Projekt 9

Beräkningsmetoder för årsvärmefaktor för värmepumpsystem för jämförelse, systemval och dimensionering

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Sammanfattning

Föreliggande projekt har genomförts gemensamt av KTH-Energiteknik och SP-Energiteknik. Formell projektledare har Prof. Per G. Lundqvist på KTH varit, biträdd av Dr. Monica Axell på SP-Energiteknik. Forskarutförande har i huvudsak varit Dr. Joachim Claesson, KTH-Energiteknik, och Dr. Roger Nordman, SP-Energiteknik. Projektet har dels innehållit gemensamma delar, dels specifika för respektive forskarutförare.

Projektet har syftat till att identifiera de behov för vidareutveckling och homogenisering av beräkningsalgoritmer för att bestämma värmepumpsbaserade uppvärmningssystems effektivitet per årsbasis (SPF = Seasonal Performance Factor) i småhus och flerfamiljshus.

Del I:

Inledande arbete inom IEA Annex angående rättvisande årsvärmefaktorer pågår löpande och annexet är under uppstart. Detta arbete har precis startat och kommer att fortlöpa ett par år framåt.

Del II:

Fältmätningar är viktig del för att se om installerade system presterar enligt leverantörerna utfästelser. En sammanställning av tidigare genomförda fältmätningar har gjorts, vilket visar att ett omfattande antal fältmätningar finns rapporterade i litteraturen. Vidare har en genomgång och sammanställning av standardiserade metoder att beräkna SPF baserat på mätningar av värmepumpenheten utförts, vilket sedan jämförts mot mätningar. Även metoder för fältmätningar har identifierats och redogörs för.

Del III:

Enkät har genomförts vilket visar att alla program inte kan hantera mer komplexa uppvärmningssystem som t.ex. solfångare tillsammans med värmepump. Det är tydligt från enkäten att inte alla programverktyg kan hantera olika kombinationssystem, inte heller finns alla värmepumpsystem i alla program. En jämförelse ur användarsynpunkt av tre program har genomförts som också pekar på skillnader på indata och möjligheter att använda olika systemelösningar.

Del IV:

Ytterligare en enkät genomfördes som direkt jämförde resultaten mellan olika dimensioneringsprogram. Skillnaden på årsvärmefaktorn (SPF) mellan de givna fiktiva fastigheterna är inte speciellt besvärande, med beaktande att en leverantör räknade avfrostning dubbelt. Dock pekar det på vikten av riktig och rak kommunikation kring den information som utbytes mellan kund och säljare. Vidare har en metodik för beräkning av byggnaders energiprestanda tagits fram med hänsyn tagen till de internationella och svenska standarder som finns. Strukturen hänvisar i stor utsträckning till standarder där så tillämpligt och i vissa fall hänvisas till det arbete som genomfördes av värmepumpbranschen i samband med effsys1 och programvaran PRESTIGE.

Till sist beskrivs de nu gällande rekommendationer från Boverket gällande miljöbedömning av byggnaders energianvändning, där i huvudsak den kontrakterade energins belastning enligt uppgift av leverantör rekommenderas användas. Detta är en skiljer sig från tidigare då framförallt el räknades som marginal-el, vilket nu längre inte är fallet.

Summary

The present project has been a joint effort of KTH-Energy Technology and SP-Energy Technology. The project responsible has been Prof. Per G. Lundqvist (KTH), assisted by Dr. Monica Axell (SP). The main researchers have been Dr. Joachim Claesson (KTH) and Dr. Roger Nordman (SP). The project consisted of common parts, as well several parts at only one of KTH or SP.

The aim of the project has been to identify the need for improvement, development, and homogenization of Seasonal Performance Factors (SPF) of heat pump based heating systems in single and multifamily dwellings.

Part I:
An IEA Annex concerning the same topic as the present project have been initiated and recently launched, in which the project have participated actively. This work is just emerging, and will continue for some years.

Part II:
Field measurement is an important aspect of heat pump installations as they serve to whether the installation comply with the contracted performance. In the present project a summary of previously conducted field measurement of heat pump installation is presented. A large number of studies are available; however only a few have the necessary sufficient detailed level. The available standards for calculating the SPF based on heat pump measurements has been summarized. These methods have also been compared to field measurement.

Part III:
A survey have been sent out to the participant in the project investigating the limitations and possibilities of the different calculation software's. It is apparent that slightly more complex systems is not able to calculated with these software´s.

Part IV:
A second survey was sent out in which the SPF of four buildings with four heat pumps were to be evaluated in the software´s. The resulted SPF between the software´s is not very large, accounting for typically 200 euros/year. This amount seems not large but could make an impact of the economy of the installation.

A methodology for calculating SPF based on European standards has been developed.

Deltagande Parter

Följande företag och parter har varit delaktiga i projektet:

Climacheck	Klas Berglöf
ETM Kylteknik	Kenneth Weber
EVI Heat	Tommy Walfridson
IVT	Jim Fredin
KTH - Energiteknik	Joachim Claesson
NIBE	Ted Holmberg
SP - Energiteknik	Roger Nordman
SVEP	Martin Forsén
Thermia	Fredrik Karlsson

Innehåll

Introduktion

Föreliggande projekt har genomförts gemensamt av KTH-Energiteknik och SP-Energiteknik. Formell projektledare har Prof. Per G. Lundqvist på KTH varit, biträdd av Monica Axell på SP-Energiteknik. Forskarutförande har i huvudsak varit Dr. Joachim Claesson, KTH-Energiteknik, och Dr. Roger Nordman, SP-Energiteknik. Projektet har dels innehållit gemensamma delar, dels specifika för respektive forskarutförare.

Projektet har syftat till att identifiera de behov för vidareutveckling och homogenisering av beräkningsalgoritmer för att bestämma värmepumpsbaserade uppvärmningssystems effektivitet per årsbasis (SPF = Seasonal Performance Factor) i småhus och flerfamiljshus.

Rapporten är uppdelad enligt de projektdelar som angavs i ansökan. Detta motsvarar fyra delar. Under varje projekdel redovisas kortfattat det arbete och resultat inom respektive del. Mer utförlig redovisning av arbetet och resultaten inom respektive del går sedan att läsa i två appendix, en från KTH och en från SP.

Under projektet har fyra projektmöten hållits, varav en var en workshop.

Bakgrund

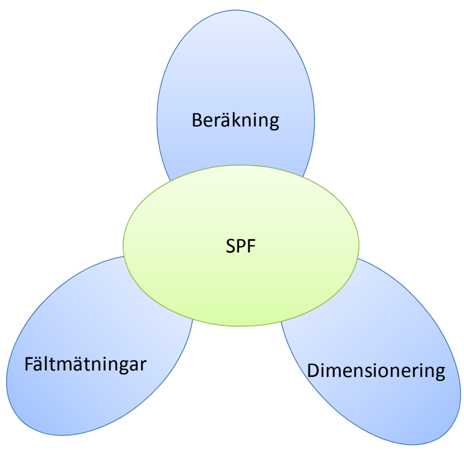
Det finns ett behov att dimensionering och energianvändning av energisystemen i byggnader beräknas och hanteras på samma sätt. Detta är viktigt eftersom vid en tänkt installation jämförs troligen olika leverantörer mot varandra, och om inte basen för deras beräkningar är gemensam går det inte egentligen att utvärdera den ene leverantörens offert mot annan leverantör.

Det finns vidare behov av två typer av beräkningsprogram för värmepumpsystem, dels ett mer detaljerat program för dimensionering av värmepumpsystem och dels en enklare transparent beräkningsmetod som kan användas för jämförelse och kvalitetssäkring av värmepumpsystem.

Grundprincipen är att en enklare transparent programvara användas för initialt val av värmepumpsystem (systemlösning, värmekälla etc.). Detta program skall även användas för jämförelser nationellt och internationellt. När kunden/slutanvändaren sedan skall installera värmepumpsystemet i sin egen fastighet är det viktigt att använda en mer detaljerad programvara så att dimensioneringen blir korrekt. För dimensionering utvecklades beräkningsprogrammet ”Prestige” inom ramen för effsys1. Detta och andra existerande dimensioneringsprogram behöver jämföras för att undersöka om de behöver utvecklas för att svara upp mot dagens behov.

Befintliga beräkningsprogram för jämförelse och kvalitetssäkring behöver vidareutvecklas så att de synliggör skillnaderna mellan olika typer av systemlösningar. Bland annat bör programmen utvecklas så att de synliggör skillnad mellan on/off reglering och kapacitetsreglering samt andra typer av teknikutveckling avseende kombinerad drift för värme och tappvarmvatten. Programvaran användas som information till slutanvändaren då de skall välja typ av värmepumpsystem och för mellan olika fabrikat. Det är viktigt att denna programvara då är transparent men ändå tillräckligt bra för att synliggöra skillnaden i energibesparing och årsvärmefaktor (SPF) mellan de olika värme-pumpsystemen, inverkan av omgivningsklimatet samt distributionssystemet i fastigheten.

Båda programtyperna bör ge som utdata både SPF och årlig energibesparing samt vara indata till beräkning av CO2 reduktion som förväntas vara en efterfrågad parameter för slutkund, beslutsfattare och tillverkare framöver.



