp/building-life-cycle-cost-programs. BLCC5 combines several parameters into "recurring" costs which includes financing, operator salaries, insurance, equipment replacement escrow funds, chemical treatment costs, electricity savings and annual equipment maintenance costs. Values included in the recurring costs are denoted with an asterisk (*) in Table 3.

For alternative 1, the district cooling provider charges are $285 ($81.20) capacity charge ($/ton/year ($/kW/year) applied to peak load) and $0.13 ($0.037) consumption charge ($/ton·h ($/kWh)) for a total annual energy cost of $1,498,320. Table 2 summarizes the major cost components (obtained from a local contractor) for each alternative. The interconnection charge (heat exchanger, piping, instrumentation, etc.) to the district cooling entity is $289,500. The building owner has decided to pay for this cost by financing it over the life of the contract in lieu of a lump-sum basis.

Table 2 Annual Utility Consumption Summary for Central Plant Alternatives

Utility	Alternative 1	Alternative 2
Electrical, kWh	0	4,389,950 (4 389 950)
Water, 1000 gal (m^3)	0	16,181 (61 252)
Sewer, 1000 gal (m^3)	0	3,773 (14 282)

For alternative 2, the on-site chilled water plant (chillers, pumps, piping and cooling tower, etc.) is estimated to cost $8,981,000, with an expected life of 25 years; 90% of the cost will be financed at 5.5% interest rate. The owner has instructed the engineer to increase the chiller sizes (three at 900 tons (3165 kW)) to accommodate for any future growth, add a little redundancy, and compensate for equipment aging. The cost estimate reflects this. An escrow account for major chiller plant overhaul ($400/ton ($113/kW)) was established and is expensed annually. Annual cost for preventative maintenance ($6.00/ton ($1.70/kW) for electrical chillers) was obtained from the local chiller vendor, there is one operator assigned to the plant with an annual salary of $99,000 (includes benefits burden of 40%), and water and sewer charges at $4/1000 gal ($1/m³) for water and $4/1000 gal ($1/m³) for blowdown to sewer and chemical treatment ($0.0025/ton·h ($0.0007/kWh)). The cost of insurance per year is based on 0.75% of the total construction costs.

Table 2 summarizes the annual utility use for each alternative, and Table 3 estimates annual maintenance and utility costs and commercial tower costs. The chiller plant (chillers, cooling towers, pumps, etc., but not including the distribution pumps) uses 4,389,950 kWh annually at a blended electrical rate of $0.10/kWh. An energy analysis determined that a district energy connection will reduce the electrical demand dramatically with a new

	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	...	23	24	25
~~Savings~~																				
~~Net annual cash flow~~	~~$0~~	~~$1,330,586~~	~~$1,376,400~~	~~$1,423,819~~	~~$1,472,897~~	~~$1,523,694~~	~~$1,576,267~~	~~$1,630,682~~	~~$1,687,000~~	~~$1,745,289~~	~~$1,805,619~~	~~$1,868,060~~	~~$1,932,688~~	~~$1,999,576~~	~~$2,068,806~~	~~$2,140,458~~	—	~~$2,811,738~~	~~$2,909,393~~	~~$3,01[illegible],467~~

25-year Net Present Value $25,575,051; Sum of 25-year Cash Flows = ~~$51,525,050~~52,991,400

Alternative 2: Design and Install On-Site 2700 ton (8440 KW) Chilled-Water System

	Year																			
	0	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**...**	**23**	**24**	**25**
Initial cost	$898,100	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
Recurring Cost ~~Financing plant first cost~~	$0 ~~$602,575~~	$923,570 ~~$602,575~~	$934,797 ~~$602,575~~	$946,416 ~~$602,575~~	$958,442 ~~$602,575~~	$970,924 ~~$602,575~~	$983,807 ~~$602,575~~	$997,141 ~~$602,575~~	$1,010,941 ~~$602,575~~	$1,025,264 ~~$602,575~~	$1,040,047 ~~$602,575~~	$1,055,348 ~~$602,575~~	$1,071,184 ~~$602,575~~	$1,087,620 ~~$602,575~~	$1,104,585 ~~$602,575~~	$1,122,143 ~~$602,575~~	—	$1,263,626 ~~$602,575~~	$1,286,746 ~~$602,575~~	$1,3[illegible] ~~$602,576~~
Energy Cost ~~Equip. replacement fund~~	$0 ~~$0~~	$517,264 ~~$44,712~~	$535,356 ~~$46,277~~	$554,080 ~~$47,897~~	$573,459 ~~$49,573~~	$593,572 ~~$51,308~~	$614,333 ~~$53,104~~	$635,820 ~~$54,962~~	$658,058 ~~$56,886~~	$681,138 ~~$58,877~~	$704,961 ~~$60,938~~	$729,618 ~~$63,071~~	$755,137 ~~$65,278~~	$781,622 ~~$67,563~~	$808,959 ~~$69,928~~	$837,253 ~~$72,375~~	—	$1,065,245 ~~$95,304~~	$1,102,502 ~~$98,640~~	$1,1[illegible]0,956 ~~$102,[illegible]92~~
Water Cost ~~Electricity cost~~	$0 ~~$0~~	$1,535,306 ~~$438,995~~	$1,573,015 ~~$454,359~~	$1,612,550 ~~$470,262~~	$1,654,028 ~~$486,724~~	$1,697,695 ~~$503,756~~	$1,743,444 ~~$521,388~~	$1,791,539 ~~$539,637~~	$1,842,133 ~~$558,524~~	$1,895,547 ~~$578,072~~	$184,634 ~~$598,305~~	$203,084 ~~$619,245~~	$223,377 ~~$640,919~~	$245,763 ~~$663,354~~	$270,322 ~~$686,568~~	$297,335 ~~$710,598~~	—	$579,459 ~~$935,722~~	$637,364 ~~$968,473~~	$700,871 ~~$1,00[illegible],369~~
Sewer Cost ~~Chemical cost~~	$0 ~~$0~~	$923,570 ~~$16,775~~	$934,797 ~~$17,363~~	$946,416 ~~$17,970~~	$958,442 ~~$18,599~~	$970,924 ~~$19,250~~	$983,807 ~~$19,924~~	$997,141 ~~$20,621~~	$1,010,941 ~~$21,343~~	$1,025,264 ~~$22,090~~	$22,029 ~~$22,863~~	$22,800 ~~$23,663~~	$23,597 ~~$24,492~~	$24,425 ~~$25,349~~	$25,279 ~~$26,236~~	$26,163 ~~$27,154~~	—	$33,288 ~~$35,757~~	$34,452 ~~$37,008~~	$35,[illegible]54 ~~$38,[illegible]04~~
Net Annual Cash Flow ~~Water cost~~	$0 ~~$0~~	$517,264 ~~$78,318~~	$535,356 ~~$86,150~~	$554,080 ~~$94,765~~	$573,459 ~~$104,244~~	$593,572 ~~$114,665~~	$614,333 ~~$126,132~~	$635,820 ~~$138,745~~	$658,058 ~~$152,620~~	$681,138 ~~$167,882~~	$184,634 ~~$184,670~~	$203,084 ~~$203,137~~	$223,377 ~~$223,450~~	$245,763 ~~$245,795~~	$270,322 ~~$270,375~~	$297,335 ~~$297,412~~	—	$579,459 ~~$637,530~~	$637,364 ~~$701,283~~	$700,871 ~~$771,414~~
~~Sewer cost~~	~~$0~~	~~$18,259~~	~~$20,085~~	~~$22,094~~	~~$24,303~~	~~$26,733~~	~~$29,407~~	~~$32,347~~	~~$35,582~~	~~$39,140~~	~~$43,054~~	~~$47,360~~	~~$52,096~~	~~$57,305~~	~~$63,036~~	~~$69,339~~	—	~~$148,635~~	~~$163,198~~	~~$179,[illegible]48~~
~~Labor cost~~	~~$0~~	~~$102,473~~	~~$106,060~~	~~$109,772~~	~~$113,614~~	~~$117,590~~	~~$121,706~~	~~$125,966~~	~~$130,375~~	~~$134,938~~	~~$139,661~~	~~$144,549~~	~~$149,608~~	~~$154,844~~	~~$160,264~~	~~$165,873~~	—	~~$218,423~~	~~$226,068~~	~~$233,[illegible]80~~
~~Equipment operation and maintenance cost~~	~~$0~~	~~$17,354~~	~~$17,961~~	~~$18,590~~	~~$19,241~~	~~$19,914~~	~~$20,611~~	~~$21,332~~	~~$22,079~~	~~$22,852~~	~~$23,652~~	~~$24,479~~	~~$25,336~~	~~$26,223~~	~~$27,141~~	~~$28,091~~	—	~~$36,990~~	~~$38,285~~	~~$39,[illegible]25~~
~~Insurance cost~~	~~$0~~	~~$72,155~~	~~$74,680~~	~~$77,294~~	~~$80,000~~	~~$82,800~~	~~$85,698~~	~~$88,697~~	~~$91,801~~	~~$95,014~~	~~$98,340~~	~~$101,782~~	~~$105,344~~	~~$109,031~~	~~$112,847~~	~~$116,797~~	—	~~$153,799~~	~~$159,182~~	~~$164,[illegible]754~~
~~Net annual cash flow~~	~~$1,500,675~~	~~$1,391,616~~	~~$1,425,510~~	~~$1,461,249~~	~~$1,498,867~~	~~$1,538,591~~	~~$1,580,545~~	~~$1,624,882~~	~~$1,671,785~~	~~$1,721,440~~	~~$1,771,058~~	~~$1,829,861~~	~~$1,889,098~~	~~$1,952,036~~	~~$2,018,970~~	~~$2,090,244~~	—	~~$2,864,735~~	~~$2,995,012~~	~~$3,1[illegible]4,958~~

25-year Net Present Value $26,171,587; Sum of 25-year Cash Flows = ~~$53,034,510~~53,616,410

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Figure 3 summarizes the major components of the analysis graphically. Note how much the water and sewer costs comprise of the total costs, and that the energy costs only makes up one-third of the life-cycle costs.

- High-temperature hot-water (HTHW) systems supply temperatures over 350°F ([redacted]).
- Medium-temperature hot-water (MTHW) systems supply temperatures in the range of 250 to 350°F ([redacted]).
- Low-temperature hot-water (LTHW) systems supply temperatures of 250°F ([redacted]) or lower.

The temperature drop at the consumer end should be as high as possible, preferably 40°F ([redacted]) or greater. A large temperature drop allows the fluid flow rate through the system, pumping power, return temperatures, return line heat loss, and condensing temperatures in cogeneration power plants to be reduced. A large customer temperature drop can often be achieved by cascading loads operating at different temperatures. Typical Scandinavian LTHW systems have temperature drops of 70°F ([redacted]) or greater. Chapter 15 provides more information on medium- and high-temperature water-heating systems.

Furthermore, the Swedish district heating industry (Werner 2013) has some further definitions classifying the generation of systems based on the time period, heating media and temperatures used.

- 1st Generation – Steam based systems (1880-1930)
- 2nd Generation – Pressurized super-heated water (1930-1980)
- 3rd Generation – Pressurized water (1980-2020)
- 4th Generation – Pressurized water (2020-2050)

In many instances, existing equipment and processes require the use of steam, which precludes use of hot water or requires smaller high-pressure boilers at the buildings. See the section on Consumer Interconnections for further information.

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Steam and Hot Water Generation

With few exceptions, boilers are constructed to meet requirements in the sections on constructing heating and power boilers of the ASME *Boiler and Pressure Vessel Code* (2013). Low-pressure boilers are constructed for maximum working pressures of 15 psig ([redacted]) steam and up to 160 psig ([redacted]) hot water. Low-pressure hot-water boilers are limited to 250°F ([redacted]) operating temperature. High-pressure boilers are designed to operate above 15 psig ([redacted]) steam or above 160 psig and/or 250°F ([redacted]) for water boilers.

In the United States, boilers are rated by boiler horsepower, with 1 hp being equal to 33,475 Btu/h ([redacted]) or the evaporation of 34.5 lb_m ([redacted]) of water per hour at standard atmospheric pressure 14.7 psia and 212°F ([redacted]). Every steam or water boiler is rated for a maximum working pressure according to its applicable boiler code. When installed, it also must be equipped at a minimum with operation and safety controls and pressure-/temperature-relief devices mandated by such codes.

Fire-tube and water-tube boilers are available for gas/oil firing (see Chapter 32 for details). If coal is used, either package-type coal-fired boilers in small sizes (less than 20,000 to 25,000 lb/h ([redacted])) or field-erected boilers in larger sizes are available. Coal-firing underfeed stokers are available up to a 30,000 to 35,000 lb/h ([redacted]) capacity; traveling grate and spreader stokers are available up to 160,000 lb/h ([redacted]) capacity in single-boiler installations. Fluidized-bed boilers can be installed for capacities over 300,000 lb/h ([redacted]). Larger coal-fired boilers are typically multiple installations of the three types of stokers or larger, pulverized fired or fluidized-bed boilers. Generally, the complexity of fluidized bed or pulverized firing does not lend itself to small central heating plant operation. Coal-fired boilers typically can be partially cofired with other solid fuels such as wood biomass, refuse, or tire-derived fuels (TDF), or retrofitted to combust a higher percentage of nonfossil fuels.

Chilled-Water Generation

As with smaller in-building chilled-water plants, district plants have options for which chiller type, refrigerant, and prime drivers to use. Chiller types in district cooling systems vary by location, depending on parameters such as water availability, power availability, maximum power demand that can be offered by the utility, steam availability, gas or fuel oil availability, distribution temperature required, plant location with respect to development, and applied environment impact regulations (including pollution and noise control).

The chillers may be classified according to

- Refrigerant and compressor technology (centrifugal, screw, absorption, etc.)
- Heat rejection source (air cooled or water cooled)
- Driving energy source (electric motor, steam turbine, hot water or steam absorption, internal combustion engine, etc.)
- Supply temperature required
- Size: capacity in tons ([redacted])

Chiller condenser heat rejection can be air cooled or water cooled. **Air-cooled chillers** are typically packaged, with controls, compressors, evaporator, and air-cooled condenser all on the same skid. Their capacity may be as high as 450 tons ([redacted]), and they have been implemented in several central plants by installing multiple units in chiller farms, either on grade or on building rooftops.

Water-cooled chillers reject heat carried by the water circuit through either bodies of water (rivers, lakes, and oceans), cooling towers (evaporative cooling), or radiators. The limiting factor of the capacity of a cooling tower is the ambient wet-bulb temperature. Because the wet-bulb temperature is lower than the coincident dry-bulb temperature, a cooling tower with a water-cooled chiller is more efficient than an air-cooled one at removing chiller heat because of the lower condensing temperature: more energy can be rejected by evaporating one pound ([redacted]) of water than by raising the temperature of an equivalent amount of air. Because water-cooled chiller plants are typically considerably more efficient than either air-cooled or water-cooled radiator plants, they require less heat rejection area. Tables 5 and 6 summarize the available chiller technologies, efficiencies with first cost and maintenance costs.

Table 5 Chiller Technology

Compression Chillers	Absorption Chillers

Thermal Storage

Both hot- and chilled-water thermal storage can be implemented for district systems. In North America, the current economic situation primarily results in chilled-water storage applications. Depending on plant design and loading, thermal storage can reduce chiller equipment requirements and lower operating costs. By shifting part of the chilling load, chillers can be sized closer to the average load than the peak load. Shifting some or all of the refrigeration load to off peak reduces the on-peak electrical demand load while using the same (or slightly larger) chiller machine capacity. Because many utilities offer lower rates (and perhaps some rebates) during off-peak periods, operating costs for electrically driven chillers can be substantially reduced.

Both ice and chilled-water storage have been applied to district-sized chiller plants. In general, the largest systems (>20,000 ton-hour {[redacted]} capacity) use chilled-water (CHW) storage and small- to moderate-sized systems use ice storage. Storage capacities in the 10,000 to 30,000 ton-hour {[redacted]} range are now common and systems have been installed up to 125,000 ton-hour {[redacted]} for district cooling systems. When CHW storage is feasible, be careful not to reduce the chilled-water temperature below 39.2°F {[redacted]}, to allow proper temperature stratification in the thermal energy storage (TES) tank. A TES tank charging temperature lower than 39°F {[redacted]} without low-temperature additives will result in mixing in the tank, loss of tank stratification, and possibly disturbance of the system supply temperature and the storage concept.