Figur 1: SPF för olika syften.

Båda programvarorna skall använda indata från gällande harmoniserade EU standarder. Ett väl fungerande dimensioneringsprogram är viktigt för fortsatt tillväxt och acceptans för tekniken. Tillväxt kräver nöjda kunder som får väl fungerande värmepumpsystem.

Ett program för jämförelse av olika värmepumpsystem är viktigt när slutkunden skall välja systemlösning. Ur ett miljöperspektiv är det viktigt att slutkunden väljer bästa möjliga system på marknaden. Ett värmepumpsystem har en livslängd på ca 20 år.

Årsvärmefaktor

Det första fråga som inställer sig vid diskussion kring ett energisystems effektivitet är vilken effektivitet som avses. Det mest naturliga ur konsumentsynpunkt är att effektivitet på ett uppvärmningssystem är ett mått på hur mycket besparing (köpt energi, kr) som denna gör i sin fastighet. Det är inte lika intressant för konsumenten att veta att värmepumpens effektivitet är (COP_1) t.ex. 4.2 vid en viss driftpunkt. Denna information kan visserligen vara intressant vid grov utgallring av värmepumpar, då det kan tas för troligt att värmepumpens effektivitet vid en viss driftpunk har viss bäring på hur effektiv den kommer att vara.

Bättre ändå är att få information kring hur effektiv värmepumpen är vid ”typisk” användning. Problemet med värmepumpar i detta fall är att inga typiska generellt accepterade användning av värmepumpen är identifierad av branschen. Detta spår har viss potential för bedömning av en värmepumps prestanda, vilket är bättre än att sortera efter COP_1. Metoden finns t.ex. omnämnd i SS-EN 15316-4-2, kap 5.2 (CEN SS-EN 15316-4-2, 2008).

Metoden behöver som nämnts ovan definierade typiska användningar, dvs. typiska hus. Den tilltänkte kunden kunde då i princip välja ett hus liknande sitt egna, välja klimat enligt ortens placering samt jämföra olika leverantörers värmepumpsprestanda för detta väldefinierade huset vilket ger mer information än enbart COP. Under projektets gång beslöts det dock i samråd med företagsrepresentanter att detta inte skulle följas upp.

Vad avses med årsvärmefaktor? Under beaktande att kunden vill veta hur mycket energi som kundens hus förväntas behöva köpa är en lämpligt definition av årsvärmefaktorn:

$$SPF_{SYST} = \frac{Använd\ energi\ för\ uppvärmning\ eller\ kyla\ i\ byggnaden\ (Värme+El+Kyla)}{Inköpt\ energi\ (Värme+El+Kyla)}$$

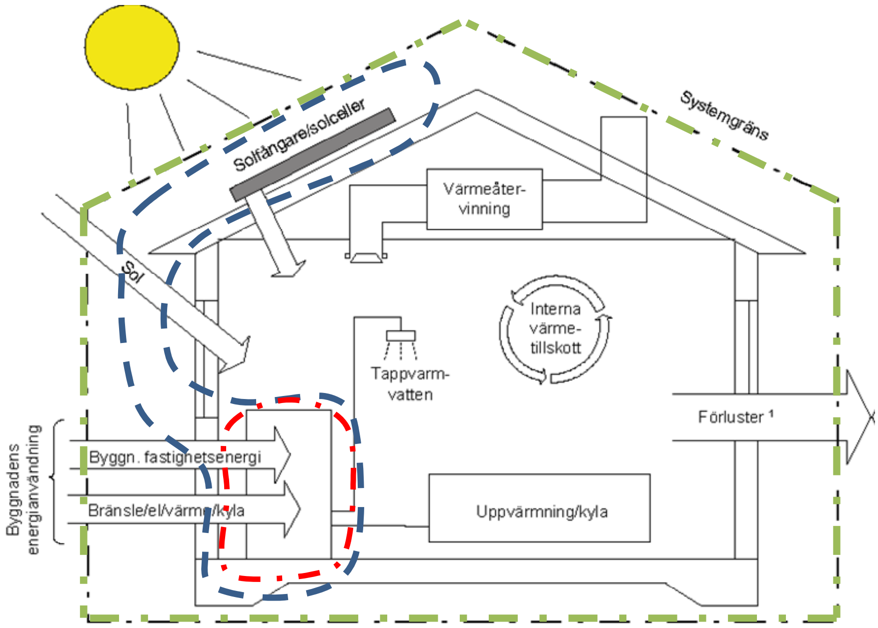
Använd energi avser alltså energi använd för uppvärmning eller kylning av byggnaden.

Det kan också vara av intresse att studera respektive värmegenererande enhets SPF. För en värmepump blir då

$$SPF_{VP} = \frac{Avgiven\ energi\ från\ värmepump}{Köpt\ energi\ tillförd\ värmepump}$$

En värmepump med liten effekt i ett system med stort effektbehov kan ha högt SPF_{VP} trots att systemet är ineffektivt.

Beroende på hur systemgränsen väljs kan olika SPF erhållas, beroende på vilken del av systemet som är av intresse, se Figur 2. Normalt är kanske byggnadens hela energieffektivitet intressant, och eventuellt de individuella värmeproducerande komponenternas effektivitet. Mer diskussion kring andra sätt att uttrycka SPF vid andra systemgränser finns i SPs unika del.



Figur 2: Olika "SPF" beroende på var systemgränsen dras.

Mer generellt beskrivs energieffektiviteten i (CEN SS-EN 15316-1, 2007) där varje (del-)system skall beräknas enligt

$$\eta_i = \frac{Q_{i,ut} + f_j \cdot E_{el,i,ut}}{f_y \cdot Q_{i,in} + f_z \cdot W_{i,aux}}$$

Exempel på primärenergifaktorer (f) att använda visas i Tabell 1 från (CEN SS-EN 15603, 2008) där det vidare hänvisas till nationell nivå, exempel ges i Annex E (CEN SS-EN 15603, 2008).

Tabell 1: Exempel på primärenergifaktorer (CEN SS-EN 15603, 2008).

	Primary energy factors f_P		CO_2 production coefficient K
	Non-renewable	Total	kg/MWh
Fuel oil	1,35	1,35	330
Gas	1,36	1,36	277
Anthracite	1,19	1,19	394
Lignite	1,40	1,40	433
Coke	1,53	1,53	467
Wood shavings	0,06	1,06	4
Log	0,09	1,09	14
Beech log	0,07	1,07	13
Fir log	0,10	1,10	20
Electricity from hydraulic power plant	0,50	1,50	7
Electricity from nuclear power plant	2,80	2,80	16
Electricity from coal power plant	4,05	4,05	1340
Electricity Mix UCPTE	3,14	3,31	617

Syftet med en sådan definition är att jämföra olika energislag och relatera dessa tillbaka till primärenergi. I föreliggande rapport kommer dock det "traditionella" begreppet att användas, utan hänsyn till primärenergifaktorer.

Del I – Inledande del (Gemensam del)

Denna del är som rubriken antyder en inledande del som syftar till att dels initiera motsvarande arbete internationellt. Vidare planerades under denna inledande del projektet tillsammans med industrirepresentanterna. I samband med detta beslutades inom projektet att fokus i slutfasen av projektet skulle vara på metodutveckling för beräkning av SPF och att det inte var realistiskt att göra ett branschgemensamt beräkningsprogram.

Under detta möte beskrevs den volym av internationell och nationella ramar som finns att ta hänsyn till, dels som regelverk, dels som standardiseringsarbete och produktmärkning.

Ett sådant direktiv som kan komma att påverka värmepumpar är Eu:s RES-direktiv[1], vilket bland annat kommer att påverka hur miljömässiga olika uppvärmningssystem är. Syftet är att öka användningen av förnybara energikällor, vilket är kopplat till EUs mål 20 % förnybara energikällor år 2020. Energibesparingar anses vara ett av de viktigaste sätten att minska energianvändningens påverkan på globala miljön. Energi från värmepumpar kan delvis få räknas som förnybar energi enligt direktivet, om värmepumpen är tillräckligt effektiv, dvs. har en SPF högre än 2.9. Det finns ingen föreslagen metod att beräkna SPF, vilket gör att föreliggande projekt kan ligga som grund för ett sådant arbete. Den förnybara andelen energi från en värmepump skall beräknas enligt

$$E_{RES} = Q_{usable} \cdot \left(1 - \frac{1}{SPF}\right)$$

vilket torde motsvara upptagen energi från omgivningen (i förångaren). Från luftsvärmepump kan inte tillgodogöra sig någon förnybar energi, då den inte tar sin energi från omgivningen. Det är specificerat i RES-direktivet vad som får räknas och vad som inte får räknas.

Inledande arbete inom ett IEA Annex pågår löpande och annexet är under uppstart. Detta arbete har precis startat och kommer att fortlöpa ett par år framåt. Projektet och Sverige är aktiva i detta arbete. Uppstartsmötet har precis varit (månadsskiftet juni/juli 2010).

[1] EU, 2008/0016 (COD)

Del II – Fältmätningar, Standards för SPF, Jämförelser (SP-unik)

Denna del av projektet är utförd av SP och mer information kan fås i ”Appendix 1 – SPs rapport”.

Rapporten beskriver först kortfattat aktuell status kring det internationella IEA Annex gällande rättvisande årsvärmefaktorer som projektet aktivt deltagit i. Uppstartsmötet var nu i sommar.

Vidare presenteras redan genomförda fältmätningar där data är av god kvalité och tillräckligt detaljerad för vidare analys. Bland en ganska omfattande volym av fältmätningar visade det sig att endast tre stycket fältmätningar motsvarade dessa kriterier, två genomförda av SP, samt det sista genomfört av Fraunhofer-Institute. Denna sista fältmätning är omfattande, ca 200 värmepumpar mäts. Tabell 2 visar vad som normalt mäts i denna omfattande studie.

Tabell 2: Mätpunkter som används i Fraunhofer´s fältmätningsprojekt.

	Running time (min) Sum	Energy content (kWh) Sum	Energy consumption (kWh) Sum	Inlet temp. (°C) Average	Outlet temp. (°C) Average	Volume flow (l/h) Average	Delivered heat during operation (kW)	Average power during operation (W) Average
Heat Pump, total	X		X				X	X
Compressor	X		X					X
Warm heat transfer medium circuit		X		X	X	X	X	
Cold heat transfer medium circuit (brine)		X		X	X	X	X	
Space heating circuit		X		X	X	X	X	
Domestic hot water circuit		X		X	X	X	X	
Supplementary heater			X					X
Measurement equipment			X					X
Pump, space heating circuit			X					X
Pump, warm heat transfer medium circuit			X					X
Pump, cold heat transfer medium circuit (brine)			X					X

Även standardliknande metoder för fältmätningar har kartlagts, där fyra metoder identifierats, tre från Nordtest (NT VVS) och en från SP. SPs metod för luft/luft värmepumpar anses ge COP-värden med mindre osäkerhet än 10 %. NT VVS har tre olika nivåer, 5 %, 10 % och 15 % osäkerhet.

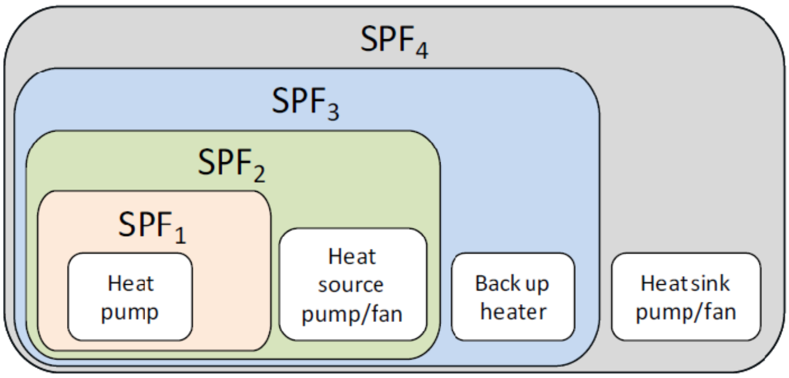
Vidare har standards för beräkning av SPF identifierats, hela 20 stycken. En genomgång av dessa görs också, där deras respektive styrkor och svagheter presenteras. Tabell 3 illustrerar de olika standarders omfång och krav.

Jämförelse görs sedan mellan olika beräkningsmetoder och fältmätningar. Fyra olika SPF-nivåer är identifierade, se Figur 3. Resulterande SPF för olika installationer visas i Figur 4. I Figur 5 visas skillnaden mellan fältmätningar och olika beräkningsmetoder. Det noteras även att samma värmepump ger olika SPF, vilket kan peka på vikten av korrekt dimensionering av värmepumpen till husets faktiska behov.

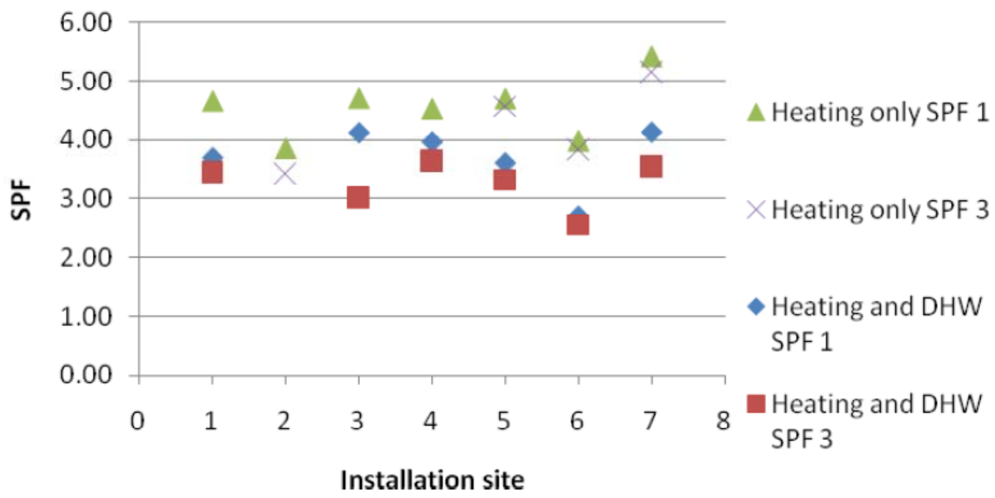
Till sist ges rekommendation för nya modeller för beräkning av SPF från laboratoriemätningar.

Tabell 3: Sammanfattning av olika standarders omfång.

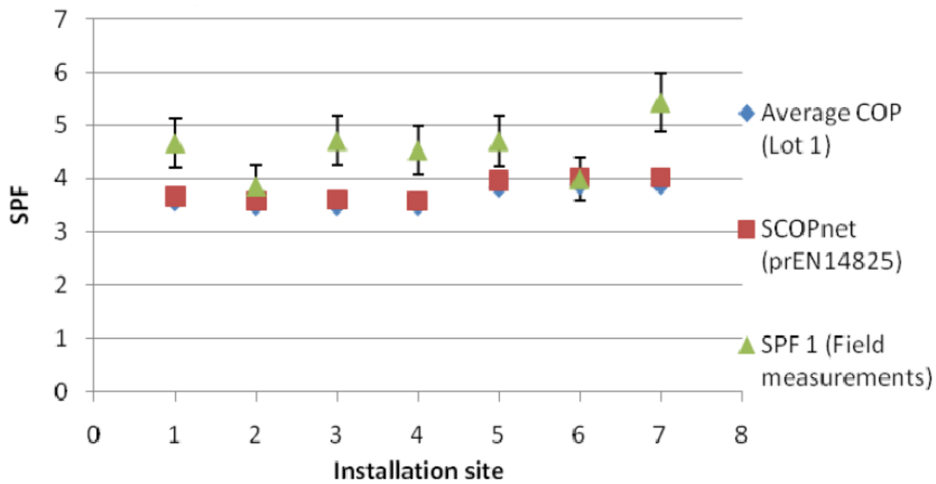
	Type of standard			Type of heat pump			Operation					Requirements				Aspects in capacity calculations					Calculations of	
	Laboratory tests	Field tests	Calculation model for SPF	ASHP	GSHP/WSHP*	AIR/AIR	Heating	Cooling	Domestic hot water	Combined operating	Part load conditions	Steady state	Permissible deviations	Uncertainty of measurements	Test set up/ performance of measurement	Pumps and fans included	Defrost period	Standby losses	On/off cycles capacity regulation	Other	COP/EER	SPF/SEER
NT VVS 076				x	x	x	x	x			x	x	x	x	x						x	x
NT VVS 115				x	x	x	x	x			x	x	x	x	x						x	x
NT VVS 116				x	x	x	x	x				Δ	Δ	x	x						x	
SP 1721						x	x	x				x	x	x	x						x	
ASHRAE standard 37	x			x	x	x	x	x				x	x	x	x		x			x		
AHRI 210/240	x					x	x	x			x	x	x	Δ	x				x		x	x
AHRI 870-2005	x				x		x	x			x	Δ	Δ	Δ	Δ						x	
AHRI 390-2003	x					x	x	x			x	Δ	Δ	Δ	Δ		x				x	
AHRI 320-1998	x				x*		x	x			x	Δ	Δ	Δ	Δ						x	
AHRI 325-1998	x				x		x	x			x	Δ	Δ	Δ	Δ	x					x	
AHRI 330-1998	x				x		x	x			x	Δ	Δ	Δ	Δ	x					x	
EN14511	x			x	x	x	x	x				x	x	x	x	x	x					
prEN14511	x			x	x	x	x	x				x	x	x	x	x	x					
EN 255-3	x			x	x				x			x	x	x	x	x	x	x			x	
prEN 255-3	x			x	x				x			Δ	Δ	Δ	Δ	x	x	x			x	
TS14825	x			x	x	x	x	x			x	Δ	Δ	Δ	Δ	x	x		x		x	
prEN14825	x		x	x	x	x	x	x			x	Δ	Δ	Δ	Δ	x	x		x		x	x
EN15316-4-2			x	x	x	x	x		x	x	x	α	α	α	α	α	α	α	x	α		x
EuP Lot 1	x		x	x	x		x				x	x	x	x	x	x	?		x		x	x
EuP Lot 10	x		x			x	x	x		x	x	Δ	Δ	Δ	Δ	x	x		x		x	x



Figur 3: Olika nivåer av SPF enligt EU-projekt SEPEMO.