For these reasons, most chilled-water plants are designed for a 40°F {[redacted]} supply temperature and a 56°F {[redacted]} or greater return temperature. Some plants may be designed for lower temperatures by using multiple cascaded heat exchangers in series, such as high-rise towers; however, note that each 1°F {[redacted]} reduction in supply temperature increases chiller-specific energy consumption (i.e., kW/ton {[redacted]}) by approximately 2%. Some plants are designed for a higher return temperature (60°F {[redacted]} or higher) to increase the Δt, but this requires a great deal of additional coordination with the design of the building's HVAC system to ensure the system operates per design intent.

In Europe, several cooling systems use naturally occurring underground aquifers (caverns) for storage of chilled water. Selection of the storage configuration (chilled-water steel tank above grade, chilled-water concrete tank below grade, ice direct, ice indirect) is often influenced by space limitations. Depending on the system design temperatures, chilled-water storage requires four to six times the volume of ice storage for the same capacity. For chilled-water storage, the footprint of steel tanks (depending on height) can be less than concrete tanks for the same volume (Andrepont 1995); furthermore, the cost of above-grade tanks is usually less than below-grade tanks.

Chapter 51 has additional information on thermal storage; for thermal storage specifically in district cooling and heating, also see Phetteplace et al. (2013a, 2013b), respectively.

Auxiliaries

Numerous pieces of auxiliary support equipment related to the boiler and chiller operations are not unique to the production plant of a DHC system and are found in similar installations. Some components of a DHC system deserve special consideration because of their critical nature and potential effect on operations.

Although instrumentation can be either electronic or pneumatic, electronic instrumentation systems offer the flexibility of combining control systems with data acquisition systems. This combination brings improved efficiency, better energy management, and reduced operating staff for the central heating and/or cooling plant. For systems involving multiple fuels and/or thermal storage, computer-based controls are indispensable for accurate decisions about boiler and chiller operation.

Boiler feedwater treatment has a direct bearing on equipment life. Condensate receivers, filters, polishers, and chemical feed equipment must be accessible for proper management, maintenance, and operation. Depending on the temperature, pressure, and quality of the heating medium, water treatment may require softeners, alkalizers, and/or demineralizers for systems operating at high temperatures and pressures.

Equipment and layout of a central heating and cooling plant should reflect what is required for proper plant operation and maintenance. The plant should have an adequate service area for equipment and a sufficient number of electrical power outlets and floor drains. Equipment should be placed on housekeeping pads to protect the bases from spills or leaks. Figure 4 presents a typical layout for a large hot-water/chilled-water plant. Notice that the layout provides space for future expansion as well as storage of spare parts. The control room is typically close to the operating equipment for ease of visual inspection. Other functions that should be considered include adequate space for maintenance, chemical treatment laboratory and chemical storage, and operator conveniences such as locker rooms and lunch rooms. Designers must follow the requirements of ASHRAE *Standard* 15 for ventilation as well as laying out the equipment room, and should coordinate with the architect regarding tight-sealing doors, minimizing penetrations, etc.

of the distribution system, and (3) design of the customer connection to the distribution system. Unless very unusual circumstances exist, most systems large enough to be considered in the district category are likely to benefit from variable-flow design. See Chapter 13 for additional information on pumping.

Constant Flow

In the past, constant chilled-water flow was applied only to smaller systems where simplicity of design and operation were important and where distribution pumping costs were low, before variable- and adjustable-speed drives were available and affordable. (However, for new systems, designers should refer to ASHRAE *Standard* 90.1, which requires variable-flow pumping.) Chillers were also arranged in series to handle higher system design temperature differentials. Flow rate through a full-load distribution system depended on the type of constant-flow system used. A common technique connected the building and its terminal units across the distribution system. The central plant pump circulated chilled water through air-side terminal units controlled by three-way valves (constant-volume direct primary pumping). Balancing problems could occur in this design when many separate flow circuits were interconnected (Figure 5).

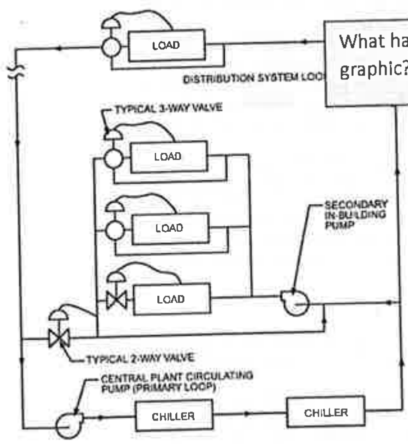

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Fig. 5 Constant-Flow Primary Distribution with Secondary Pumping

Constant-flow distribution was also applied to in-building (secondary or tertiary) circuits with separate pumps. This arrangement isolates the flow balance problem between buildings. In this case, flow through the distribution system could be significantly lower than the sum of the flows needed by the terminal if the in-building system supply temperature were higher than the distribution system supply temperature (Figure 5). The water temperature rise in the distribution system was determined by the connected in-building systems and their controls.

In constant-flow design, chillers arranged in parallel have decreased entering water temperatures at part load; thus, several machines might have needed to run simultaneously, each at a reduced load, to produce the required chilled-water flow. In this case, chillers in series were better because constant flow could be maintained though the chilled-water plant at all times, with only the chillers required for producing chilled water energized. Legacy constant-flow systems should be analyzed thoroughly when considering multiple chillers in a parallel arrangement, because the auxiliary electric loads of condenser water pumps, tower fans, and central plant circulating pumps are a significant part of the total energy input. Contemporary control systems mitigate the reason for constant flow through the chillers, and variable flow should be used for larger systems.

Variable Flow

Variable-flow design can significantly reduce energy use and expand the capacity of the distribution system piping by using diversity. To maintain a high temperature differential at part load, the distribution system flow rate must track the load imposed on the central plant. Multiple parallel pumps or, more commonly, variable-speed pumps can reduce flow and pressure, and lower pumping energy at part load. Terminal device controls should be selected to ensure that variable flow objectives are met. Correctly sized terminal unit flow-throttling (two-way) valves, and especially pressure-independent control valves (PICV), provide the continuous high return temperature needed to correlate the system load change to a system flow change.

Systems in each building are usually two-pipe, with individual in-building pumping. In some cases, the pressure of the distribution system may cause flow through the in-building system without in-building pumping. Distribution system pumps can provide total building system pumping if (1) the distribution system pressure drops are minimal, and (2) the distribution system is relatively short-coupled (3000 ft (1000 m) or less). To implement this pumping method, the total flow must be pumped at a pressure sufficient to meet the requirements of the building with the largest pressure differential requirement. Consequently, all buildings on the system should have their pressure

- Size the distribution system for a low overall total pressure loss. Short-coupled distribution systems (3000 ft {[redacted]} or less) can be used for a total pressure loss of 20 to 40 ft of water {[redacted]}. With this maximum differential between any points in the system, size the distribution pumps to provide the necessary pressure to circulate chilled water through the in-building systems, eliminating the need for in-building pumping systems. This decreases the complexity of operating central chilled-water systems. Newer controls on chillers enable all-variable-flow systems. Check with the manufacturer about minimum flows on chiller evaporators to achieve stable operation over all load ranges.
- All two-way valves must have proper close-off ratings and a design pressure drop of at least 20% of the maximum design pressure drop for controllability. Commercial quality automatic temperature control valves generally have low shutoff ratings; but industrial valves can achieve higher ratings. Three-way control valves should be avoided except to accommodate minimum pump flow and turndown. See Chapter 43 for more information on control valves.
- Although chillers can easily produce colder water, the lower practical limit for chilled-water supply temperatures is 39°F {[redacted]}. Temperatures below that should be carefully analyzed for energy usage, although systems with thermal energy storage may operate advantageously at lower temperatures.

3. DISTRIBUTION SYSTEM

3.1 HYDRAULIC CONSIDERATIONS

Objectives of Hydraulic Design

Although the distribution of a thermal utility such as hot water encompasses many of the aspects of domestic hot-water distribution, many dissimilarities also exist; thus the design should not be approached in the same manner. Thermal utilities must supply sufficient energy at the appropriate temperature and pressure to meet consumer needs. Within the constraints imposed by the consumer's end use and equipment, the required thermal energy can be delivered with various combinations of temperature and pressure. Computer-aided design methods are available for thermal piping networks (Bloomquist et al. 1999; COWIconsult 1985; Rasmussen and Lund 1987; Reisman 1985). Using these methods allows rapid evaluation of many alternative designs.

General steam system design can be found in Chapter 11, as well as in IDHA (1983) and Phetteplace et al. (2013b). For water systems, see Chapter 13, IDEA (2008b), IDHA (1983), and Phetteplace et al. (2013a, 2013b).

Water Hammer

The term water hammer is used to describe several phenomena that occur in fluid flow. Although these phenomena differ in nature, they all result in stresses in the piping that are higher than normally encountered. Water hammer can have a disastrous effect on a thermal utility by bursting pipes and fittings and threatening life and property.

In steam systems (IDHA 1983), water hammer is caused primarily by condensate collecting in the bottom of the steam piping. Steam flowing at velocities 10 times greater than normal water flow picks up a slug of condensate and accelerates it to a high velocity. The slug of condensate subsequently collides with the pipe wall at a point where flow changes direction. To prevent this type of water hammer, condensate must be prevented from collecting in steam pipes by using proper steam pipe pitch and adequate condensate collection and return facilities.

Water hammer also occurs in steam systems because of rapid condensation of steam during system warm-up. Rapid condensation decreases the specific volume and pressure of steam, which precipitates pressure shock waves. This form of water hammer is prevented by controlled warm-up of the piping. Valves should be opened slowly and in stages during warm-up. Large steam valves should be provided with smaller bypass valves to slow the warm-up.

Water hammer in hot- and chilled-water distribution systems is caused by sudden changes in flow velocity, which causes pressure shock waves. The two primary causes are pump failure (i.e., power failure) and sudden valve closures. A simplified method to determine maximum resultant pressure may be found in Chapter 22 of the 2013 *ASHRAE Handbook—Fundamentals*. More elaborate methods of analysis can be found in Fox (1977), Stephenson (1981), and Streeter and Wylie (1979). Preventive measures include operational procedures and special piping fixtures such as surge columns.

Pressure Losses

Frictional pressure losses occur at the interface between the inner wall of a pipe and a flowing fluid due to shear stresses. In steam systems, these pressure losses are compensated for with increased steam pressure at the point of steam generation. In water systems, pumps are used to increase pressure at either the plant or intermediate points in the distribution system. Calculation of pressure loss is discussed in Chapters 3 and 22 of the 2013 *ASHRAE Handbook—Fundamentals*. Hydraulic calculations reveal that a great deal of the system pressure drop is caused by pipe fittings and offsets; therefore, it is prudent to lay out distribution systems with as few offsets as possible. Some designers prefer two 45° elbows in lieu of a single 90° elbow, assuming that the pressure drop is lower. As pressure drop calculations show, this is not so.

Pipe Sizing

Ideally, the appropriate pipe size should be determined from an economic study of the life-cycle cost for construction and operation. In practice, however, this study is seldom performed because of the effort involved. Instead, criteria that have evolved from practice are frequently used for design. These criteria normally take the form of constraints on the maximum flow velocity or pressure drop. Chapter 22 of the 2013 *ASHRAE Handbook—Fundamentals* provides velocity and pressure drop constraints. Noise generated by excessive flow velocities

1. The "normal" condition used for the life-cycle cost analysis determines appropriate insulation thickness. Average values for the temperatures, burial depth, and thermal properties of the materials are used for design. If the thermal properties of the insulating material are expected to degrade over the useful life of the system, make appropriate allowances in the cost analysis.
2. Maximum heat transfer rate determines the load on the central plant due to the distribution system. It also determines the temperature drop (or increase, in the case of chilled-water distribution), which determines the delivered temperature to the consumer. For this calculation, each component's thermal conductivity must be taken at its maximum value, and the temperatures must be assumed to take on their extreme values, which would result in the greatest temperature difference between the carrier medium and the soil or air. The burial depth is normally at its lowest value for this calculation. During operation, none of the thermal capabilities of the materials (or any other materials in the area influenced thermally by the system) must exceed design conditions. To satisfy this objective, each component and the surrounding environment must be examined to determine whether thermal damage is possible. A numerical heat transfer analysis may be necessary in some cases.
3. The conditions of these analyses must be chosen to represent the worst-case scenario from the perspective of the component being examined. For example, in assessing the suitability of a coating material for a metallic conduit, the thermal insulation is assumed to be saturated, the soil moisture is at its lowest probable level, and the burial depth is maximum. These conditions, combined with the highest anticipated pipe and soil temperatures, give the highest conduit surface temperature to which the coating could be exposed.

Heat transfer in buried systems is influenced by the thermal conductivity of the soil and by the depth of burial, particularly when the insulation has low thermal resistance. Soil thermal conductivity changes significantly with moisture content; for example, Bottorf (1951) indicated that soil thermal conductivity ranges from 0.083 Btu/h·ft·°F {0.14 W/(m·K)} during dry soil conditions to 1.25 Btu/h·ft·°F {2.16 W/(m·K)} during wet soil conditions.

For details on calculating thermal effects on district energy distribution piping, see Phetteplace et al. (2013a, 2013b).

Thermal Properties of Pipe Insulation and Soil

Uncertainty in heat transfer calculations for thermal distribution systems results from uncertainty in the thermal properties of materials involved. Generally, the designer must rely on manufacturers' data to obtain approximate values. The data in this chapter should only be used as guidance in preliminary calculations until specific products have been identified; then specific data should be obtained from the manufacturer of the product in question.

Insulation. Insulation provides the primary thermal resistance against heat loss or gain in thermal distribution systems. Thermal properties and other characteristics of insulations normally used in thermal distribution systems are listed in Table 7. Material properties such as thermal conductivity, density, compressive strength, moisture absorption, dimensional stability, and combustibility are typically reported in ASTM standards for the respective material. Some properties have more than one associated standard. For example, thermal conductivity for insulation material in block form may be reported using ASTM *Standards* C177, C518, or C1114. For piping containing hot media, thermal conductivity for insulation material fabricated or molded for use on piping is measured using ASTM *Standard* C335.

Table 7 Comparison of Commonly Used Insulations in Underground Piping Systems

	Calcium Silicate Type I/II ASTM C533	Urethane Foam	Cellular Glass ASTM C552	Mineral Fiber/Preformed Glass Fiber Type 1 ASTM C547
Thermal conductivity[a] (Values in parentheses are maximum permissible by ASTM standard listed), Btu/h·ft·°F				
Mean temp. = 100°F	0.028	0.013	0.033 (0.030)	0.022 (0.021)
200°F	0.031 (0.038/0.045)	0.014	0.039 (0.037)	0.025 (0.026)
300°F	0.034 (0.042/0.048)		0.046 (0.045)	0.028 (0.033)
400°F	0.038 (0.046/0.051)		0.053 (0.054)	(0.043)
Density (max.), lb/ft³	15/22		6.7 to 9.2	8 to 11
Maximum temperature, °F	1200	250	800	850
Compressive strength (min),[b] psi	100 at 5% deformation		60	N/A
Dimensional stability, linear shrinkage at maximum use temperature	2%		N/A	2%
Flame spread	0		5	25
Smoke index	0		0	50
Water absorption	As-shipped moisture content, 20% max. (by weight)		0.5	Water vapor sorption, 5% max. (by weight)

[a]Thermal conductivity values in this table are from previous editions of this chapter and have been retained as they were used in examples. Thermal conductivity of insulation may vary with temperature, temperature gradient, moisture content, density, thickness, and shape. ASTM maximum values given are comparative for establishing quality control compliance, and are suggested for preliminary calculations where actual values are not available. They may not represent installed performance of insulation under actual conditions that may differ substantially from test conditions. The manufacturer should be able to supply appropriate design values.
[b]Compressive strength for cellular glass shown is for flat material, capped as per ASTM *Standard* C240.