Figur 4: SPF för 7 installationer.



Figur 5: Jämförelse mellan fältmätningar och olika beräkningsmetoder.

Del III – Kartläggning av dimensioneringsprogram (KTH-unik)

Denna del av projektet har utförts av KTH och mer finns att läsa i ”Appendix 2 – KTHs rapport”. Här identifieras begränsningar och möjligheter i systemlösningar och metodik hos dimensionerings-program liknande PRESTIGE. En viktig del i arbetet var att kartlägga hur samtidig produktion av tappvarmvatten och värme hanteras i programmen.

För att undersöka bl.a. detta har en enkät skickats ut som leverantörerna har besvarat. I Tabell 4 visas just för tappvarmvatten att alla leverantörer som besvarat enkäten hanterar tappvarmvatten efter beräkningen, utom PRESTIGE, som gör varmvatten varje timme istället.

Tabell 4: Hantering av tappvarmvatten i fyra olika program.

	Används inte	Post Schablon	Post Faktisk	Integrerad Schablon	Integrerad Faktisk
Program 1				X	X
Program 2			X		
Program 3			X		
Program 4			X		

En annan aspekt som var extra intressant att jämföra är om avancerad kapacitetsstyrning hanteras i programmen. Inte alla program hanterar mer avancerad kapacitetsreglering, vilket kan se i Tabell 5. Detta kan bero på flera saker, bland annat att leverantören inte har just den typen i sitt sortiment.

Tabell 5: Kapacitetsstyrningsmöjligheter i fyra olika program.

	Program 1	Program 2	Program 3	Program 4
Kompressor, ON/OFF Pump kont.	X	X	X	X
Kompressor, ON/OFF Pumpar, frekvens			X	X
Kompressor, frekvens Pump kont.				X
Kompressor, frekvens Pumpar, frekvens		X		X
Modulinkoppling		X	X	

Noterbart kan också vara att alla program kan hantera el och olja som kompletterande värmekälla, medans ingen av de som svarat kan hantera solpaneler som kompletterande värmekälla.

Det kan konstateras att visst behov av utveckling är önskvärd kring möjligheterna att hantera komplicerade systemlösningar, då denna möjlighet inte finns för kunder som t.ex. redan har solpaneler installerade. Dessa kan i dagsläget inte få bra data kring eventuell besparing vid installation av värmepump i sin byggnad.

Hela denna ovantstående enkät finns bifogad i KTHs delrapport (Appendix 2 – KTHs rapport).

Vidare i KTHs delrapport redovisas ett studentarbete som använt tre olika program och jämför deras användbarhet och begränsningar.

Del IV – Fördjupad analys och metodutveckling (KTH-unik)

Denna del av arbetet har utförts av KTH och finns att läsa mer utförligt i ”Appendix 2 – KTHs rapport”

Denna del innehåller arbete som inbegriper direkt jämförelse mellan olika leverantörers resultat när det gäller en fiktiv värmepumps prestanda i form av SPF. Vidare har metodik utformats som kan användas för bedömning av värmesystems prestanda i byggnader. Metodiken bygger till stor det på befintliga standarder, där så varit möjligt.

Jämförelse av SPF för fyra fiktiva byggnader och värmepumpar

I detta avsnitt har befintliga program använts för att uppskatta energianvändningen i fyra olika hus med fyra olika värmepumpar, alla fiktiva. Nedan visas vilka indata som användes för byggnaderna och de tänkta värmepumparna.

Hus 1:

Indata:
124 m², Fristående 1.5 plan, ingen källare, ej inredd vind.
Stockholm, Bromma.
Vattenburen värme (45/35).
Innetemperatur 22°C.

Uteluft VP med eltillsats (95% verkningsgrad).

13 500 kWh för uppvärmning (värmeenergi).
4 500 kWh för tappvarmvatten.

Avgiven värmeeffekt [kW]

VB	35	50	Max
Utetemp -7(-8)	2.47	2.05	1.96
+2(1.5)	3.47	3.09	2.975
+7(6)	4	3.63	3.485

Tillförd eleffekt [kW]

VB	35	50	Max
Utetemp -7(-8)	0.96	1.05	1.09
+2(1.5)	1.03	1.15	1.23
+7(6)	1.06	1.208	1.303

Max framledningstemperatur [°C]	55
Max returledningstemperatur [°C]	48
Nominell eleffekt KB-pump [W] (anges endast för indirekt system)	
Lägsta drifttemperatur [°C]	-10

Hus 2

Indata:
176 m², Fristående, byggår 1966, Tung byggnad (150timmar tidskonstant), 2 plan, med inredd källare, ej inredd vind.
Luleå
Vattenburen värme (55/45)
Innetemperatur 22°C

Bergvärme med eltillsats (95% verkningsgrad). Berg Granit med 3.4 W/mK. Medeltemperatur på Brine -3°C

31 500 kWh värmebehov (värmeenergi).
4 500 kWh varmvattenbehov

Vätska/Vatten

Avgiven värmeeffekt [kW]

VB	35	50	Max
KB -5	6.963	5.742	5.28
0	8.338	7.205	6.743
+5	9.636	8.635	8.217

Tillförd eleffekt [kW]

VB	35	50	Max
KB -5	1.907	2.177	2.248
0	2	2.377	2.483
+5	2.095	2.554	2.707

Max framledningstemperatur [°C]	55
Max returledningstemperatur [°C]	48
Nominell eleffekt KB-pump [W]	150

Hus 3

Indata:
153 m², Radhus, byggår 1970, Medeltung byggnad (95 timmar tidskonstant), 2 plan, ingen källare, ingen vind.
Nybro
Direktverkande elradiatorer
Innetemperatur 22°C

Luft/Luft värmepump, elradiatorer tillsats (100% verkningsgrad)

19 890 kWh värmebehov (värmeenergi).
3 500 kWh varmvattenbehov

Värmepumpprestanda:

Uteluft (°C)	Värme (kW)	Kompressor (kW)
7	5	3.2
2	3.4	2.6
-7	3	2.4
-15	2.8	2.3

Hus 4

Indata:
180 m², Fristående, byggår 2009, Lätt byggnad (24 timmar tidskonstant), 2 plan, ingen källare, ingen vind.
Borås
Vattenburen golvvärme (35/28)
Innetemperatur 22°C

Mark/Vatten värmepump, eltillsats (95% verkningsgrad)

19 800 kWh värmebehov (värmeenergi).
4 500 kWh varmvattenbehov

Vätska/Vatten

Avgiven värmeeffekt [kW]

KB \ VB	35	50	Max
-5	3.608	2.904	2.673
0	4.29	3.652	3.421
+5	4.972	4.356	4.125

Tillförd eleffekt [kW]

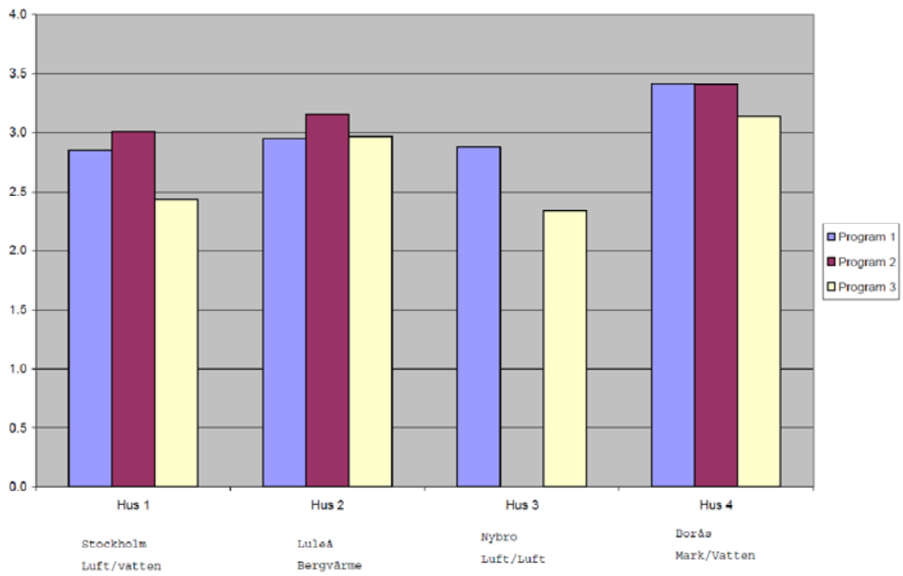
KB \ VB	35	50	Max
-5	1.083	1.165	1.177
0	1.165	1.306	1.342
+5	1.212	1.424	1.483