Table 7 Comparison of Commonly Used Insulations in Underground Piping Systems

Calcium Silicate Type I/II ASTM C533 | Urethane Foam | Cellular Glass ASTM C552 | Mineral Fiber/Preformed Glass Fiber Type 1 ASTM

Table 8 Effect of Moisture on Underground Piping System Insulations

Characteristics	Polyurethane[a]	Cellular Glass	Mineral Wool[b]	Fibrous Glass
Heating Test	Pipe temp. 1.8 to 127°C Water bath 8 to 38°C	Pipe temp. 1.8 to 215°C Water bath 8 to 38°C	Pipe temp. 1.8 to 230°C Water bath 8 to 38°C	Pipe temp. 1.8 to 230°C Water bath 8 to 38°C
Length of submersion time to reach steady-state k-value	70 days	[c]	10 days	2 h
Effective k-value increase from dry conditions after steady state achieved in submersion	14 to 19 times at steady state. Estimated water content of insulation 70% (by volume).	Avg. 10 times, process unsteady.[c] Insulation showed evidence of moisture zone on inner diameter.	Up to 50 times at steady state. Insulation completely saturated.	52 to 185 times. Insulation completely saturated at steady state.
Primary heat transfer mechanism	Conduction	[c]	Conduction and convection	Conduction and convection
Length of time for specimen to return to dry steady-state k-value after submersion	Pipe at 127°C, after 16 days moisture content 10% (by volume) remaining	Pipe at 215°C, 8 h	Pipe at 230°C, 9 days	Pipe at 193°C, 6 days
Cooling Test	Pipe temp. 2.8°C Water bath at 11°C	Pipe temp. 2.2°C Water bath 8 to 14°C	Pipe temp. 1.8 to 7°C Water bath 13°C	Insulation 1.8 to 230°C Water bath 8 to 38°C
Length of submersion time to reach steady-state conditions for k-value	16 days	Data recorded at 4 days constant at 12 days	6 days	1/2 h
Effective k-value increase from dry conditions after steady state achieved	2 to 4 times. Water absorption minimal, ceased after 7 days.	None. No water penetration.	14 times. Insulation completely saturated at steady state.	20 times. Insulation completely saturated at steady state.
Primary heat transfer mechanism	Conduction	Conduction	Conduction and convection	Conduction and convection
Length of time for specimen to return to dry steady-state k-value after submersion	Pipe at 3.3°C, data curve extrapolated to 10+ days	Pipe at 0.6°C, no change	Pipe at 1.8°C, data curve extrapolated to 25 days	Pipe at 1.8°C, 15 days

Source: Chyu et al. (1997a, 1997b; 1998a, 1998b). ***This item requires renewed permission from the copyright holder to reprint.***

[a]Polyurethane material tested had a density of 46 kg/m^3.

[b]Mineral wool tested was a preformed molded basalt designed for pipe systems operating up to 650°C. It was specially formulated to withstand the Federal Agency Committee (FAC) 96 h boiling water test. See Phetteplace (2013b) for the FAC test protocol.

[c]Cracks formed on heating for all samples of cellular glass insulation tested. Flooded heat loss mechanism involved dynamic two-phase flow of water through cracks; period of dynamic process was about 20 min. Cracks had negligible effect on thermal conductivity of dry cellular glass insulation before and after submersion. No cracks formed during cooling test.

Painting chilled-water piping before insulating is recommended in areas of high humidity. Insulations used today for chilled water include polyurethane, polyisocyanurate, phenolics, cellular plastics, and fiberglass. No insulation system is totally water- and vaportight; thus, water from the atmosphere can enter the insulation system in small amounts, which can cause corrosion of the pipe. Chloride ions present in most atmospheric water exacerbate corrosion. The best way to minimize corrosion is to make the insulation system highly water resistant by using a closed-cell insulation material coupled with a high-performance vapor retarder, and painting the pipe exterior with a strong rust-preventative coating (two-part epoxy) before insulating. This is good engineering practice and most insulation manufacturers suggest this, but it may not be in their literature. In addition, a good vapor retarder is required on the exterior of the insulation.

When assessing the potential for pipe corrosion under insulation, the pipe operating temperature must be considered. Pipes operating permanently or predominantly at temperatures below about 50°F {[redacted]} or above 250°F {[redacted]} are not prone to corrosion. Pipes operating between these temperatures, particularly if they are subject to temperature cycling in use or from frequent shutdowns, are most susceptible to corrosion.

Soil. If a soil analysis is available or can be done, the thermal conductivity of the soil can be estimated from published data [e.g., Farouki (1981); Lunardini (1981)]. The thermal conductivity factors in Table 9 may be used as an estimate when detailed information on the soil is not known. Because dry soil is rare in most areas, only assume a low moisture content for system material design, or where it can be validated for calculation of heat losses in the normal operational condition. Values of 0.8 to 1 Btu/h·ft·°F {[redacted]} are commonly used where soil moisture content is unknown. Because moisture migrates toward a chilled pipe, use a thermal conductivity value of 1.25 Btu/h·ft·°F {[redacted]} for chilled-water systems in the absence of any site-specific soil data. For steady-state analyses, only the thermal conductivity of the soil is required. If a transient analysis is required, the specific heat and density are also required.

Table 9 Soil Thermal Conductivities

	Thermal Conductivity, Btu/h·ft·°F {[redacted]}		
Soil Moisture Content (by mass)	**Sand**	**Silt**	**Clay**
Low, <4%	0.17 {[redacted]}	0.08 {[redacted]}	0.08 {[redacted]}
Medium, 4 to 20%	1.08 {[redacted]}	0.75 {[redacted]}	0.58 {[redacted]}
High, >20%	1.25 {[redacted]}	1.25 {[redacted]}	1.25 {[redacted]}

c_s = dry soil specific heat, Btu/lb·°F {[redacted]}
c_w = specific heat of water = 1.0 Btu/lb·°F {[redacted]}
w = moisture content of soil, % (dry basis)
k_s = soil thermal conductivity, Btu/h·ft·°F {[redacted]}

Because the specific heat of dry soil is nearly constant for all types of soil, c_s may be taken as 0.175 Btu/lb·°F {[redacted]}.

4. For buried systems, the undisturbed soil temperatures may be estimated for any time of the year as a function of depth, soil thermal properties, and prevailing climate. This temperature may be used in lieu of the soil surface temperature normally called for by the steady-state heat transfer equations when estimates of heat loss/gain as a function of time of year are desired. Substituting the undisturbed soil temperatures at the pipe depth allows the steady-state equations to be used as a first approximation to the solution to the actual transient heat transfer problem with its annual temperature variations at the surface. The following equation may be used to estimate the undisturbed soil temperature at any depth at any point during the yearly weather cycle (ASCE 1996). (*Note*: The argument for the sine function is in radians.)

$$t_{s,z} = t_{ms} - A_s \, e^{-z\sqrt{\pi/\alpha\tau}} \sin\left[\frac{2\pi(\theta - \theta_{lag})}{\tau} - z\sqrt{\frac{\pi}{\alpha\tau}}\right] \qquad (4)$$

where

θ = Julian date, days
θ_{lag} = phase lag of soil surface temperature, days

Use Equation (3) to calculate soil thermal diffusivity. Values for the climatic constants t_{ms}, A_s, and θ_{lag} may be found at tc62.ashraetcs.org/pdf/ASHRAE_Climatic_Data.pdf for all worldwide weather stations included in the CD accompanying the 2009 *ASHRAE Handbook—Fundamentals*. Phetteplace et al. (2013b) contains equations that may be used to calculate the climatic constants for any weather data set, real or contrived.

Equation (4) does not account for latent heat effects from freezing, thawing, or evaporation. However, for soil adjacent to a buried heat distribution system, the equation provides a good estimate, because heat losses from the system tend to prevent the adjacent ground from freezing. For buried chilled-water systems, freezing may be a consideration; therefore, systems that are not used or drained during the winter months should be buried below the seasonal frost depth. For simplicity, the ground surface temperature is assumed to equal the air temperature, which is an acceptable assumption for most design calculations. If a more accurate calculation is desired, use the following method to compensate for the convective thermal resistance to heat transfer at the ground surface.

Convective Heat Transfer at Ground Surface

Heat transfer between the ground surface and the ambient air occurs by convection. In addition, heat transfer with the soil occurs due to precipitation and radiation. The heat balance at the ground surface is too complex to warrant detailed treatment in the design of buried district heating and cooling systems. However, in some circumstances, approximations that included impacts beyond an average convective heat transfer coefficient were found. For example, McCabe et al. (1995) observed significant temperature variations caused by the type of surfaces over district heating and cooling systems. Phetteplace et al. (2013a) contains methods to approximate the impacts of surface type.

Normally, only convection is considered, and as a first approximation, an **effective thickness** of a fictitious soil layer may be added to the burial depth to account for the effect of the convective heat transfer resistance at the ground surface. The effective thickness is calculated as follows:

$$\delta = k_s/h \qquad (5)$$

where

δ = effective thickness of fictitious soil layer, ft {[redacted]}
k_s = thermal conductivity of soil, Btu/h·ft·°F {[redacted]}
h = convective heat transfer coefficient at ground surface, Btu/h·ft²·°F {[redacted]}

The effective thickness calculated with Equation (5) is simply added to the actual burial depth of the pipes in calculating the soil thermal resistance using Equations (6), (7), (20), (21), and (27).

Uninsulated Buried Pipe

For this case (Figure 7), an estimate for soil thermal resistance may be used. This estimate is sufficiently accurate (within 1%) for the ratios of burial depth to pipe radius indicated next to Equations (6) and (7). Both the actual resistance and the approximate resistance are presented, along with the depth/radius criteria for each.

q = heat loss or gain per unit length of system, Btu/h·ft {[redacted]}

The negative result indicates a heat gain rather than a loss. Note that the thermal resistance of the fluid/pipe interface has been neglected, which is a reasonable assumption because such resistances tend to be very small for flowing fluids. Also note that, in this case, the thermal resistance of the pipe comprises a significant portion of the total thermal resistance. This results from the relatively low thermal conductivity of PVC compared to other piping materials and the fact that no other major thermal resistances exist in the system to overshadow it. If any significant amount of insulation were included in the system, its thermal resistance would dominate, and it might be possible to neglect that of the piping material.

Insulated Buried Pipe

Equation (8) can be used to calculate the thermal resistance of any concentric circular region of material, including an insulation layer. When making calculations using insulation thickness, use actual rather than nominal thickness to obtain the most accurate results.

Example 3. Consider the effect of adding 1 in. {[redacted]} of urethane foam insulation and a 1/8 in. {[redacted]} thick PVC jacket to the chilled-water line in Example 2. Calculate the thermal resistance of the insulation layer from Equation (8) as follows:

$$R_i = \frac{\ln(0.229/0.146)}{2\pi \times 0.0125} = 5.75 \text{ h·ft·°F/Btu}$$ {[redacted]}

For the PVC jacket material, use Equation (8) again:

$$R_j = \frac{\ln(0.240/0.229)}{2\pi \times 0.10} = 0.07 \text{ h·ft·°F/Btu}$$ {[redacted]}

The thermal resistance of the soil as calculated by Equation (7) decreases slightly to R_s = 0.51 h·ft·°F/Btu {[redacted]} because of the increase in the outer radius of the piping system. The total thermal resistance is now

$$R_t = R_p + R_i + R_j + R_s = 0.21 + 5.75 + 0.07 + 0.51 = 6.54 \text{ h·ft·°F/Btu}$$

{[redacted]}

Heat gain by the chilled-water pipe is reduced to about 2 Btu/h·ft {[redacted]}. In this case, the thermal resistance of the piping material and the jacket material could be neglected with a resultant error of <5%. Considering that the uncertainties in the material properties are likely greater than 5%, it is usually appropriate to neglect minor resistances such as those of piping and jacket materials if insulation is present.

Buried Pipe in Conduit with Air Space

Systems with air spaces (Figure 8) may be treated by adding an appropriate resistance for the air space. For simplicity, assume a heat transfer coefficient of 3 Btu/h·ft²·°F {[redacted]} (based on the outer surface area of the insulation), which applies in most cases. The resistance caused by this heat transfer coefficient is then

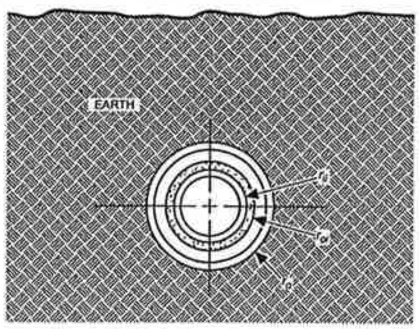

Fig. 8 Insulated Buried Pipe with Air Space

$$R_a = 1/(3 \times 2\pi \times r_{oi}) = 0.053/r_{oi} \quad (9)$$ {[redacted]}

where

r_{oi} = outer radius of insulation, ft {[redacted]}
R_a = resistance of air space, h·ft·°F/Btu {[redacted]}

A more precise value for the resistance of an air space can be developed with empirical relations available for convection in enclosures such as those given by Grober et al. (1961). Consider the effect of radiation in the annulus when high temperatures are expected in the air space. For the treatment of radiation, refer to Siegel and Howell (1981).

Then use Equation (1) to calculate the maximum soil temperature at the installation depth as follows:

$$t_{r,z} = 71.8 + 15.0\exp\left[-10\sqrt{\frac{\pi}{0.20 \times 365}}\right] = 73.7°\text{F}$$ {[redacted]}

Now calculate the first estimates of the thermal resistances of the pipe insulations. For calcium silicate, assume a mean temperature of the insulation of 300°F {[redacted]} to establish its thermal conductivity. From the data listed previously, calcium silicate's thermal conductivity k_i = 0.042 Btu/h·ft·°F {[redacted]} at this temperature. For the polyurethane foam, assume the insulation's mean temperature is 200°F {[redacted]}. From the data listed previously, the polyurethane foam's thermal conductivity k_i = 0.014 Btu/h·ft·°F {[redacted]} at this temperature. Now calculate the insulation thermal resistances using Equation (8):

$$\text{Calcium silicate } R_{i,1} = \frac{\ln(0.401/0.276)}{2\pi(0.042)} = 1.42 \text{ h·ft·°F/Btu}$$ {[redacted]}

$$\text{Polyurethane foam } R_{i,2} = \frac{\ln(0.484/0.401)}{2\pi(0.014)} = 2.14 \text{ h·ft·°F/Btu}$$ {[redacted]}

Calculate the thermal resistance of the soil from Equation (7):

$$R_s = \frac{\ln(2 \times 10/0.526)}{2\pi(0.2)} = 2.89 \text{ h·ft·°F/Btu}$$ {[redacted]}

The total thermal resistance is

$$R_t = 1.42 + 2.14 + 2.89 = 6.45 \text{ h·ft·°F/Btu}$$ {[redacted]}

The first estimate of the heat loss is then

$$q = (400 - 73.7)/6.45 = 50.6 \text{ Btu/h·ft}$$ {[redacted]}

Next, calculate the estimated insulation surface temperature with this first estimate of the heat loss. Find the temperature at the interface between the calcium silicate and polyurethane foam insulations.

$$t_{io,1} = t_{po} - qR_{i,1} = 400 - (50.6 \times 1.42) = 328°\text{F}$$ {[redacted]}

where t_{po} is the outer surface temperature of the pipe and $t_{io,1}$ is the outer surface temperature of first insulation (calcium silicate).

$$t_{io,2} = t_{po} - q(R_{i,1} + R_{i,2}) = 400 - 50.6(1.42 + 2.14) = 220°\text{F}$$ {[redacted]}

where $t_{io,2}$ is the outer surface temperature of second insulation (polyurethane foam).

The new estimate of the mean insulation temperature is (400 + 328)/2 = 364°F {[redacted]} for the calcium silicate and (328 + 220)/2 = 274°F {[redacted]} for the polyurethane foam. Thus, the insulation thermal conductivity for the calcium silicate is interpolated to be 0.045 Btu/h·ft·°F {[redacted]}, and the resulting thermal resistance is $R_{i,1}$ = 1.32 h·ft·°F/Btu {[redacted]}.

For the polyurethane foam insulation, the thermal conductivity is interpolated to be 0.015 Btu/h·ft·°F {[redacted]}, and the resulting thermal resistance is $R_{i,2}$ = 2.00 h·ft·°F/Btu {[redacted]}. The soil thermal resistance remains unchanged, and the heat loss is recalculated as q = 52.5 Btu/h·ft {[redacted]}.

The calcium silicate insulation outer surface temperature is now approximately $t_{io,1}$ = 331°F {[redacted]}, and the outer surface temperature of the polyurethane foam is calculated to be $t_{io,2}$ = 226°F {[redacted]}. Because these temperatures are within a few degrees of those calculated previously, no further calculations are needed.

The maximum temperature of the polyurethane insulation of 331°F {[redacted]} occurs at its inner surface (i.e., the interface with the calcium silicate insulation). This temperature clearly exceeds the maximum accepted 30-year service temperature of polyurethane foam of 250°F {[redacted]} (CEN *Standard* EN 253). Thus, the amount of calcium silicate insulation needs to be increased significantly to achieve a maximum temperature for the polyurethane foam insulation within its long-term service temperature limit. Under the conditions of this example, it would take about 5 in. {[redacted]} of calcium silicate insulation to reduce the insulation interface temperature to less than 250°F {[redacted]}.

Example 6 examines the effects of soil thermal conductivity and decreasing burial depth for a system with composite insulation and an air space.

Example 6. As in Example 5, assume a high-temperature water line operating at 400°F {[redacted]} installed in southern Texas. Here it also consists of a 6 in. {[redacted]} nominal diameter (6.625 in. outer diameter {[redacted]}) insulated carrier pipe, but the first layer of insulation is initially proposed to be 1.5 in. {[redacted]} of mineral wool. The carrier pipe and mineral wool insulation are contained inside a 0.125 in. {[redacted]} thick steel conduit with a 1 in. {[redacted]} air space between the insulation exterior and the conduit interior. On the exterior of the steel conduit is 1 in. {[redacted]} polyurethane foam insulation. The polyurethane insulation is encased in a 0.25 in. {[redacted]} thick high-density polyethylene (HDPE) jacket. The piping system is buried 10 ft {[redacted]} deep to the pipe centerline. The initial assumption for the burial depth and soil thermal properties is the same as those used in Example 5, but the effect of those design parameters will be examined. Neglect the thermal resistances of the pipe, conduit, and HDPE jacket.

Mineral wool thermal conductivity
- 0.026 Btu/h·ft·°F at 200°F {[redacted]}
- 0.033 Btu/h·ft·°F at 300°F {[redacted]}
- 0.043 Btu/h·ft·°F at 400°F {[redacted]}

Polyurethane foam thermal conductivity
- 0.013 Btu/h·ft·°F at 100°F {[redacted]}
- 0.014 Btu/h·ft·°F at 200°F {[redacted]}
- 0.015 Btu/h·ft·°F at 275°F {[redacted]}

Assumed soil properties
- Thermal conductivity = 0.2 Btu/h·ft·°F {[redacted]}
- Density (dry soil) = 105 lb/ft³ {[redacted]}

of course also reduces their heat loss. The formulation for calculating heat losses for adjacent supply and return lines presented later in this chapter does not allow for direct calculation of system temperatures. As an approximation, it is suggested that the individual heat losses for the supply pipe, where the limiting condition will occur, be first calculated using Equations (18) to (25) (shown in Example 8). To approximate the system temperatures, use the methods outlined in this example as if the pipeline were buried by itself, but adjust the soil temperature upward to achieve the same rate of heat loss as determined by Equations (18) to (25) for the supply pipe in the supply and return configuration. This somewhat underestimates the impacts of the adjacent return pipeline, so be conservative when using this method.

Examples 5 and 6 show the need to fully consider burial conditions when assessing systems that have limitations on any of their component temperatures that are below the carrier fluid's operating temperature. Although the examples here are from systems with multiple insulations with temperature limitation on one of the insulations, this consideration also applies to any system, with or without composite insulation. Nonmetallic jacket materials as well as corrosion-resistant coatings are other examples of materials that often have temperature limitations well below carrier pipe temperatures encountered in practice. Deep burial of systems and/or burial of systems in low-thermal-conductivity soil requires that calculations such as those from Examples 5 and 6 be conducted where materials with temperature limitations below the carrier pipe temperature exist.

Two Pipes Buried in Common Conduit with Air Space

For this case (Figure 9), make the same assumption as made in the Buried Pipe in Conduit with Air Space section. For convenience, add some of the thermal resistances as follows:

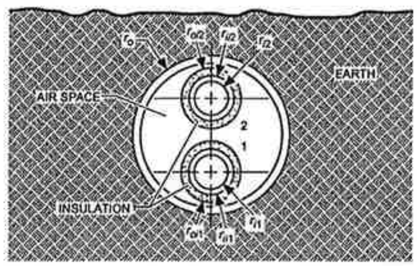

Fig. 9 Two Pipes Buried in Common Conduit with Air Space

$$R_1 = R_{p1} + R_{i1} + R_{a1} \tag{10}$$

$$R_2 = R_{p2} + R_{i2} + R_{a2} \tag{11}$$

Subscripts 1 and 2 differentiate between the two pipes within the conduit. The combined heat loss is then given by

$$q = \frac{[(t_{f1} - t_s)/R_1] + \left[(t_{f2} - t_s)/R_2\right]}{1 + (R_{cs}/R_1) + (R_{cs}/R_2)} \tag{12}$$

where R_{cs} is the total thermal resistance of conduit shell and soil.

Once the combined heat flow is determined, calculate the bulk temperature in the air space:

$$t_a = t_s + qR_{cs} \tag{13}$$

Then calculate the insulation outer surface temperature:

$$t_{i1} = t_a + (t_{f1} - t_a)(R_{a1}/R_1) \tag{14}$$

$$t_{i2} = t_a + (t_{f2} - t_a)(R_{a2}/R_2) \tag{15}$$

The heat flows from each pipe are given by

$$q_1 = (t_{f1} - t_a)/R_1 \tag{16}$$

$$q_2 = (t_{f2} - t_a)/R_2 \tag{17}$$

When the insulation thermal conductivity is a function of its temperature, as is usually the case, an iterative calculation is required, as shown in the following example.

Example 7. A pair of 4 in. (100 mm) NPS medium-temperature hot-water supply and return lines run in a common 21 in. (530 mm) outside diameter conduit. Assume that the supply temperature is 325°F (160°C) and the return temperature is 225°F (100°C). The supply pipe is insulated with 2.5 in. (60 mm) of mineral wool insulation, and the return pipe has 2 in. (50 mm) of mineral wool insulation. This insulation has the same thermal properties as those given in Example 4. (The insulation's thermal conductivity is 0.038 W/(m·K) at 50°C) The pipe is buried 4 ft (1.25 m) to centerline in soil with a thermal conductivity of 1 Btu/h·ft·°F (1.7 W/(m·K)). Assume the thermal resistance of the pipe, the conduit, and the conduit coating are negligible. As a first estimate, assume the bulk air temperature within the conduit is 100°F (40°C). In addition, use this temperature as a first estimate of the insulation surface temperatures to obtain the mean insulation temperatures and subsequent insulation thermal conductivities.

$$P_2 = \frac{1}{2\pi k_s}\ln\left[\frac{(d_2+d_1)^2+a^2}{(d_2-d_1)^2+a^2}\right]^{0.5} \tag{21}$$

where a = horizontal separation distance between centerline of two pipes, ft {m}.

The thermal resistance for each pipe or conduit is given by

$$R_{e1} = \frac{R_{t1}-(P_1^2/R_{t2})}{1-(P_1\theta_1/R_{t2})} \tag{22}$$

$$R_{e2} = \frac{R_{t2}-(P_2^2/R_{t1})}{1-(P_2\theta_2/R_{t1})} \tag{23}$$

where

θ = temperature correction factor, dimensionless
P = geometric/material correction factor, h·ft·°F/Btu {(m·K)/W}
R_e = effective thermal resistance of one pipe/conduit in two-pipe system, h·ft·°F/Btu {(m·K)/W}
R_t = total thermal resistance of one pipe/conduit if buried separately, h·ft·°F/Btu {(m·K)/W}

Heat flow from each pipe is then calculated from

$$q_1 = (t_{p1}-t_s)/R_{e1} \tag{24}$$

$$q_2 = (t_{p2}-t_s)/R_{e2} \tag{25}$$

Example 8. Consider buried supply and return lines for a low-temperature hot-water system. The carrier pipes are 4 in. {100 mm} NPS (4.5 in. {112 mm} outer diameter) with 1.5 in. {40 mm} of urethane foam insulation. The insulation is protected by a 0.25 in {6 mm} thick PVC jacket. The thermal conductivity of the insulation is 0.013 Btu/h·ft·°F {0.023 W/(m·K)} and is assumed constant with respect to temperature. The pipes are buried 4 ft {1.2 m} deep to the centerline in soil with a thermal conductivity of 1 Btu/h·ft·°F {1.7 W/(m·K)} and a mean annual temperature of 60°F {15°C}. The horizontal distance between the pipe centerlines is 2 ft {600 mm}. The supply water is at 250°F {120°C}, and the return water is at 150°F {65°C}.

Solution: Neglect the thermal resistances of the carrier pipes and the PVC jacket. First, calculate the resistances from Equations (7) and (8) as if the pipes were independent of each other:

$$R_{s1} = R_{s2} = \boxed{\frac{\ln(8.0/0.333)}{2\pi\times 1.0}} = 0.51\ \text{h·ft·°F/Btu}\ \left\{R_{s1}=R_{s2}=\frac{\ln[(2\times 1.2)/(102\times 10^{-3})]}{2\pi\times 1.7}=0.30\ \text{(m·K)/W}\right\}$$

$$R_{i1} = R_{i2} = \boxed{\frac{\ln(0.313/0.188)}{2\pi\times 0.013}} = 6.25\ \text{h·ft·°F/Btu}\ \left\{R_{i1}=R_{i2}=\frac{\ln[(112/2+40)/56]}{2\pi\times 0.023}=3.73\ \text{(m·K)/W}\right\}$$

$$R_{t1} = R_{t2} = 0.51 + 6.25 = 6.76\ \text{h·ft·°F/Btu}\ \{R_{t1}=R_{t2}=0.30+3.73=4.03\ \text{(m·K)/W}\}$$

From Equations (20) and (21), the correction factors are

$$P_1 = P_2 = \boxed{\frac{1}{2\pi\times 1}\ln\left[\frac{(4+4)^2+2^2}{(4-4)^2+2^2}\right]^{0.5}} = 0.225\ \text{h·ft·°F/Btu}\ \left\{P_1=P_2=\frac{1}{2\pi\times 1.7}\ln\left[\frac{(1.2+1.2)^2+0.6^2}{(1.2-1.2)^2+0.6^2}\right]^{0.5}=0.133\ \text{(m·K)/W}\right\}$$

$$\theta_1 = (150-60)/(250-60) = 0.474\ \{\theta_1=(65-15)/(120-15)=0.476\}$$

$$\theta_2 = 1/\theta_1 = 2.11\ \{\theta_2=1/\theta_1=2.10\}$$

Calculate the effective total thermal resistances as

$$R_{e1} = \boxed{\frac{6.76-(0.225^2/6.76)}{1-(0.225\times 0.474/6.76)}} = 6.87\ \text{h·ft·°F/Btu}\ \left\{R_{e1}=\frac{4.03-(0.133^2/4.03)}{1-(0.133\times 0.476/4.03)}=4.09\ \text{(m·K)/W}\right\}$$

$$R_{e2} = \boxed{\frac{6.76-(0.225^2/6.76)}{1-(0.225\times 2.11/6.76)}} = 7.32\ \text{h·ft·°F/Btu}\ \left\{R_{e2}=\frac{4.03-(0.133^2/4.03)}{1-(0.133\times 2.10/4.03)}=4.33\ \text{(m·K)/W}\right\}$$

The heat flows are then

$$q_1 = (250-60)/6.87 = 27.7\ \text{Btu/h·ft}\ \{q_1=(120-15)/4.09=25.7\ \text{W/m}\}$$

$$q_2 = (150-60)/7.32 = 12.3\ \text{Btu/h·ft}\ \{q_2=(65-15)/4.33=11.5\ \text{W/m}\}$$

$$q_t = 27.7 + 12.3 = 40.0\ \text{Btu/h·ft}\ \{q_t=25.7+11.5=37.2\ \text{W/m}\}$$

$$q = \frac{(t_{p1} - t_s)/R_1 + (t_{p2} - t_s)/R_2}{1 + (R_{ss}/R_1) + (R_{ss}/R_2)} \tag{29}$$

where

R_1, R_2 = thermal resistances of two-pipe/insulation systems within trench/tunnel, h·ft·°F/Btu {[redacted]}
R_{ss} = total thermal resistance on soil side of air within trench/tunnel, h·ft·°F/Btu {[redacted]}

Once the total heat loss has been found, the air temperature in the trench/tunnel may be found as

$$t_{ta} = t_s + qR_{ss} \tag{30}$$

where t_{ta} is air temperature in the trench/tunnel.

The individual heat flows for the two pipes in the trench/tunnel are then

$$q_1 = (t_{p1} - t_{ta})/R_1 \tag{31}$$

$$q_2 = (t_{p2} - t_{ta})/R_2 \tag{32}$$

If the thermal conductivity of the pipe insulation is a function of temperature, assume an air temperature for the air space before starting calculations. Iterate the calculations if the air temperature calculated with Equation (30) differs significantly from the initial assumption.

Example 9. The walls of a buried trench are 6 in. {[redacted]} thick, and the trench is 3 ft {[redacted]} wide and 2 ft {[redacted]} tall. The trench is constructed of concrete, with a thermal conductivity of k_w = 1 Btu/h·ft·°F {[redacted]}. The soil surrounding the trench also has a thermal conductivity of k_s = 1 Btu/h·ft·°F {[redacted]}. The centerline of the trench is 4 ft {[redacted]} below grade, and the soil temperature is assumed be 60°F {[redacted]}. The trench contains supply and return lines for a medium-temperature water system with the physical and operating parameters identical to those in Example 7.

Solution: Assuming the air temperature within the trench is 100°F {[redacted]}, the thermal resistances for the pipe/insulation systems are identical to those in Example 7, or

$$R_1 = 4.61 \text{ h·ft·°F/Btu}$$ {[redacted]}

$$R_2 = 4.26 \text{ h·ft·°F/Btu}$$ {[redacted]}

The thermal resistance of the soil surrounding the trench is given by Equation (27):

$$R_{u} = \boxed{\frac{\ln[(3.5 \times 4)/(3^{0.75} \times 4^{0.25})]}{1[(4/6) + 5.7]}} = 0.231 \text{ h·ft·°F/Btu}$$ {[redacted]}

The trench wall thermal resistance is calculated with Equation (26):

$$R_w = 0.5/[2(3 + 2)] = 0.050 \text{ h·ft·°F/Btu}$$ {[redacted]}

If the thermal resistance of the air/trench wall is neglected, the total thermal resistance on the soil side of the air space is

$$R_{ss} = R_w + R_u = 0.050 + 0.231 = 0.281 \text{ h·ft·°F/Btu}$$ {[redacted]}

Find a first estimate of the total heat loss using Equation (29):

$$q = \boxed{\frac{(325 - 60)/4.61 + (225 - 60)/4.26}{1 + (0.281/4.61) + (0.281/4.26)}} = 85.4 \text{ Btu/h·ft}$$ {[redacted]}

The first estimate of the air temperature in the trench is given by Equation (30):

$$t_{ta} = 60 + (85.4 \times 0.281) = 84.0\text{°F}$$ {[redacted]}

Refined estimates of the pipe insulation surface temperatures are then calculated using Equations (14) and (15):

$$t_{s1} = 84.0 + [(325 - 84.0)(0.13/4.61)] = 90.8\text{°F}$$

$$t_{s2} = 84.0 + [(225 - 84.0)(0.15/4.26)] = 89.0\text{°F}$$

{[redacted]}

{[redacted]}

From these estimates, calculate the revised mean insulation temperatures to find the resultant resistance values. Repeat the calculation procedure until satisfactory agreement between successive estimates of the trench air temperature is obtained. Calculate the individual heat flows from the pipes with Equations (31) and (32).

If the thermal resistance of the trench walls is added to the soil thermal resistance, the thermal resistance on the soil side of the air space is

$$R_{ss} = \boxed{\frac{\ln[14/(2^{0.75} \times 3^{0.25})]}{1[(3/4) + 5.7]}} = 0.286 \text{ h·ft·°F/Btu}$$ {[redacted]}

The result is less than 2% higher than the resistance previously calculated by treating the trench walls and soil separately. In the event that the soil and trench wall material have significantly different thermal conductivities, this simpler calculation will not yield as favorable results and should not be used.