Max framledningstemperatur [°C] 55

Nominell eleffekt KB-pump [W] 60

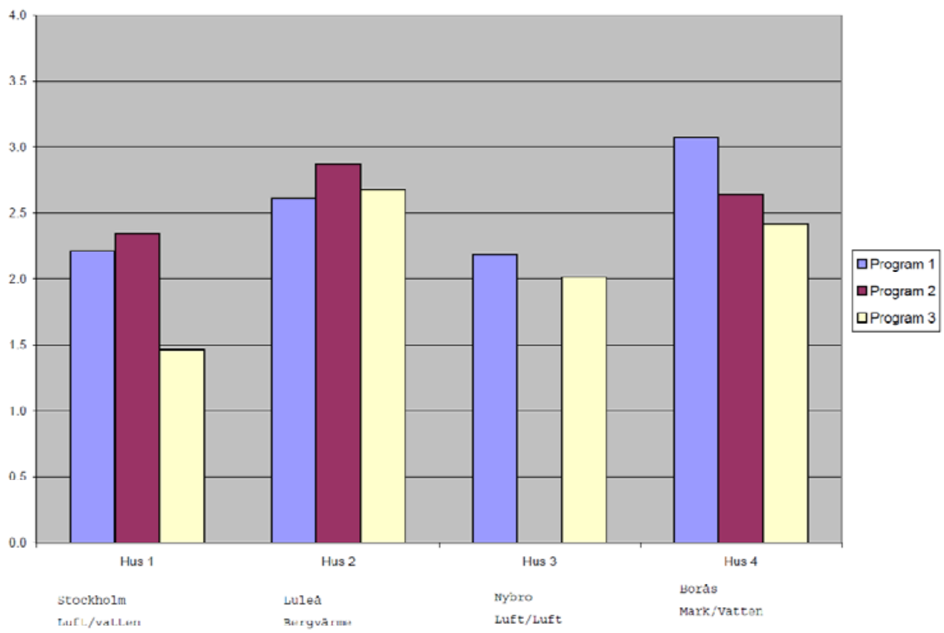
Max returledningstemperatur [°C] 48

Leverantörerna fick själva räkna igenom byggnaderna i sitt respektive program. Två olika nivåer av SPF har beräknats, dels bara för värmepumpsenheten, dels för uppvärmningssystemet. Figur 6 visar resultaten för värmepumpsenheten. Det är tydligt att Program 3 beräknar lägre SPF än de övriga två. Delvis förklaras detta av att i prestandadata ingår energi för avfrostning, men Program 3 har lagt till den ytterligare en gång.



Figur 6: SPF för värmepumpenheten i de simulerade byggnaderna.

I Figur 7 redovisas motsvarande resultat för värmesystemet, dvs inklusive all köpt energi, vilket uteslutande är tillsatsvärme. Skillnaderna mellan Program 3 och övriga blir nu naturligt mycket större, då en för lågt uppskattad prestanda på värmepumpsenheten kompenseras genom mer tillsatsvärme, som i detta fall är elektricitet med en värmefaktor på 1 (i bästa fall). Det är tydligt att viss skillnad föreligger även om undersökningen inte är komplett då det bara är tre program. Skillnaden är dock inte exceptionellt stora. Å andra motsvarar skillnaderna i SPF en skillnad på upp till två tusen kronor i energikostnad per år för de båda brine/vatten-värmepumparna. Denna differens är inte helt försumbar. En förklaring till viss del av skillnaden kan vara att olika klimatdata används i simuleringarna. Det är den enda återstående ”externa” felkälla.

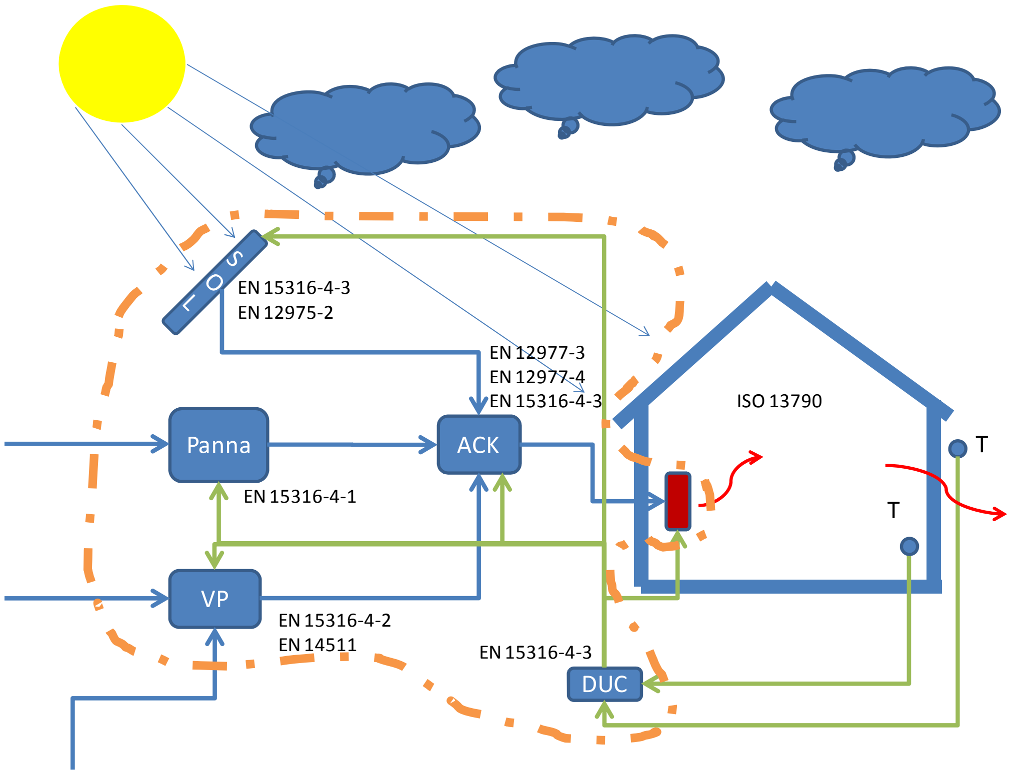


Figur 7: SPF för värmesystemet i de simulerade byggnaderna.

Det kan alltså konstateras att det är viktigt att samma indata används för de olika programmen, att modellerna är lika, så att ett rättvisande resultat kan erhållas.

Metod för beräkning av SPF för uppvärmningssystem i bostäder

Baserat på de tidigare redovisade resultaten bedömdes det att en gemensam beräkningsgrund för dimensionering och prestandabedömning behöver utvecklas, vilket redovisas i denna del av rapporten. Mer detaljer finns att läsa i KTHs rapport i appendix. Metoden baseras till största delen på europeiska standarder för beräkning av energianvändning i byggnader och dess energisystem, se Figur 8. Arbetet kan möjligen ses som en harmonisering av beräkningsmetodik för i huvudsak svenska värmepumpsleverantörer till europeisk standard. I vissa fall har avsteg gjorts, där istället delar ur PRESTIGE har använts. För andra delar har ingen lämpligt metodik identifierats vilket då kan vara öppet för modifieringar i framtiden.



Figur 8: Metoduppbyggnad för beräkning av en byggnads energisystems prestanda

Ett viktigt avsteg från standard är att både värmepumpar och solfångare skall beräknas i tidssteg per timme. Detta följer principen från den mer övergripande standarden för byggnadens energi-användning, SS-EN ISO 13790.

Anledningen är att syftet med projektet är utarbeta en metodik som beaktar modern systemlösningar och styrprinciper. Inverkan av ”smart” styrning torde vara svår då man dels för värmepumpar antas jobba med temperatur-bin, och för solfångare med månadssteg. Det föreslås alltså att alla komponenter beräknas per timme.

Ett exempel har implementerats i MS Excel för en uteluft/vatten-värmepump, för att illustrera konceptet. Metodiken som föreslås är alltså helt transparent, inga osynliga funktioner eller algoritmer finns. Excelmodellen är lite långsam och syftet är inte att alla ska använda denna, utan att den kan användas som referens.

Ytterligare versioner av värmepumpar får implementeras vid ett senare tillfälle. Avsikten då är att implementera de värmepumpsmodeller som finns i PRESTIGE. För närvarande saknas då i huvudsak markvärmepump och bergvärmepump.

Appendix 1 – SPs rapport

Calculation methods for SPF for heat pump systems for comparison, system choice and dimensioning

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SP Technical Research Institute of Sweden

Energy Technology

SP Report 2010:49

Calculation methods for SPF for heat pump systems for comparison, system choice and dimensioning

Roger Nordman, Kajsa Andersson, Monica Axell, Markus Lindahl

SP Technical Research Institute of Sweden

Abstract
In this project, results from field measurements of heat pumps have been collected and summarised. Also existing calculation methods have been compared and summarised. Analyses have been made on how the field measurements compare to existing calculation models for heat pumps Seasonal Performance Factor (SPF), and what deviations may depend on. Recommendations for new calculation models are proposed, which include combined systems (e.g. solar – HP), capacity controlled heat pumps and combined DHW and heating operation.

Key words: Heat pump, SPF, calculation model, field measurements

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SP Technical Research Institute of Sweden

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Preface

This report summarize the findings from SP Technical Research Institute of Sweden in the joint KTH-SP project “Calculation methods for SPF for heat pump systems for comparison, system choice and dimensioning”, project P9 in the Effsys-2 research programme, financed by the Swedish Energy Administration and participating companies and organizations.
The project was set up so that SP and KTH performed separate parts of the projects, but with discussions and meetings in between.
The project parts are reported according to the parts stipulated in the application.

Sammanfattning

I denna rapport redovisas de delar av projektet ”Beräkningsmetoder för årsvärmefaktor för värmepumpsystem för jämförelse, systemval och dimensionering” som SP Sveriges tekniska forskningsinstitut svarat för. Projektet har genomförts av SP och KTH. KTH:s del av projektet redovisas i en separat rapportdel.