[redacted]}. Using Equation (8), the thermal resistance of the insulation is R_i = 3.25 h·ft·°F/Btu {[redacted]}. The forced convective heat transfer coefficient at the surface of the insulation can be found using the following equation (ASTM *Standard* C680):

$$h_{cv} = 1.016\boxed{\left(\frac{1}{d}\right)^{0.2}\left(\frac{2}{t_{oi}+t_a}\right)^{0.181}}(t_{oi}-t_a)^{0.266}(1+1.277\,V)^{0.5}$$

$$= 1.016\boxed{\left(\frac{1}{11.625}\right)^{0.2}\left(\frac{2}{160}\right)^{0.181}}(40)^{0.266}(6.11)^{0.5}$$

$$= 1.86\ \text{Btu/h·ft}^2\text{·°F}$$

{[redacted]}

where

d = outer diameter of surface, in. {[redacted]}
t_a = ambient air temperature, °F {[redacted]}
V = wind speed, mph {[redacted]}

The radiative heat transfer coefficient must be added to this convective heat transfer coefficient. Determine the radiative heat transfer coefficient as follows (ASTM *Standard* C680):

$$h_{rad} = \varepsilon\sigma\frac{(T_a^4 - T_s^4)}{T_a - T_s}$$

$$h_{rad} = 0.26 \times 1.713 \times 10^{-9}\boxed{\frac{(560^4 - 520^4)}{560-520}} = 0.28\ \text{Btu/h·ft}^2\text{·°F}$$ {[redacted]}

Add the convective and radiative coefficients to obtain a total surface heat transfer coefficient h_t of 2.14 Btu/h·ft²·°F {[redacted]}. The equivalent thermal resistance of this heat transfer coefficient is calculated from the following equation:

$$R_{surf} = \boxed{\frac{1}{2\pi r_{oi} h_t} = \frac{1}{2\pi \times 0.484 \times 2.14}} = 0.15\ \text{h·ft·°F/Btu}$$ {[redacted]}

With this, the total thermal resistance of the system becomes R_t = 3.40 h·ft·°F/Btu {[redacted]}, and the first estimate of the heat loss is q = 92.6 Btu/h·ft {[redacted]}.

An improved estimate of the insulation surface temperature is t_{oi} = 375 – (92.6 × 3.25) = 74°F {[redacted]}. From this, a new mean insulation temperature, insulation thermal resistance, and surface resistance can be calculated. The heat loss is then 90.0 Btu/h·ft {[redacted]}, and the insulation surface temperature is calculated as 77°F {[redacted]}. These results are close enough to the previous results that further iterations are not warranted.

Note that the contribution of thermal radiation to the heat transfer could have been omitted with negligible effect on the results. In fact, the entire surface resistance could have been neglected and the resulting heat loss would have increased by only about 4%.

In Example 11, the convective heat transfer was forced. In cases with no wind, where the convection is free rather than forced, the radiative heat transfer is more significant, as is the total thermal resistance of the surface. However, in instances where the piping is well insulated, the thermal resistance of the insulation dominates, and minor resistances can often be neglected with little resultant error. By neglecting resistances, a conservative result is obtained (i.e., the heat transfer is overpredicted).

Economical Thickness for Pipe Insulation

A life-cycle cost analysis may be run to determine the economical thickness of pipe insulation. Because the insulation thickness affects other parameters in some systems, each insulation thickness must be considered as a separate system. For example, a conduit system or one with a jacket around the insulation requires a larger conduit or jacket for greater insulation thicknesses. The cost of the extra conduit or jacket material may exceed that of the additional insulation and is therefore usually included in the analysis. It is usually not necessary to include excavation, installation, and backfill costs in the analysis.

A system's life-cycle cost is the sum of the initial capital cost and the present worth of the subsequent cost of heat lost or gained over the life of the system. The initial capital cost needs only to include those costs that are affected by insulation thickness. The following equation can be used to calculate the life-cycle cost:

$$\text{LCC} = \text{CC} + (q t_u C_h \text{PWF}) \quad (33)$$

where

LCC = present worth of life-cycle costs associated with pipe insulation thickness, \$/ft {[redacted]} material should design loops and offsets in conduit and poured-envelope systems because clearance and design features are critical to the performance of both the loop and the pipe.

All expansion joints require maintenance, and should therefore always be accessible for service. Joints in direct-buried and poured- envelope systems and trenches without removable covers should be located in access ports.

Cold springing is normally used when thermal expansion compensation is used. In DHC systems with natural flexibility, cold springing minimizes the clearance required for pipe movement only. The pipes are sprung 50% of the total amount of movement, toward the anchor. However, ASME *Standard* B31.1 does not allow cold springing in calculating the stresses in the piping. When expansion joints are used, they are installed in an extended position to achieve maximum movement. Contact the manufacturer of the expansion joint for the proper amount of extension.

In extremely hot climates, anchors may also be required, to compensate for pipe contraction when pipes are installed in high ambient temperatures and then filled with cold water. This can affect buried tees in the piping, especially at branch service line runouts to buildings. Crushable insulation may be used in the trench as part of the backfill, to compensate for the contractions. Anchors should be sized using computer-aided design software.

Pipe Supports, Guides, and Anchors

For premanufactured conduit and poured-envelope systems, the system manufacturer usually designs the pipe supports, guides, and anchors in consultation with the expansion joint manufacturer, if such devices are used. For example, the main anchor force of an in-line axial expansion joint is the sum of the pressure thrust (system pressure times the cross-sectional area of the expansion joint and the joint friction or spring force) and the pipe friction forces. Consult the manufacturer of the expansion device when determining anchor forces. Anchor forces are normally less when expansion is absorbed through the system instead of with expansion joints.

Pipe guides used with expansion joints should be spaced according to the manufacturer's recommendations. They must allow longitudinal or axial motion and restrict motion perpendicular to the axis of the pipe. Guides with graphite or low-friction fluorocarbon slide surfaces are often desirable for long pipe runs (Figure 12). In addition, these surface finishes do not corrode or increase sliding resistance in aboveground installations. Select guides to handle twice the expected movement, so they may be installed in a neutral position without the need for cold-springing the pipe. A computerized stress analysis program can help the designer calculate stresses and moments in the piping system to adequately size any anchors and anchor blocks to ensure compliance with ASME *Standard* B31.1.

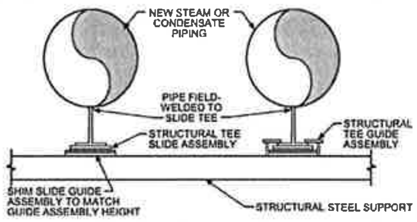

Fig. 12 Slide and Guide Detail

3.5 DISTRIBUTION SYSTEM CONSTRUCTION

The combination of aesthetics, first cost, safety, and life-cycle cost naturally divide distribution systems into two distinct categories: aboveground and underground. The materials needed to ensure long life and low heat loss further classify DHC systems into low-, medium-, and high-temperature systems. The temperature range for medium-temperature systems is usually too high for materials used in low-temperature systems; however, the same materials used in high-temperature systems are typically used for medium-temperature systems. Because low-temperature systems have a lower temperature differential between the working fluid temperatures and the environment, heat loss is inherently less. In addition, the options for efficient insulation materials and inexpensive pipe materials that resist corrosion are much greater for low-temperature systems.

The aboveground system has the lowest first cost and the lowest life-cycle cost because it can be maintained easily and constructed with readily available materials. Generally, aboveground systems are acceptable where they are hidden from view or can be hidden by landscaping. Poor aesthetics and the risk of vehicle damage to the aboveground system remove it from contention for many projects.

Although the aboveground system is sometimes partially factory prefabricated, more typically it is entirely field fabricated of components such as pipes, insulation, pipe supports, and insulation jackets or protective enclosures that are commercially available. Other common systems that are completely field fabricated include walk-through tunnel (see Figure 14), concrete surface trenches (see Figure 15), deep-burial small tunnels (see Figure 16), and underground systems that use poured insulation (see Figure 17) or rigid closed-cell insulation (e.g., cellular glass) (see Figure 18) to form an envelope around the carrier pipes.

Field-assembled systems must be designed in detail, and all materials must be specified by the project design engineer. Evaluation of the project site conditions indicates which type of system should be considered for the site. For instance, the shallow trench system is best where utilities that are buried deeper than the trench bottom need to be avoided and where the covers can serve as sidewalks. Direct-buried conduit, with a thicker steel casing coated in either epoxy or HDPE or wrapped in fiberglass-reinforced polymer/plastic (FRP), may be the only system that can be used in flooded sites where the conduit is in direct contact with groundwater. The conduit system is used where aesthetics is important. It is often used for short distances between buildings and the main distribution system, and where the owner is willing to accept higher life-cycle costs.

perature distribution indicates otherwise. Rigid extruded polystyrene insulation board may be used to insulate adjacent chilled-water lines from the effects of a buried heat distribution pipe; however, care must be taken not to exceed the temperature limits of the extruded polystyrene insulation; numerical analysis of the thermal problem may be required. Finally, at valve locations or when transitioning from ductile or plastic piping to steel at the buildings, flanged connections are usually best, but should be located inside the building or a small access port and not directly buried. Experience shows that buried flanges, or flanged connections such as to valves, are prone to leakage, are a weak link in the piping system, and should be avoided. Proper gasket selection and bolt torque are also critical to a leakfree system.

Table 12 summarizes some of the important aspects of the various piping materials, and Figure 13 shows approximate relative costs of the most popular materials. The major advantages and disadvantages of each of these materials, as well as applicable standards when used for the carrier pipe, are as follows:

Table 12 Relative Merits of Piping Materials Commonly Used for District Cooling Distribution Systems

Piping System	Carrier Pipe Joint Integrity	Joint Inspection	Insulated Joints Possible*	Corrosion Resistance	Installation Skill Level	Installation Time	Strength under Burial Conditions	Relative Installed Cost
Welded steel	Excellent	NDT (x-ray, etc.), pressure testing	Yes	Low, requires protection	High	High	Excellent	High
Soldered copper	Medium	Pressure testing	Yes	Good	Medium	Medium	Good	Small D = High
Ductile iron	Low	Pressure testing	No	Low, requires protection	Low/medium	Low	Very good	Low/medium
Cement pipe	Low	Pressure testing	No	Excellent	Low/medium	Low	Good	Low/medium
FRP	Low/medium	Pressure testing	Yes	Excellent	Medium	Low/medium	Low	Low/medium
PVC	Low	Pressure testing	No	Excellent	Low/medium	Low	Low	Low
HDPE	High	Pressure testing	Yes	Excellent	Medium	Small D = Low		
Large D = Medium	Excellent	Small D = Low						
Large D = Medium/high								

*Insulated joints are not recommended for piping systems that have allowable leakage rates for joints.

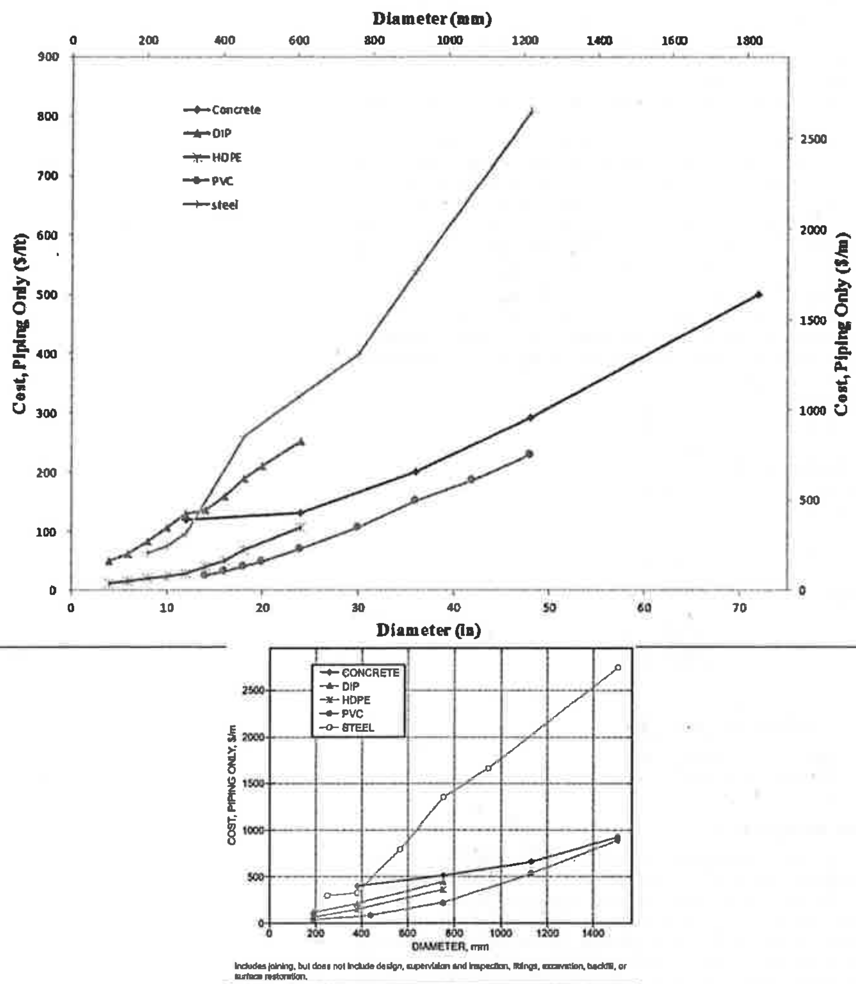

Fig. 13 Relative Costs for Piping Alone, Uninsulated

Cost data are from RSMeans-CostWorks® (www.rsmeansonline.com) for ~~third~~first-quarter ~~2012~~ 2018 water utilities.

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Commented [st1]: Source of data is from RSMeans but you have to search for the database to find the costs for materials. A specific table such as this doesn't exist. No permission required.

- Steel
  Advantages: High strength and good flexibility, can be joined by welding for a high-integrity joint that can be inspected for quality control, widely available in all sizes, familiar material to most workforces.
  Disadvantages: Relatively high cost, highly susceptible to corrosion and will require corrosion protection. Skilled labor force required for welding. Slower installation, especially in larger diameters.
  Standards: ASTM A53/A53M, ASTM A106A/106M.
- Copper

ent temperature. Underground systems cost almost 10 times as much to build, and require much more to operate and maintain. Heat distribution systems must be designed for zero leakage and must account for thermal expansion, degradation of material as a function of temperature, high- pressure and transient shock waves, heat loss restrictions, and accelerated corrosion. In the past, resolving one problem in underground systems often created a new, more serious problem that was not recognized until premature failure occurred. Segan and Chen (1984) describe the types of premature failures that may occur if this guidance is ignored.

Common types of underground systems are the walk-through tunnel, concrete surface trench, deep-buried small tunnel, poured insulation envelope, rigid closed-cell insulation, and conduit system.

Walk-Through Tunnel. This system (Figure 14) consists of a field-erected tunnel large enough for a person to walk through after the distribution pipes are in place. It is essentially an aboveground system enclosed with a tunnel. The tunnel is buried deep enough to cover the top with earth, and is large enough for routine maintenance and inspection to be done easily without excavation. The preferred construction material for the tunnel walls and top cover is reinforced concrete. Masonry units and metal preformed sections have been used to construct the tunnel and top with less success, because of groundwater leakage and metal corrosion. The distribution pipes are supported from the tunnel wall or floor with pipe supports that are commonly used on aboveground systems or in buildings. Some groundwater will penetrate the top and walls of the tunnel; therefore, a water drainage system must be provided. Usually, electric lights and electric service outlets are provided for ease of inspection and maintenance. This system has the highest first cost of all underground systems; however, it can have the lowest life-cycle cost because of its ease of maintenance, the ability to correct construction errors easily, and an extremely long life. If steam or HTHW piping is located in the tunnel, ambient temperatures may become extreme; such tunnels are typically either forced ventilated or gravity ventilated. Additional insulation may be required for chilled-water piping that shares the tunnel with steam or HTHW piping, because of the higher ambient temperatures. Selected pipe insulation material should be coupled with a high-performance vapor retarder and possibly a protective jacket.

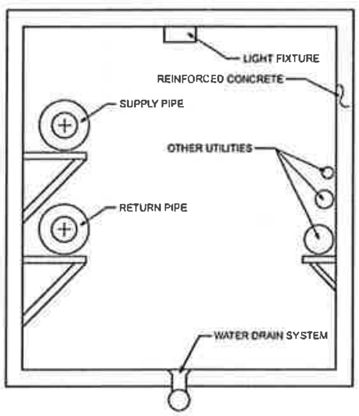

Fig. 14 Walk-Through Tunnel

Shallow Concrete Surface Trench. This system (Figure 15) is a partially buried system. The floor is usually about 3 ft {1 m} below the surface grade. It is only wide enough for the carrier pipes and the pipe insulation plus some additional width to allow for pipe movement and possibly enough room for a person to stand on the floor. The trench usually is about as wide as it is deep. The top is constructed of reinforced concrete covers that protrude slightly above the surface and may also serve as a sidewalk. The floor and walls are usually cast-in-place reinforced concrete and the top is either precast or cast-in-place concrete. Precast concrete floor and wall sections have not been successful because of the large number of oblique joints and nonstandard sections required to follow the surface topography and to slope the floor for drainage. This system is designed to handle stormwater and groundwater that enters the system, so the floor is always sloped toward a drainage point, which typically is a steam trap pit, valve vault, or access port where a sump pump or other positive-drainage method is provided.

Construction of this system is typically started in an excavated trench by pouring a cast-in-place concrete base that is sloped so intruding groundwater can drain to the valve vaults. The slope selected must also be compatible with the pipe slope requirements of the distribution system. The concrete base may have provisions for the supports for the distribution pipes, the groundwater drainage system, and the mating surface for the side walls. The side walls may have provisions for the pipe supports if the pipes are not bottom supported. If the upper portion is to have cast-in-place concrete walls, the bottom may have reinforcing steel for the walls protruding upward. The pipe supports, distribution pipes, and pipe insulation are all installed before the top cover is installed.

The groundwater drainage system may be a trough formed into the concrete bottom, a sanitary drainage pipe cast into the concrete bottom, or a sanitary pipe that is located slightly below the concrete base. The cover for the system is typically either of cast-in-place concrete or preformed sections such as precast concrete sections or half-round clay tile sections. The top covers must mate to the bottom and each other as tightly as possible to limit entry of groundwater. After the covers are installed, the system is covered with earth to match the existing topography.

Poured Insulation. This system (Figure 17) is buried with the distribution system pipes encased in an envelope of insulating material and the insulation envelope covered with a thick layer of earth as required to match existing topography. This system is used on sites where the groundwater is typically well below the piping system. Like other underground systems, experience indicates that it will be flooded because the soil will become saturated with water several times during the design life; therefore, the design must accommodate flooding.

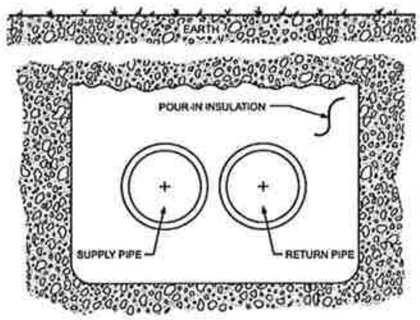

Fig. 17 Poured Insulation System

The insulation material serves several functions. It may support the distribution pipes as bedding and backfill, with additional support as recommended by the product manufacturer, and it must support earth loads. The insulation must prevent groundwater from entering the interior of the envelope, and it must have long-term resistance to physical breakdown caused by heat and water. The insulation envelope must allow the distribution pipes to expand and contract axially as the pipes change temperature. In elbows, expansion loops, and bends, the insulation must allow formed cavities for lateral movement of the pipes, or be able to migrate around the pipe without significant distortion of the insulation envelope while still retaining the required structural load-carrying capacity. Pay special attention to corrosion of metal parts and water penetration at anchors and structural supports that penetrate the insulation envelope.

Hot distribution pipes tend to drive moisture out of the insulation as steam; however, pipes used to distribute a cooling medium tend to condense water in the insulation, which reduces the insulation's thermal resistance. A groundwater drainage system may be required, depending on the insulation material selected and severity of the groundwater; however, if such drainage is needed, it is a strong indicator that this is not the proper system for the site conditions.

This system is constructed by excavating a trench with a bottom slope that matches the desired slope of the distribution piping. The width of the bottom of the trench is usually the same as the width of the insulation envelope because it serves as a form. The distribution piping is then assembled in the trench and supported at the anchors and by blocks that are removed as the insulation is poured in place. The form for the insulation can be the trench bottom and sides, wooden forms, or sheets of plastic, depending on the type of insulation used and the site conditions. The insulation envelope is covered with earth to complete the installation.

The project design engineer is responsible for finding an insulation material that fulfills all of the previously mentioned requirements. At present, no standards have been developed for insulation used in this type of application. **Hydrophobic powders**, which are a special type of pulverized rock treated to be water repellent, have been used successfully. The hydrophobic characteristic of this powder prevents water from dampening the powder and has some ability as a barrier for preventing water from entering the insulation envelope. This insulating powder typically has a much higher thermal conductivity than mineral wool or fiberglass pipe insulation; therefore, the thickness of the poured envelope must be significantly greater. In addition, Phetteplace et al. (1998) found installed heat losses to be much higher than would be expected using manufacturer's data for one poured insulation material, and that actual installed dimensions were less than manufacturer's recommendations in the majority of installations. This may have been the result of the design or construction errors, contractors purposely "shorting" on dimensions, or compaction during or after backfilling. The user is cautioned to verify dimensions during construction and also use appropriate in-place densities for the poured insulation material; do not use bulk or loose density. Measure installed density by a meaningful test for noncohesive soils (e.g., ASTM *Standard* D4253). Furthermore, thermal conductivity is a strong function of density and thus must also be measured at realistic in-place densities. As with any system, the designer should analyze the cost of the alternatives, including preinsulated (i.e., conduit) or field-insulated piping.

Each conduit section is shipped in lengths up to 40 ft {12.2 m}. Elbows, tees, loops, and bends are factory prefabricated to match the straight sections. The prefabricated components are assembled at the construction site; therefore, a construction contract is typically required for trenching, backfilling, connecting to buildings, connecting to distribution systems, constructing valve vaults, and performing some electrical work associated with sump pumps, power receptacles, and lights.

Much of the design work is done by the factory that manufactures the prefabricated sections; however, the field work must be designed and specified by the project design engineer or architect. Prefabricated components create a serious problem with accountability. For comparison, when systems are entirely field assembled, the design responsibility clearly belongs to the project design engineer, and system assembly and installation are clearly the responsibility of the construction contractor. When a condition arises where a conduit system cannot be built without modifying prefabricated components, or if the construction contractor does not follow the instructions from the prefabricator, a serious conflict of responsibility arises. The design engineer, as the engineer of record (EOR), assumes the responsibility of review and approval of any design work by others. For these reasons, it is imperative that the project design engineer or architect clearly delineate the responsibilities of the factory prefabricator.

Crushing loads have been used (erroneously) to size the casing thickness, assuming that corrosion was not a factor. However, corrosion rate is usually the controlling factor because the casing temperature can range from less than 100°F {40°C} to more than 300°F {150°C}, a range that encompasses the maximum corrosion rate of steel (Figure 20). As shown in the figure, the steel casing of a district heating pipe experiences corrosion rates several times that of domestic water pipes. The casing temperature varies with burial depth, soil conditions, carrier pipe temperature, and pipe insulation thickness. The casing must be strong and thick enough to withstand expansion and contraction forces and corrosion degradation.

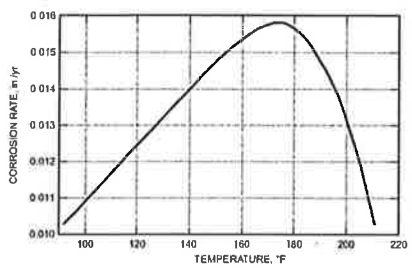

Fig. 20 Corrosion Rate in Aggressive Environment Similar to Mild Steel Casings in Soil

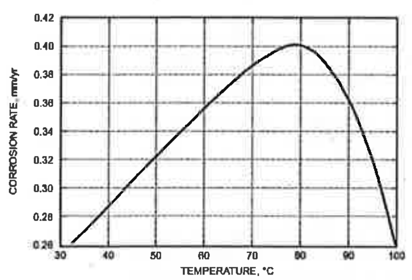

Fig. 20 Corrosion Rate in Aggressive Environment Similar to Mild Steel Casings in Soil

All insulation must be kept dry for it to maintain its thermal insulating properties; the exception is cellular glass in cold applications. Because underground systems may be flooded several times during their design life, even on sites that are thought to be dry, a reliable water intrusion removal system is necessary in the valve vaults. Two designs are used to ensure that the insulation performs satisfactorily for the life of the system. In the **air space system**, an annular air space between the pipe insulation and the casing allows the insulation to be dried out if water enters. In the **water spread limiting (WSL) system**, which has no air space, the conduit is designed to keep water from entering the insulation. If water enters one section, a WSL system slows or prevents its spread to adjacent sections of piping.

The air space conduit system (Figures 21 and 22) should have an insulation that can survive short-term flooding without damage. The conduit manufacturer usually runs a boiling test with the insulation installed in the typical factory casing. No U.S. standard has been approved for this boiling test; however, the U.S. government uses a Federal Agency Committee 96 h boiling test for conduit insulation [see Phetteplace et al. (2013b) for protocol]. The insulation must have demonstrated that it can be dried with air flowing through an annular air space, and it must retain nearly new thermal insulating properties when dried.

the waterproof insulation envelope in each individual prefabricated conduit section, using the casing and carrier pipe to form part of the envelope and a waterproof bulkhead to seal the casing to the carrier pipe. In another type of construction, a second pipe fits tightly over the carrier pipe and seals the insulation between the second pipe and casing to achieve a watertight insulation envelope.

Conduit Design Conditions. The following three design conditions must be addressed to have reasonable assurance that the system selected will have a satisfactory service life:

- **Maximum heat loss** occurs when the soil is wettest and the conduit is shallowly buried (minimum burial depth), usually with about 2 ft ([redacted]) of earth cover. This condition represents the highest gross heat transfer and is used to size the distribution piping and equipment in the central heating plant. For heating piping, because the casing is coldest during start-up, the relative movement of the carrier pipe with respect to the casing may be maximum during this condition.
- A **dry-soil condition** may occur when the conduit is buried deep. The soil plays a more significant role in the heat transfer than the pipe insulation because of the soil's thickness (and thus, insulating value). The highest temperature of the insulation, casing, and casing coating occur during this condition. Paradoxically, the minimum heat loss occurs during this condition because the soil acts as a good insulator. This condition is used to select temperature-sensitive materials and to design for casing expansion. The relative movement of the carrier pipe with respect to the casing may be at minimum during this condition; however, if the casing is not restrained, its movement with respect to the soil will be at maximum. If restrained, the casing axial stresses and axial forces will be highest and the casing allowable stresses will be lowest because of the high casing temperature. Buckling of the casing is possible in extreme situations.

Figure 24A shows the effect of burial depth on casing temperature as a function of soil thermal conductivity for a typical system. Figure 24B shows the effect of insulation thickness on casing temperature, again as a function of soil thermal conductivity. Analysis of Figure 24A and 24B suggests some design solutions that could lower the effects of the dry-soil condition: reducing carrier pipe temperature, using thicker carrier pipe insulation, providing a device to keep the soil wet, or minimizing burial depth. However, if these solutions are not feasible or cost effective, a different type of material or an alternative system should be considered.

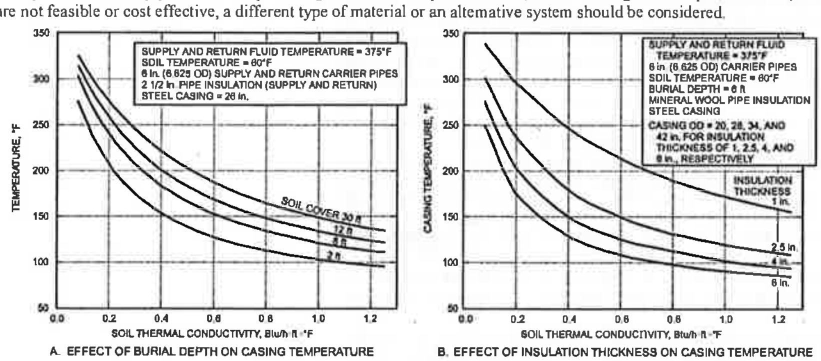

Fig. 24 Conduit Casing Temperature Versus Soil Thermal Conductivity

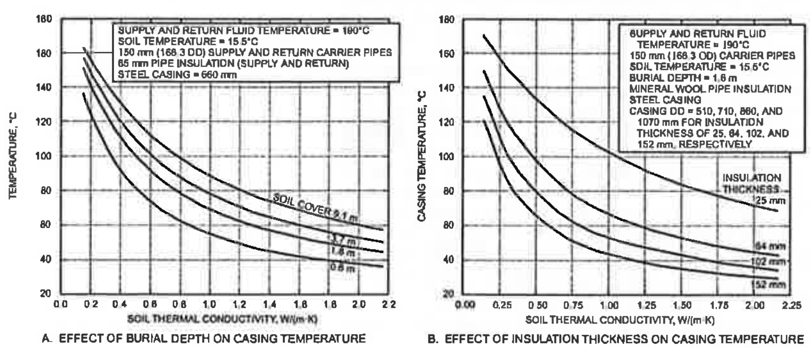

Fig. 24 Conduit Casing Temperature Versus Soil Thermal Conductivity

Although it is possible that the soil will never dry out, given the variability of climate in most areas, it is likely that a drought will occur during the life of the system. Only one very dry condition can cause permanent damage to the insulation or other system compoof the piping. Chilled-water systems should be pressure tested during the winter, and hot-water and steam systems tested during the summer.

A leak is difficult to locate without the aid of a cable type leak detector. Finding a leak typically involves excavating major sections between valve vaults. Infrared detectors and acoustic detectors can help narrow down the location of a leak, but they do not work equally well for all underground systems. Also, they are not as accurate with underground systems as with an aboveground system.

Chilled- and Hot-Water Systems. Chilled-water piping systems are usually insulated with urethane foam with a vaporproof jacket (HDPE, urethane, PVC, CPVC, etc.). Copper wires can be installed during fabrication to aid in detecting and locating liquid leaks. The wires may be insulated or uninsulated, depending on the manufacturer. Some systems monitor the entire wire length, whereas others only monitor at the joints of the piping system. The detectors either look for a short in the circuit using Ohm's law or monitor for impedance change using time domain reflectometry (TDR).

Steam, High-Temperature Hot-Water, and Other Conduit Systems. Air gap designs, which have a gap between the inner wall of the outer casing and the insulation, can have probes installed at the low points of drains or at various points to detect leaks. Leaks can also be detected with a continuous cable that monitors liquid leakage. The cable is installed at the bottom of the conduit with a minimum air gap required, typically 1 in. {[redacted]}. Pull points or access ports are installed every 400 to 500 ft {[redacted]} on straight runs, with changes in direction reducing the length between pull points. Systems monitor either by looking for a short on the cable using Ohm's law or by sensing the impedance on a coaxial cable using resistance temperature devices (RTDs). During installation, care must be taken to keep the system clean and dry to keep any contamination from the leak detection system that might cause it to fail. The system must be sealed airtight to prevent condensation from accumulating in the piping at the low points.

Geotechnical Considerations

Underground district heating or cooling distribution systems have more stringent burial requirements than most other building utilities. These piping systems normally have the coatings or jackets needed for corrosion protection, or insulation protection that must not be damaged by rocks, debris, or construction equipment. Thus, proper burial conditions must be established for the district heating or cooling distribution system to achieve its design life. Requirements vary, and manufacturers of the piping system should be able to provide guidance specific to their system. It is the EOR's responsibility to ensure that these requirements are included in the contract documents. Preferably, a licensed geotechnical engineer familiar with local conditions should conduct a site survey before construction, to recommend any soil testing required and to develop the specifications for excavation and backfill. The geotechnical and structural engineer should also be responsible for designing any thrust blocks or anchors that are needed, based on forces provided by the EOR.