I en inledande del av projektet har förberedelser för ett IEA samarbete, samt gemensam övergripande projektplanering tillsammans med industriparterna utförts. IEA-projektet har godkänts att starta av styrelsen för IEA Heat Pump Programme, och ett första inledande möte har hållits.

SP har koordinerat samt sammanställt resultat av fältmätningar. Väl genomförda fältmätningar är en förutsättning för validering av olika beräkningsalgoritmer. Sammanställningen visar att det finns ett flertal utförda fältmätningar i Sverige under de senaste 20 åren, men få har gjorts med SPF som fokus, utan ofta har mätningarna gjorts med syfte att studera en viss teknikförändring, eller andra faktorer. Det har inte under de senaste 10 åren utförts någon stor mätning på värmepumpar liknande de välkända Fraunhofermätningarna eller FAVA-studien i Schweiz. Den enda studie som syftat till att mäta SPF är den som SP utfört. Detta kan ses som en brist i ett land där värmepumpar har ett så stort genomslag för uppvärmningen av bostäder.

En kravspecifikation för mätdata som behövs för att användas för validering har tagits fram.

En sammanställning av befintliga standardliknande beräkningsmetoder (existerande algoritmer) för SPF har gjorts. Syftet med analysen har varit att beskriva existerande algoritmer (modeller) samt kartlägga om nuvarande program (Annex 28, SP´s beräkningsprogram mm) innefattar alla typer av värmepumpsystem som finns på marknaden idag. En viktig del är att undersöka hur kombinerad drift dvs. tappvarmvatten och värme behandlas i modellerna. En annan fråga är huruvida olika typer av kapacitetsreglering behandlas. Sammanställningen har visat att det finns en stor brist bland förekommande program och metoder vad gäller att ta hänsyn till :

- Kombisystem, såsom sol-vp
- Kapacitetsreglerade system
- System med kombinerad varmvattentillverkning och uppvärmning

Existerande algoritmer har jämförts med resultat från fältmätningar. Från existerande fältmätningar har data tagits för att jämföra resultaten med befintliga metoder för att beräkna SPF. En analys av hur väl dessa metoder förmådde beräkna SPF för de studerade systemen har gjorts. Denna analys visar att resultaten från fältmätningarna ofta visar på högre SPF än vad som beräkningsmodellerna ger. Det finns flera orsaker till detta, bland annat att modellerna använder sig av konstant marktemperatur (som i förekommande fall är lägre än verklig marktemperatur), att modellerna använder en bivalent punkt som aldrig uppträtt i de verkliga mätningarna mm. Den gjorda jämförelsen visar på ett antal viktiga faktorer att studera vidare.

För att utveckla ett enkelt program för jämförelse av värmepumpsystem är det viktigt att begränsa beräkningarna till ett antal klimatzoner och ett antal typhus. Målet är att beräkningsmetoden skall kunna användas både nationellt och internationellt. I ett dimensioneringsprogram skall däremot stor frihet ges att definiera det specifika huset för att utförligt kunna studera de behov som finns för de specifika installationerna.

En ny beräkningsmetodik för SPF och årsenergibesparing baserad på, eller som ersättning för existerande algoritmer som input för nytt Annex inom IEA HPP och Europastandard (CEN) har diskuterats. Det gemensamma beräkningsprogrammet skall baseras på indata från gällande

Europeiska standarder (EN 14511) för kombinerad drift med värme och tappvarmvatten. Det skall även till fullo implementera rutiner för drift med kapacitetsreglerade värmepumpar (kompressorer och pumpar/fläktar).

Förslag till vad som bör ingå i ett nytt transparent gemensamt beräkningsprogram som kan användas för jämförelse och certifiering har getts. Industrigruppen menade tidigt att det viktiga i denna del är att ta fram de samband som bör implementeras i ett beräkningsprogram, men att de själva oftast skriver in-house kod som de kan implementera dessa samband i. Detta gör att förutsättningarna blir likartade, men att tillverkarna fortfarande kan ha sina specifika (ofta hemliga) indata själva.

1 Introduction

The existing calculation tools for 1) design and 2) comparison need to be further developed to show the potential with new technology such as capacity controlled systems and more efficient system for combined operation with space heating and domestic hot water production. The overall aim is to develop existing tools for future needs. The outcome from the calculation tools should be useable for calculation of environmental impact. The purpose is to compare existing tools for calculation of seasonal performance factor and annual energy savings in order to propose needs for further development. For validation of the calculation tools existing data from laboratory and field measurements will be used.

Seasonal Performance Factor, SPF, is a term used mainly for real installations, compared to the Coefficient of performance, COP, which is evaluated in controlled lab environment. How SPF is estimated depends on the situation under which it is evaluated, see Figure 1 below.

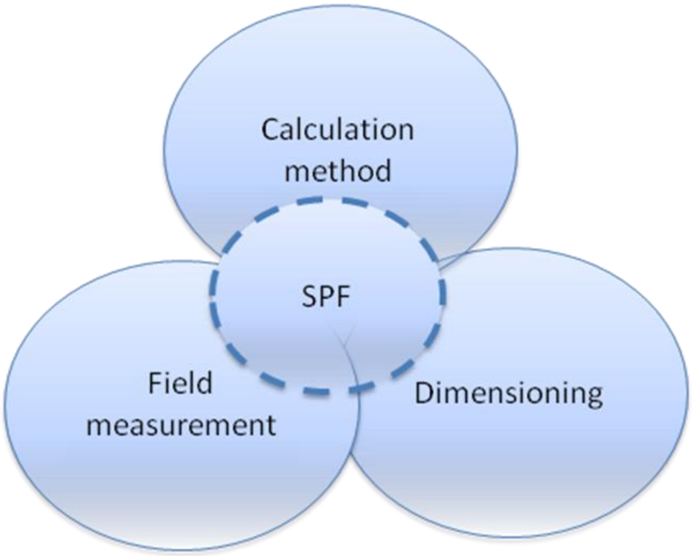


Figure 1. SPF can be determined in various ways, including field measurement, calculation methods and dimensioning software.

Based on lab measured performance data, SPF can be calculated according to calculation methods, that normally relates performance data in specific operating modes to annual climatic conditions, expressed as “bin models” where the number of hours in a year the temperature is between certain values are binned together. Model buildings are normally used to give annual heat demands and overall heat transfer resistances of the building.
For the installer of heat pumps, more specific details of the building must be prompted, as well as detailed data about the ground properties in the case of GSHP’s. Local climatic data is also used for estimating the heat demand. The climatic data contains a cold shock in order to dimension the heat pump capacity to extreme conditions that may occur during the lifetime of the installation. Other data such as the number of occupants, Domestic Hot Water (DHW) energy consumption is also normally entered in the software models for dimensioning.
To evaluate the real performance of the installed heat pump, field measurements are carried out to relate the useful heat produced to the energy input, often electrical power (but it could also be heat driven processes). The SPF of the heat pump is then often expressed as the ratio of the heat delivered to the heat distribution system (including DHW when relevant) to the electricity to operate the heat pump (including electricity to operate pumps and fans to bring the heat source to the heat pump).
The different level of detail given as input in the different stages of SPF calculation will lead to different SPF values. The main objective of this project is to identify what needs to be included in a new calculation method in order to better represent the real SPF of the heat pump in the building system.

2 Preparing an IEA HPP Annex on SPF

Preparations for an IEA annex on SPF have included preparatory meetings, and communication with research communities involved in the IEA HPP sphere. Meetings include a meeting during the ASHRAE winter Conference 2009 [1.1.1.1.11], NT meeting in Borås, September 2009 , and a Meeting in Paris march 5th, 2010 [2].
A draft legal text was prepared and circulated among interested parties and the executive committee in HPP. The draft legal text was discussed in the ExCo meetings in Rome, November 2009 and in Helsinki June 2010. In the Helsinki meeting it was suggested that the annex proposal for "Dynamic testing of heat pumps" should be integrated with the SPF annex. The kick-off meeting for the SPF Annex in June 30th- July 1st 2010 will discuss the possibility for this integration. The legal was just recently approved by the ExCo [3].
The preparation and starting up of the international Annex has taken much more time than expected, mainly due to constraints in timing and funding. However, on June 30 –July 1st, the kick-off meeting for the new annex is held in Albuquerque, New Mexico.

3 Summary of already performed field measurements.

In order to evaluate already made field measurements in Sweden, or made by Swedish manufacturers, meetings in the project discussed earlier made field measurements. The result is that there has been a large number of field measurements made during the last decades, see Appendix 1 and references [4-6], but few studies have had the specific goal to examine the SPF.
In order to make detailed analyses of the performance, also detailed data from the measurements are needed, and this was only available in two studies, the SP study ”Erfarenheter från fältutvärdering av fem bergvärmepumpar i Sjuhärad” and the Fraunhofer study “Heat Pump Efficiency” where a number of Swedish heat pump manufacturers participated with heat pump units. For Air-air heat pumps, only one study has been found [7]. These three studies are describes more in detail below.