In general, trenches must be overexcavated by at least 4 in. {[redacted]} in depth to remove any unyielding material; overexcavation may need to be greater at the locations of the field joints, depending on the type of system and construction method. The overexcavation is generally filled with a select backfill material; normally, this is a sandy, noncohesive material free of any stones greater than 0.75 in. {[redacted]}. Any unstable materials encountered in the excavation should be removed and properly backfilled and compacted. The selected backfill in the trench bottom should be prepared to achieve the minimum slope for the carrier pipe of 1 in. in 20 ft (0.2%) {[redacted]} and compacted to 95% of laboratory maximum density per ASTM *Standard* D698. Some of the methods of carrier pipe joining, such as welding, require a working area around the entire circumference of the field joint; one way to achieve this is to overexcavate under the pipe, and potentially even at the sides of the trench at the locations of the field joints. If this is done, be sure to fully compact the backfill material under the field joint area.

Another method to provide working clearance for making the field joint is to block the piping up off the bottom of the trench during that process. When this method is used, care must be taken to block the pipe sufficiently to achieve proper alignment for joining and to emulate the pipe as it will ultimately lie on the sloped trench bottom. Once the field joints have been completed, remove the blocking and carefully and uniformly place the piping in the prepared trench bottom. The blocking should not be left in place, because this creates point loading on the piping and may contribute to differential settlement as well. In some situations when welded-steel piping is used, for example, it may be possible to join two sections of piping together adjacent to the trench and then lift the assembly in as a unit and thus reduce the work required in the trench.

The elevation of the trench bottom must not have slope reversals between valve vaults and building entry locations. After the piping is placed in the trench and all field joints and pressure tests have been completed, immediately before backfilling, the elevation of the top invert of the pipe/jacket should be taken at each pipe section midpoint and field joint. These elevations should be recorded and subsequently transferred to the as-built drawings. Backfill of the piping should then be accomplished in layers of no more than 6 in. {[redacted]} with the same select backfill material used for the pipe bedding. The selected backfill should be extended to approximately 12 in. {[redacted]} above the top of the pipe or jacketing. Include buried utility warning tape in the trench at this depth. Compaction of this backfill material should also be to 95% of laboratory maximum density per ASTM *Standard* D698. Ensure that the backfill adequately fills the void created under the pipe and between the supply and return pipes. Also, take care not to damage any pipe coating or insulation jacketing material; if any such damage does occur, repair it according to the pipe system manufacturer's field repair instructions. Final backfill of the remainder of the trench should be accomplished using the native soil (removing any stones greater than 3 in. {[redacted]}), compacted in layers of no more than 6 in {[redacted]}. This final backfill should be compacted to 95% of laboratory maximum density per ASTM *Standard* D698 for noncohesive soils, or 90% of laboratory maximum density per ASTM *Standard* D698 for cohesive soils. Note that it is not advisable to complete the final backfilling with anything other than native soil, because the permeability of other substances may be much different than that of the native material. For example, using a permeable backfill material in an impermeable native soil is essentially placing the district heating or cooling system in a drainage ditch for surface water.

Also, note that horizontal boring, jacking, and microtunneling have become popular methods of installing buried pipelines where the normal cut-and-cover methods described previously may be difficult or impossible, or simply cost prohibitive. These alternative burial methods preclude the use of protective backfilling, but often may not need such steps. However, metallic pipelines must still be protected from corrosion, and methods appropriate for the installation method must be used.

cations allow with respect to any item placed in the vault. To achieve desired results, the vault layout must be shown to scale on the contract drawings.

High Humidity. High humidity develops in a valve vault when it has no positive ventilation. Gravity ventilation is often provided, in which cool air enters the valve vault and sinks to the bottom. At the bottom of the vault, the air warms, becomes lighter, and rises to the top of the vault, where it exits. In the past, some designers used a closed-top valve vault with an exterior ventilation pipe with an elbow that directs the exiting air down. However, the elbowed-down vent hood tends to trap the exiting air and prevent gravity ventilation from working. Open structural grate tops are the most successful covers for ventilation purposes. Open grates allow rain to enter the vault; however, the techniques mentioned in the section on Ponding Water are sufficient to handle the rainwater. Open grates with sump basins have worked well in extremely cold climates and in warm climates. Some vaults have a closed top and screened, elevated sides to allow free ventilation. In this design, the solid vault sides extend slightly above grade; then, a screened window is placed in the wall on at least two sides. The overall above-grade height may be only 18 in ([redacted]).

High Temperatures. The temperature in the valve vault rises when no systematic way is provided to remove heat losses from the distribution system. The gravity ventilation rate is usually not sufficient to transport heat from the closed vaults. Part of this heat transfers to the earth; however, an equilibrium temperature is reached that may be higher than desired. Ventilation techniques discussed in the section on High Humidity can resolve the problem of high temperature if the heat loss from the distribution system is near normal. Typical problems that greatly increase the amount of heat released include

- Leaks from a carrier pipe, gaskets, packings, or appurtenances
- Insulation that has deteriorated because of flooding or abuse
- Standing water in a vault that touches the distribution pipe
- Steam vented to the vault from partial flooding between valve vaults
- Vents from flash tanks
- Insulation removed during routine maintenance and not replaced

To prevent heat release in a new system, a workable ventilation system must be designed. On existing valve vaults, the valve vault must be ventilated properly, all leaks corrected, and all insulation that was damaged or left off replaced. Commercially available insulation jackets that can easily be removed and reinstalled from fittings and valves should be installed. If flooding occurs between valve vaults, portions of the distribution system may have to be excavated and repaired or replaced. Vents from vault appurtenances that exhaust steam into the vault may have to be routed aboveground if the ventilation technique is insufficient to handle the quantity of steam exhausted.

Deep Burial. When a valve vault is buried too deeply, (1) the structure is exposed to groundwater pressures, (2) entry and exit often become a safety problem, (3) construction becomes more difficult, and (4) the cost of the vault is greatly increased. Ideally, valve vault spacing should be less than 500 ft ([redacted]) (NAS 1975). If greater spacing is desired, use an accurate life-cycle analysis to determine spacing. The most common way to limit burial depth is to place the valve vaults closer together. Sawtooth-shaped steps in the distribution system slope are accommodated in the valve vault (i.e., the carrier pipes enter the valve vault at a low elevation and leave at a higher elevation). If the slope of the distribution system is changed to more nearly match the earth topography, the valve vaults will be shallower; however, the allowable range of slope of the carrier pipes restricts this method. In most systems, the slope of the distribution system can be reversed in a valve vault, but not out in the system between valve vaults. The minimum slope for the carrier pipes is 1 in. in 20 ft ([redacted]). Lower slopes are outside the range of normal construction tolerance. If the entire distribution system is buried too deeply, the designer must determine the maximum allowable burial depth of the system and survey the topography of the distribution system to determine where the maximum and minimum depth of burial will occur. All elevations must be adjusted to limit the minimum and maximum allowable burial depths.

Freezing Conditions. Failure of distribution systems caused by water freezing in components is common. The designer must consider the coldest temperature that may occur at a site and not the 99% or 99.6% condition used in building design (as discussed in Chapter 27 of the 2013 *ASHRAE Handbook—Fundamentals*). Drain legs or vent legs that allow water to stagnate are usually the cause of failure. Insulation should be on all items that can freeze, and it must be kept in good condition. Electrical heat tape and pipe-type heat tracing can be used under insulation. If part of a chilled-water system is in a ventilated valve vault, the chilled water may have to be circulated or be drained if not used in winter.

Safety and Access. Some working fluids used in underground distribution systems can cause severe injury and death if accidentally released in a confined space such as a valve vault. The shallow valve vault with large openings is desirable because it allows personnel to escape quickly in an emergency. The layout of the pipes and appurtenances must allow easy access for maintenance without requiring maintenance personnel to crawl underneath or between other pipes. The goal of the designer is to keep clear work spaces for maintenance personnel so that they can work efficiently and, if necessary, exit quickly. Engineering drawings must show pipe insulation thickness; otherwise they will give a false impression of the available space.

The location and type of ladder is important for safety and ease of egress. It is best to lay out the ladder and access openings when laying out the valve vault pipes and appurtenances as a method of exercising control over safety and ease of access. Ladder steps, when cast in the concrete vault walls, may corrode if not constructed of the correct material. Corrosion is most common in steel rungs. Either cast-iron or prefabricated, OSHA-approved, galvanized steel ladders that sit on the valve vault floor and are anchored near the top to hold it into position are best. If the design uses lockable access doors, the locks must be operable from inside or have some keyed-open device that allows workers to keep the key while working in the valve vault. Extremely deep vaults (greater than 20 ft ([redacted])) may require an intermediate landing and caged ladder with fall protection to satisfy OSHA requirements.

Vault Construction. The most successful valve vaults are those constructed of cast-in-place reinforced concrete. These vaults conform to the earth excavation profile and show little movement when backfilled properly. Leakproof connections can be made with mating tunnels and conduit casings, even though they may enter or leave at oblique angles. In contrast, prefabricated valve vaults may settle and

4.1 DIRECT CONNECTIONS

Because a direct connection offers no barrier between the district water and the building's own system (e.g., air-handling unit cooling and heating coils, fan-coils, radiators, unit heaters, process loads), the water circulated at the district plant has the same quality as the customer's water (and vice versa). Direct connections, therefore, are at a greater risk of incurring damage or contamination based on the poor water quality of either party. Typically, district systems have contracts with water treatment vendors and monitor water quality continuously. This may not be the case with all consumers. A direct connection is often more economical than an indirect connection because the consumer is not burdened by the installation of heat exchangers, additional circulation pumps, or water treatment systems; therefore, investment costs are reduced and return temperatures identical to design values are possible.

In general, consider using a direct connection under the following conditions: the building owner is the district system owner or they are related entities; control of first cost is important; buildings are generally low rise; building systems are new or in good condition; in-building space for interconnection is limited; and the building owner, if different than the district utility, respects the need for high Δt and will maintain the building systems accordingly and retrofit the building equipment where necessary to achieve adequate Δt.

Figures 25 and 26 show a simple chilled-water direct connection using building circulation pumps and using the district cooling provider's pumps, respectively. Figure 26's method is preferred, because it is the simplest, with no control valve, but must have high return water temperatures at varying flows. Consequently, this method requires the building design engineer and controls contractor to implement a design that operates per the design intent. See the Temperature Differential Control section for further discussion regarding achieving high system Δt.

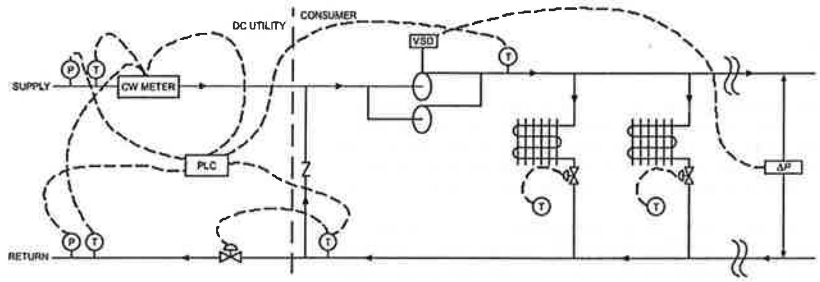

Fig. 25 Direct Connection of Building System to District Chilled Water with Building Pumps
(Phetteplace et al. 2013a)

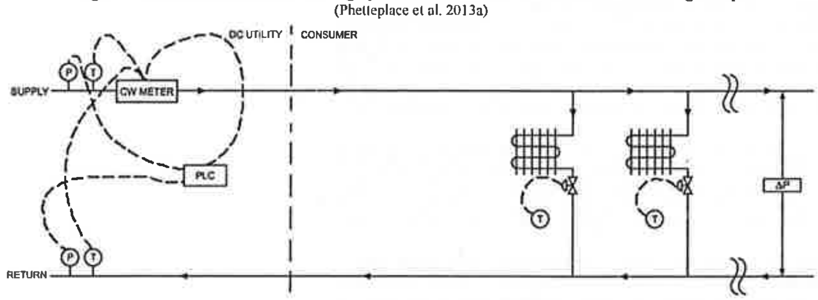

Fig. 26 Direct Connection of Building System to District Chilled Water Without Building Pumps
(Phetteplace et al. 2013a)

Similarly, Figure 27 shows the simplest form of hot-water direct connection, where the district heating plant pumps water through the consumer building. This figure includes a pressure differential regulator (which may be required to reduce system differential pressure to meet any lower building system parameters), a thermostatic control valve on each terminal unit, and a pressure relief valve. Most commercial systems have a flowmeter installed as well as temperature sensors and transmitters to calculate the energy used. Pressure transmitters may be installed as input for plant circulating pump speed control. The location of each device may vary from system to system, but all of the major components are indicated. The control valve is the capacity regulating device that restricts flow to maintain either a water supply or return temperature on the consumer's side.

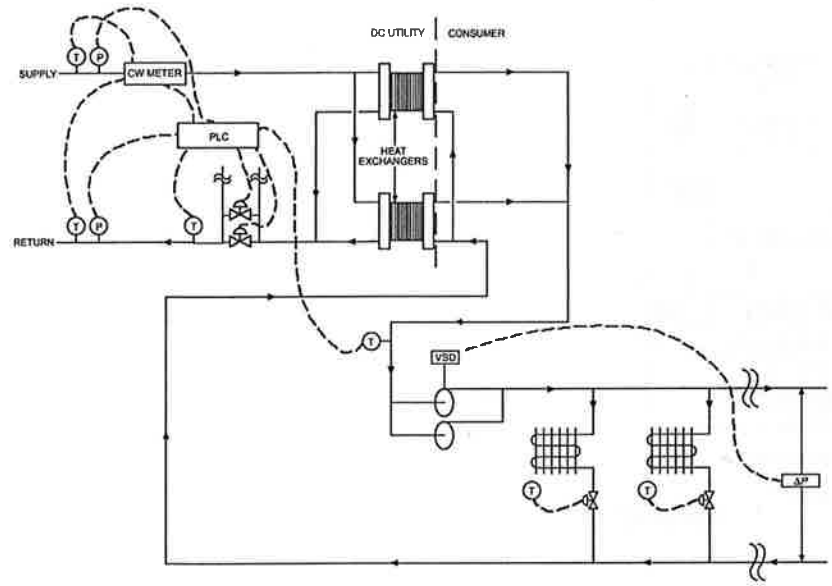

Fig. 28 Indirect Connection of Building System to District Chilled Water
(Phetteplace et al. 2013a)

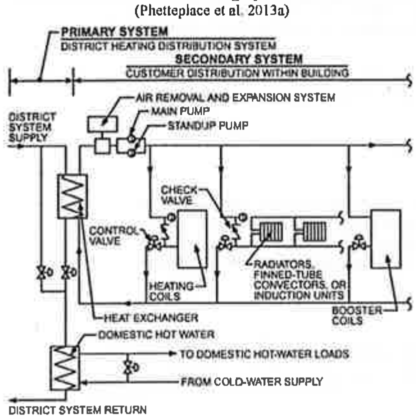

Fig. 29 Basic Cascading Indirect Heating-System Schematic

4.3 STEAM CONNECTIONS

Although higher pressures and temperatures are sometimes used, most district heating systems supply saturated steam at pressures between 5 and 250 psig ([redacted]) to customers' facilities. The steam is pretreated to maintain a neutral pH, and the condensate is cooled and discharged to the building sewage system (not preferred) or returned back to the central plant for recycling (preferred). Many consumers run the condensate through a heat exchanger to heat the domestic hot-water supply of the building before returning it to the central plant or, sometimes, to the building drains if the district system does not have a condensate return pipe. This energy-saving process extracts the maximum amount of energy out of the delivered steam. Again, it is best to return the condensate back to the plant

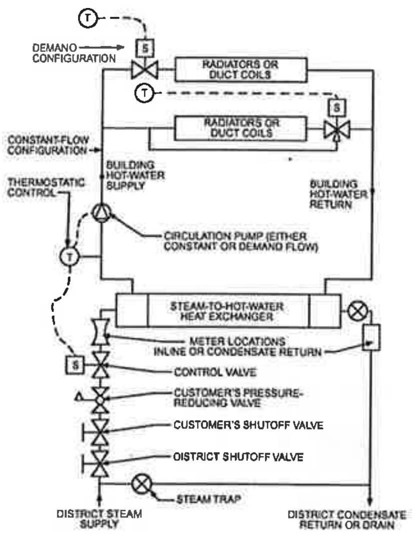

Fig. 31 District/Building Interconnection with Heat Exchange Steam System

Other components of the steam connection may include condensate pumps, flowmeters (steam and/or condensate), and condensate conductivity probes, which may dump condensate if contaminated by unacceptable debris. Often, energy meters are installed on both the steam and condensate pipes to allow the district energy supplier to determine how much energy is used directly and how much energy (condensate) is not returned back to the plant. Using customer energy meters for both steam and condensate is desirable for the following reasons:

- Offers redundant metering (if the condensate meter fails, the steam meter can detect flow or vice versa)
- Bills customer accordingly for makeup water and chemical treatment on all condensate that is not returned or is contaminated
- Meter is in place if customer requires direct use of steam in the future
- Assists in identifying steam and condensate leaks
- Improves customer relations (may ease customer's fears of overbilling because of a faulty meter)
- Provides a more accurate reading for peak demand measurements and charges

Monitor each level of steam pressure reduction as well as the temperature of the condensate. Where conductivity probes are used to monitor the quality of water returned to the steam plant, adequate drainage and cold-water quenching equipment may be required to satisfy local plumbing code requirements (temperature of fluid discharging into a sewer). The probe status should also be monitored at the control panel, to communicate high-conductivity alarms to the plant and, when condensate is being dumped, to notify the plant that a conductivity problem exists at a customer.

Building Conversion to District Heating

Table 14 (Sleiman et al. 1990) summarizes the suitability or success rate of converting various heating systems to be served by a district hot-water system. As shown, the probability is high for water-based systems, lower for steam, and lowest for fuel oil or electric systems. Low-suitability systems usually require expensive replacement of the entire heating terminal and generating units with suitable water-based equipment, including piping, pumps, controls, and heat transfer media.

Table 14 Conversion Suitability of Heating System by Type

Type of System	Low	Medium	High
Steam equipment			
One-pipe cast iron radiation	X		
Two-pipe cast iron radiation		X	
Finned-tube radiation		X	
Air-handling unit coils		X	
Terminal unit coils	X		

Hot-water equipment

Because PHEs require turbulent flow for proper heat transfer, pressure drops may be higher than those for comparable shell-and-tube models. High efficiency leads to a smaller package. The designer should consider specifying that the frame be sized to hold 20% additional plates. PHEs require very little maintenance because the high velocity of the fluid in the channels keep surfaces clean from fouling. However, larger particles may become lodged in fine cavities between the plates and choke flow; automatic back-flushing valves may be used to address this issue. PHEs generally have a cost advantage and require one-third to one-half the surface required by shell-and-tube units for the same operating conditions. PHEs are also capable of closer approach temperatures.

PHEs are typically used for district heating and cooling with water and for cooling tower water heat recovery (free cooling). Double-wall plates are also available for potable-water heating, chemical processes, and oil quenching. PHEs have three to five times greater heat transfer coefficients than shell-and-tube units and can achieve 1°F {[redacted]} approach, but for economic reasons the approach is traditionally 2°F {[redacted]}. Gasketed PHEs can be disassembled in the field to clean the plates and replace the gaskets. Typical applications go up to 365°F and 400 psig {[redacted]}. Plates are typically made from stainless steel, but are available in titanium for more corrosive uses such as seawater cooling.

Brazed PHEs are suitable for steam, vapor, or water solutions. They feature a close approach temperature (within 2°F {[redacted]}), large temperature drop, compact size, and a high heat transfer coefficient. Construction materials are stainless steel plates and sometimes frames brazed together with copper or nickel. Tightening bolts are not required as with the gasketed design. These units cannot be disassembled and cleaned; therefore, adequate strainers must be installed ahead of an exchanger and it must be periodically flushed clean in a normal maintenance program. Brazed PHEs usually peak at a capacity of under 200,000 Btu/h {[redacted]} (about 200 plates and 600 gpm {[redacted]}) and are suitable for 435 psig and 435°F {[redacted]}. Typical applications are district heating using hot water and refrigeration process loads. Double-wall plates are also available for domestic hot-water use. Avoid applications where the PHE may be exposed to large, sudden, or frequent changes in temperature and load, because of risk of thermal fatigue.

Welded PHEs can be used in any application for which shell-and-tube units are used that are outside the accepted range of gasketed PHE units, in liquid-to-liquid, steam-to-liquid, gas-to-liquid, gas-to-gas, and refrigerant applications. Construction is very similar to gasketed units except gaskets are replaced with laser welds. Materials are typically stainless steel, but titanium, monel, nickel, and various alloys are available. Models have design ratings that range from 500°F at 150 psig to 1000°F at 975 psig {[redacted]}; however, they are available only in small sizes. Normally, these units are used in ammonia refrigeration and aggressive process fluids. They are more suitable to pressure pulsation or thermal cycling because they are thermal fatigue resistant. A semiwelded PHE is a hybrid of the gasketed and the all-welded units in which the plates are alternatively sealed with gaskets and welds.

Shell-and-Coil Heat Exchangers. These European-designed heat exchangers are suitable for steam-to-water and water-to-water applications and feature an all-welded-and-brazed construction. This counter/cross-flow heat exchanger consists of a hermetically sealed (no gaskets), carbon-steel pressure vessel with hemispherical heads. Copper or stainless helical tubes within are installed in a vertical configuration. This type of heat exchanger offers a high temperature drop and close approach temperature. Its vertical arrangement requires less floor space than other designs and has better heat transfer characteristics than shell-and-tube units.

Shell-and-Tube Heat Exchangers. These exchangers are usually a multiple-pass design. The shell is usually constructed from steel and the tubes are often of U-bend construction, usually 3/4 in. {[redacted]} (nominal) OD copper, but other materials are available. These units are ASME U-1 stamped for pressure vessels.

Heat Exchanger Load Characteristics. To provide high Δt under multiple load conditions, variable flow is required on both sides of the heat exchanger (Perdue and Ansbro 1999; Skagestad and Mildenstein 2002; Tredinnick 2007). Without variable flow on the customer side, more water flow is required on the district side during reduced load. This condition results in both increased pumping for the district energy provider as well as reduced Δt. In addition, the customer side also experiences increased pumping costs without the use of variable flow. The specific degradation in Δt and the increases in flow depend on the actual heat exchanger selection, and can easily be determined for a specific heat exchanger by selection and sizing software available from the heat exchanger manufacturer.

An example provided by Skagestad and Mildenstein (2002) for a 427 ton {[redacted]} design load indicates that, at 50% load and constant flow on the consumer side, 75% of the design flow would be required on the district cooling system side, compared to 45% if the consumer side used variable flow. However, consumer-side constant flow reduces the Δt from the design value of 15°F {[redacted]} to just 10°F {[redacted]} at 50% load; when variable flow is used on the consumer's side of the PHE, the Δt actually increased from the design value of 15°F to 16.7°F {[redacted]}.

Another example of the need for variable-flow pumping on the consumer's side of PHE is provided by Tredinnick (2007) for a 500 ton {[redacted]} application. In Figure 32, the consumer side of the heat exchanger has constant flow with the consumer-side design supply temperature of 42°F {[redacted]}. The PHE has been sized such that, at 100% of design load, the district cooling return temperature is 54°F {[redacted]}; thus, at maximum load the Δt is 14°F {[redacted]}, assuming a 2°F {[redacted]} approach. However, with constant flow on the consumer side at 50% of design load, over 83% of the peak design flow on the district cooling side is required and the district cooling return temperature has decreased to 49.6°F {[redacted]}, thus lowering the Δt on the district cooling side to 9.6°F {[redacted]}.

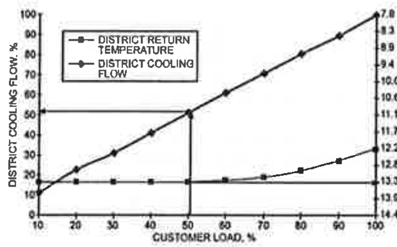

Fig. 33 Plate Heat-Exchanger Performance with Variable Flow on Customer Side and Customer-Side Supply Temperature of 5.6°C

(Tredinnick 2007)

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Variable flow also saves electrical pump energy and aids in controlling comfort. These examples, as well as others [e.g., Perdue and Ansbro (1999)] should make clear the need for variable flow on the consumer side of a PHE in an indirect connection of district cooling.

Typical constant-flow systems are found in older buildings and may be converted to simulate a variable-flow system by blocking off the bypass line around the air handler heat exchanger coil three-way control valve. At low operating pressures, this potentially may convert a three-way bypass-type valve to a two-way modulating shutoff valve. Carefully analyze the valve actuator, because the shutoff requirements and control characteristics are totally different for a two-way valve than for a three-way valve. For more information on building conversion, see Skagestad and Mildenstein (2002).

In theory, a cooling coil should have higher return water temperature when partially loaded than at full load, because the coil is oversized for the duty and thus has closer approach temperatures. In many real systems, as the load increases, the return water temperature tends to rise, and under low loads, the supply water temperature rises. Consequently, process or critical humidity control systems may suffer when connected to a system where return water temperature control is used to achieve high temperature differentials. Other techniques, such as separately pumping each CHW coil, may be used where constant supply water temperatures are necessary year round.

Flow Control Devices

In commercial systems, after the flowmeter, control valves are the most important element in the interface with the district energy system because proper valve adjustment and calibration save energy. High-quality, industrial-grade control valves provide more precise control, longer service life, and minimum maintenance.

All control valve actuators should take longer than 60 s to close from full open to mitigate pressure transients or water hammer, which occurs when valves slam closed. Actuators should also be sized to close against the anticipated system pressure so the valve seats are not forced open, thus forcing water to bypass and degrading temperature differential.

The wide range of flows and pressures expected makes selection of control valves difficult. Typically, only one control valve is required; however, for optimal response to load fluctuations and to prevent cavitation, two valves in parallel are often needed. The two valves operate in sequence and for a portion of the load (i.e., one valve is sized for two-thirds of peak flow and the other sized for one-third of peak flow). The designer should review the occurrence of these loads to size the proportions correctly. The possibility of overstating customer loads complicates the selection process, so accurate load information is important. It is also important that the valve selected operates under the extreme pressure and flow ranges foreseen. Because most commercial-grade valves will not perform well for this installation, industrial-quality valves are typically specified.

Electronic control valves should remain in a fixed position when a power failure occurs and should be manually operable. Pneumatic control valves should close upon loss of air pressure. A manual override on the control valves allows the operator to control flow. All chilled-water control valves must fail in the closed position. Then, when any secondary in-building systems are deenergized, the valves close and will not bypass chilled water to the return system. All steam pressure-reducing valves should close as well.

Oversizing results in reduced valve and actuator life span and causes hunting. Select control valves having a wide range of control; low leakage; and proportional-plus-integral control for close adjustment, balancing, temperature accuracy, and response time. Control valves should have actuators with enough force to open and close under the maximum pressure differential in the system. The control valve should have a pressure drop through the valve equal to at least 10 to 30% of the static pressure drop of the distribution system. This pressure drop gives the control valve the "authority" it requires to properly control flow. The relationship between valve travel and capacity output should be linear, with an equal percentage characteristic.

In hot-water systems, control valves are normally installed in the return line because the lower temperature in the line reduces the risk of cavitation and increases valve life. In chilled-water systems, control valves can be installed in either location; typically, however, they are installed the return line to reduce the potential for condensation on exposed external surfaces and to minimize water turbulence upstream of the flowmeter.

Instrumentation

In many systems, where energy to the consumer is measured for billing purposes, temperature sensors assist in calculating the energy consumed as well as in diagnosing performance. Sensors and their transmitters should have an accuracy range commensurate to the accuracy of the flowmeter. In addition, pressure sensors are required for variable-speed pump control (water systems) or valve control for

IDHA (1969) and Stultz and Kitto (1992) have more information on steam metering. For steam, as with hot- and chilled-water system metering, electronic and computer technology provide direct, integrating, and remote input to central control/measurement energy management systems.

Hot- and chilled-water systems are metered by measuring the temperature differential between the supply and return lines and the flow rate of the energy transfer medium. Thermal (Btu or kWh_{th}) meters compensate for the actual volume and heat content characteristics of the energy transfer medium. Thermal transducers, resistance thermometer elements, or liquid expansion capillaries are usually used to measure the differential temperature of the energy transfer medium in supply and return lines.

Water flow can be measured with a variety of meters, usually pressure differential, turbine or propeller, or displacement meters. Chapter 36 of the 2013 *ASHRAE Handbook—Fundamentals*, the *District Heating Handbook* (IDHA 1983), and Pomroy (1994) have more information on measurement. Ultrasonic meters are sometimes used to check performance of installed meters. Various flowmeters are available for district energy billing purposes. Critical characteristics for proper installation include clearances and spatial limitations as well as the attributes presented in Table 16. The data in the table only provide general guidance; meter manufacturers should be contacted for data specific to their products.

Table 16 Flowmeter Characteristics

Meter Type	Accuracy	Range of Control	Pressure Loss	Straight Piping Requirements (Length in Pipe Diameters)
Orifice plate	±1% to 5% full scale	3:1 to 5:1	High (>5 psi [redacted])	10*D* to 40*D* upstream; 2*D* to 6*D* downstream
Electromagnetic	±0.15% to 1% rate	30:1 to 100:1	Low (<3 psi [redacted])	5*D* to 10*D* upstream; 3*D* downstream
Vortex	±0.5% to 1.25% rate	10:1 to 25:1	Medium (3 to 5 psi [redacted])	10*D* to 40*D* upstream; 2*D* to 6*D* downstream
Turbine	±0.15% to 0.5% rate	10:1 to 50:1	Medium (3 to 5 psi [redacted])	10*D* to 40*D* upstream; 2*D* to 6*D* downstream
Ultrasonic	±1% to 5% rate	>10:1 to 100:1	Low (<3 psi [redacted])	10*D* to 40*D* upstream; 2*D* to 6*D* downstream

The meter should be located upstream of the heat exchanger and the control valve(s) should be downstream from the heat exchanger. This orientation minimizes the possible formation of bubbles in the flow stream and provide a more accurate flow indication. The transmitter should be calibrated for zero and span as recommended by the manufacturer.

Wherever possible, the type and size of meters selected should be standardized to reduce the number of stored spare parts, technician training, etc.

Displacement meters are more accurate than propeller meters, but they are also larger. They can handle flow ranges from less than 2% up to 100% of the maximum rated flow with claimed ±1% accuracy. Turbine-type meters require the smallest physical space for a given maximum flow. However, like many meters, they require at least 10 diameters of straight pipe upstream and downstream of the meter to achieve their claimed accuracy.

The United States has no performance standards for thermal meters. ASHRAE *Standard* 125 describes a test method for rating liquid thermal meters. Several European countries have developed performance standards and/or test methods for thermal meters, and CEN *Standard* EN 1434, developed by the European Community, is a performance and testing standard for heat meters.

District energy plant meters intended for billing or revenue require means for verifying performance periodically. Major meter manufacturers, some laboratories, and some district energy companies maintain facilities for this purpose. In the absence of a single performance standard, meters are typically tested in accordance with their respective manufacturers' recommendations. Primary measurement elements used in these laboratories frequently obtain calibration traceability to the National Institute of Standards and Technology (NIST).

For district energy cogeneration systems that send out and/or accept electric power to or from a utility grid, demand and usage meters must meet the existing utility requirements. For district energy systems that send out electric power directly to customers, the electric demand and usage meters must comply with local and state regulations. American National Standards Institute (ANSI) standards are established for all customary electric meters.

4.5 TEMPERATURE DIFFERENTIAL CONTROL

Maintaining a high water system temperature differential Δt between supply and return lines is most cost effective because it allows smaller pipes to be used in the primary distribution system. These savings must be weighed against higher building conversion costs that may result from the need for a low primary return temperature.

For district heating and cooling providers, hydronic system efficiency is usually measured in terms of the temperature differential. Proper control of the district heating and cooling temperature differential is not dictated at the plant but at the consumer. If the consumer's system is not compatible with the temperature parameters of the DHC system, operating efficiency suffers unless components in the consumer's system are modified. Low system Δt requires additional equipment to be energized, thus using more energy to satisfy the flow needs than the actual load demand requires. Therefore, the customer's Δt must be monitored and controlled.

To optimize the Δt, meet the customer's chilled-water demand, and save pump energy, flow from the plant should vary. Chilled-water flow in the customer's side must be varied as well. Terminal units in the building connected to the chilled-water loop (e.g., air-handling units, fan-coils) may require modifications (change three-way valves to two-way, etc.) to operate with variable water flow to ensure a maximum return water temperature.

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INDEX AND KEYWORDS

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