3.1 Description of evaluated field measurements

3.1.1 Fraunhofer

The Fraunhofer-Institute for Solar Energy Systems ISE is running two large field monitoring project including approximately 200 heat pumps in total. The heat pump efficiency project includes approximately 110 installed heat pumps with a heating capacity of 5-10 kW. In the Replacement of Central Oil boilers with Heat Pumps in Existing Building Project 75 heat pumps are included. The heat pump types included are air to water, ground source and water to water heat pumps. In this study two heat pump producers, IVT and Nibe, have provided the project with data based on the field measurements in the Fraunhofer study.

3.1.1.1 Measured parameters

Table 1 gives an overview of the parameters normally measured in the Fraunhofer field measurements. Exactly what parameters tested might differ from test site to test site. For some test sites additional equipment are measured as well. Examples of such equipment are circulation pumps or control equipment.

Table 1. Measured parameters for brine to water heat pumps in the Fraunhofer study.

	Running time	Energy content	Energy consumption	Inlet temp.	Outlet temp.	Volume flow	Delivered heat during operation	Average power during operation
	(min)	(kWh)	(kWh)	(˚C)	(˚C)	(l/h)	(kW)	(W)
	Sum	Sum	Sum	Average	Average	Average		Average
Heat Pump, total	X		X				X	X
Compressor	X		X					X
Warm heat transfer medium circuit		X		X	X	X	X	
Cold heat transfer medium circuit (brine)		X		X	X	X	X	
Space heating circuit		X		X	X	X	X	
Domestic hot water circuit		X		X	X	X	X	
Supplementary heater			X					X
Measurement equipment			X					X
Pump, space heating circuit			X					X
Pump, warm heat transfer medium circuit			X					X
Pump, cold heat transfer medium circuit (brine)			X					X

For air to water heat pumps included in the study many of the measured parameters are the same. The data related to the cold heat transfer medium are replaced with data regarding fans in the outdoor unit. Additional the outdoor temperature and the humidity are measured for air to water heat pumps whereas it is not for the brine to water heat pumps.

Table 2. Measured parameters for air to water heat pumps in the Fraunhofer study.

	Running time	Energy content	Energy consumption	Inlet temp.	Outlet temp.	Volume flow	Delivered heat during operation	Average power during operation
	(min)	(kWh)	(kWh)	(˚C)	(˚C)	(l/h)	(kW)	(W)
	Sum	Sum	Sum	Average	Average	Average		Average
Heat Pump, total	X		X				X	X
Compressor	X		X					X
Warm heat transfer medium circuit		X		X	X	X	X	
Space heating circuit		X		X	X	X	X	
Domestic hot water circuit		X		X	X	X	X	
Supplementary heater			X					X
Measurement equipment			X					X
Pump, space heating circuit			X					X
Pump, warm heat transfer medium circuit			X					X
Fan			X					X

3.1.1.2 System boundaries

The system overview below shows the placement of the measurement equipment. The figure shows a general system, the real systems are many times more complicated and will not fit into the general description. In these cases additional meters are installed in order to be able to monitoring the system in a good way.

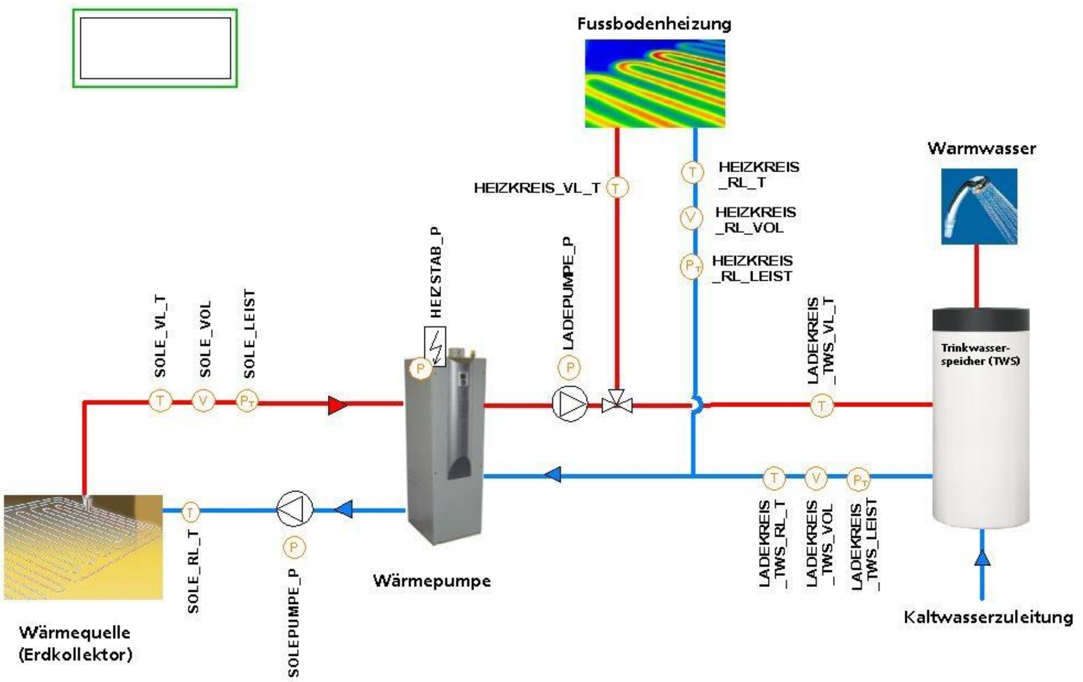


Figure 2. System overview, placement of measurement equipment.

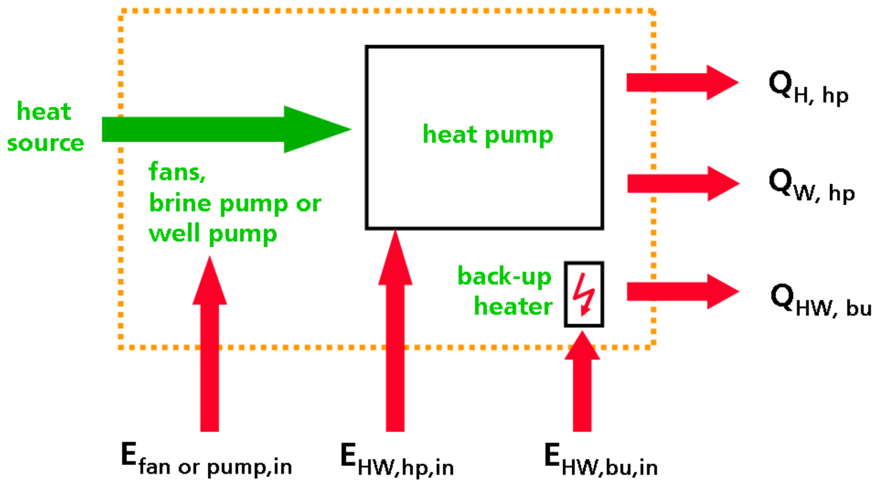


Figure 3. Schematic overview of used and delivered energy**.**

3.1.1.3 Sampling interval

The data are collected automatically and stored every minute. The stored data are remotely accessible by a GSM modem and transferred to the Fraunhofer Institute, followed by an automatic saving and sorting of the data. An automatic test of plausibility is also done, using specially made software.

The data used in this SPF project are presented as daily averages.

3.1.1.4 Measurement equipment

The meters are generally located at both the source and the heat side. For systems equipped with buffer tanks, the meters are installed before the tanks if possible. The meters are installed as close to the heat pump as possible but after the split of warm hot water transfer hot water into space heating and domestic water circuits. This in order to be able to measure energy amounts used for both space heating and domestic water separately.

Ultrasonic heat measuring device combined with data loggers are used to measure the produced heat. Temperatures, volume flows, amount of accumulated heat, electricity consumption of pumps and other equipment are measured by means data loggers.

3.1.1.5 Measurement uncertainty

No information about measurement uncertainty was provided in the Fraunhofer studies.

3.2 Measurement of ground source heat pumps

In 2003-2004 SP made a field measurements including five ground source heat pumps located in the Boras area. The study named "Årsmätningar av fem bergvärmeanläggningar i Sjuhärad" [6]. The measurements were performed from November 2003 to November 2004.

3.2.1 Measured parameters

The following parameters are measured:

- Thermal heat content, space heating
- Thermal heat content, tapped sanitary hot water
- Electricity consumption, total heat pump
- Electricity consumption, supplementary heater

- Indoor temperature
- Outdoor temperature
- Brine temperature, inlet (3 of 5 units)
- Brine temperature, outlet (3 of 5 units)
- Compressor, running time

Heat meters were installed between the space heating system and the heat pump, the same was done for the tapped sanitary hot water. Thereby internal heat losses were not measured. The meters was installed as close to the heat pump as possible in order to minimize the influence of these losses.

The electricity consumption of the supplementary heater was measured indirectly by measuring the running time and the instantaneous power for each efficiency step.

The indoor and outdoor temperatures were logged continuously. The indoor meter was placed centrally in the building with no influence of sunshine or other sources of interference. The outdoor meter was placed on the north or northeast façade.

3.2.1.1 Sampling interval

Table 3. Measured parameters and sampling interval

Thermal heat content, space heating	Once per week
Thermal heat content, tapped sanitary hot water	Once per week
Electricity consumption, total heat pump	Once per week
Electricity consumption, supplementary heater	Once per week
Indoor temperature	Every 20 minutes
Outdoor temperature	Every 20 minutes
Brine temperature, inlet (3 of 5 units)	Every 10 minutes
Brine temperature, outlet (3 of 5 units)	Every 10 minutes
Compressor, running time	Once per week

3.2.1.2 Measurement equipment

The measurement equipment used is listed in **Fel! Hittar inte referenskälla.**. The equipment used for measuring the brine temperature is not specified.

Table 4. Measurement equipment

Electrical energy	ABB Deltameter CBB 211700
Running time	Paladin
Electrical power	Siemens B4301
Heat meter	Siemens Ultraheat 2WR5151
Indoor temperature	Easy Log 24 RFT
Outdoor temperature	Easy Log 40 KH
Brine temperature	Not specified

3.2.1.3 Measurement uncertainty

No information about measurement uncertainty in the report.

3.3 Field measurement of air-to-air heat pumps

From March 2008 to February 2009 SP Technical Research Institute of Sweden made a field measurement of five air-to-air heat pumps in the Borås area. The results from the measurements are presented in SP report 2009:26 “Fältmätning av Luft/Luft värmepumpar I svenska småhus”. [7]

Electricity consumption and temperatures was logged continually and five performance tests were made during the year. The performance tests were planned to be made at different outdoor temperatures. Two test during spring and autumn and one during the winter. But due to the mild winter and divergence between the weather forecast and the actual weather conditions at the test site the planed dissemination was not reached. The performance test follows SP method no. 1721 [11].

3.3.1.1 Measured parameters

The following parameters are measured and logged continually:

- Electricity consumption, total to the building
- Electricity consumption, heat pump
- Electricity consumption, supplementary heat
- Indoor temperatures in tree rooms
- Outdoor temperatures
- Outdoor humidity

The following parameters are measured during the performance test due to SP method no. 1721:

- Airflow from indoor unit
- Air temperature before the indoor unit
- Air temperature after the indoor unit
- Electrical power, heat pump
- Air pressure

Figure 4. Measurement equipment due to SP Method no. 1721

3.3.1.2 Sampling interval

Table 5. Measured parameters and sampling interval

Electricity consumption, total to the building	
Electricity consumption, heat pump	Every 5 minutes
Electricity consumption, supplementary heat	Every 5 minutes
Indoor temperature	Every 20 minutes
Outdoor temperature	Every 20 minutes
Thermal heat content, space heating	5 measurements
Electricity consumption, heat pump	5 measurements

3.3.1.3 Measurement equipment

Table 6. Measurement equipment

Electricity consumption	ABB Deltameter CBB 211700
Logger pulse	Easy Log 40 IMP
Logger air temperature and humidity	Easy Log 24 RTF
Logger outdoor temperature	Easy Log 40 KH
Flow meter, air	VEAB
Air pressure meter	Testo 511
Temperature meters	PT100
Pressure meter	
Data logger	
Meter electrical power	

3.3.1.4 Measurement uncertainty

If the demands stated in SP method no. 1721 is fulfilled the Coefficient of performance (COP) can be calculated with an uncertainty lower than ±10%. The yearly delivered heat from the heat pump can be calculated with an uncertainty of ±20%.

The results presented follow the standard SP 1721. The capacity of the heat pump is measured during stable conditions and is not including any defrost cycle. Thereby the results for the SPF are based on data from the heat pump running at stable conditions, which will lead to an overestimation of the SPF. For COP calculations uncertainty will be smaller, since the output of heat is more or less proportional to the electricity consumption.

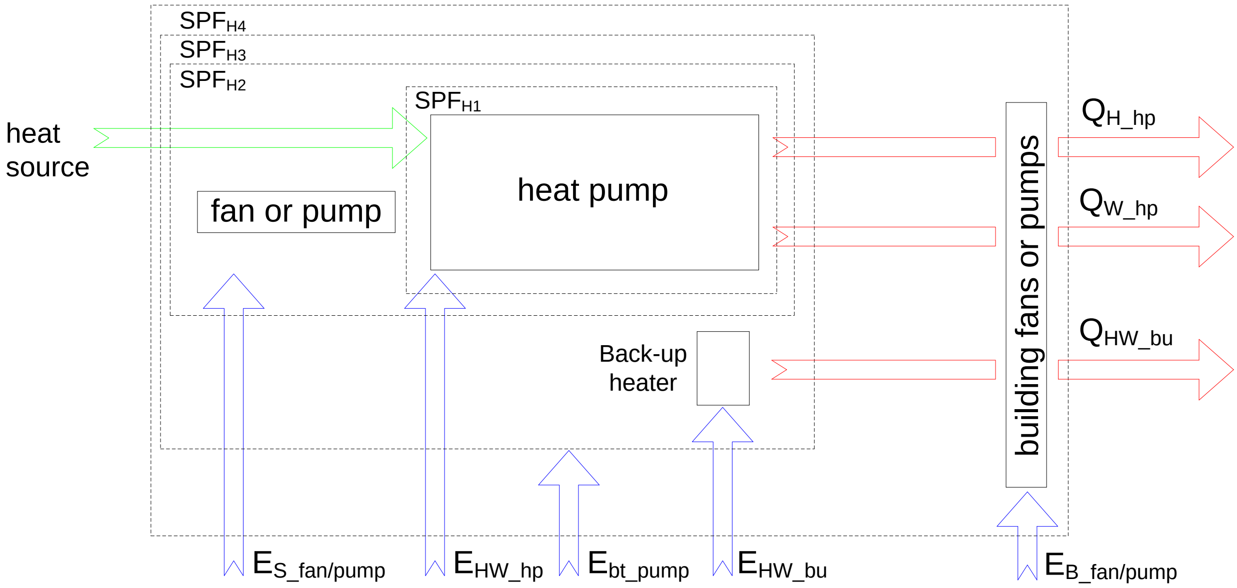
During the field measurements the conditions under a whole cycle was measured for internal use. But due to problems to have equivalent measurement conditions at all test sites it was decided to not include this information in the report.

4 Minimum required measured parameters in field measurements

The SPF-value can be calculated for different levels of the heating system. The level is described by defined system boundaries. This project relates to four different system boundaries developed in the SEPEMO EU project. The system boundaries are more detailed described in section 8.4.

The system boundaries are named SPF_1-SPF_4, each number describing its own system boundary. Different system boundaries mean different requirements of data to be measured. Before performing field measurements it must be clear what SPF level that is to be measured.

The figure below shows the different system boundaries developed in SEPEMO. SPF_1 includes SPF for the heat pump itself *only*. SPF_2 also includes heat source pumps and fans, the equipment to make the heat source available for the heat pump. SPF_3 also includes auxiliary heating, back up heating. SPF_4 includes heat sink equipment like fans or liquid pumps, to make the heat available in the house.



The required measurements differ between different types of heat pumps. The required measurements related to each type of heat pump is shown in Table 7.

Table 7. Minimum results for different heat pump types.

		A/W	DX/W	B/W	W/W	A/A
Electric energy input - total	kWh	x	x	x	x	x
Electric energy input backup heater	kWh	x	x	x	x	x
Electric energy input pumps/fans heat source side	kWh	x	x	x	x	
Electric energy input pumps/fans heat sink side	kWh	x	x	x	x	
Energy output heating / cooling	kWh	x	x	x	x	x
Energy output DHW	kWh	x	x	x	x	optional
SPF according the system boundaries	-	x	x	x	x	x
Average supply temperature heat sink*	°C	x	x	x	x	x
Average return temperature heat sink*	°C	x	x	x	x	x
Average supply temperature DHW*	°C	x	x	x	x	Optional
Average return temperature DHW*	°C	x	x	x	x	Optional
Average supply temperature heat source*, 1	°C			x	x	
Average return temperature heat source*, 1	°C			x	x	
Average outdoor temperature*	°C	x	x	x	x	x
Average indoor temperature*	°C	x	x	x	x	x
Outdoor humidity	%	x				x

*During heating season (operating season). 1Ground temperature should me measured in direct expansion systems

The performance of air to air heat pumps is measured according to SP method 1721. This method is more detailed explained in section 5.2. The boundary condition that is used in this method differs from the boundaries stated in the figure above. This method includes separate measurements of the auxiliary heater and the total electrical input to the heat pump, the fans in the indoor and outdoor unit included. For an air to air heat pump the auxiliary heating is not a part of the heat pump system, but a part of the building that is to be heated. The energy used for auxiliary heating should be measured in order to be able to calculate the energy cover ratio from the heat pump. The DHW production is also outside the heat pump system regarding air to air heat pumps. These parameters are optional to measure, but are interesting for information purposes.

4.1 Minimum results for the different SPF levels

The minimum result from the measurements according to each SPF level is stated in Table 8 below. Some parameters are necessary to measure in order to get data for the SPF equations, while some parameters are necessary to measure in order to understand the operating conditions for the heat pump and to be able to read and compare the results from different systems. The energy output can be measured either by using an energy meter or by measuring the supply and return temperatures together with the liquid flow.

Table 8. Required measurements for meeting the SPF levels according to SEPEMO.

		SPF_{H1}	SPF_{H2}	SPF_{H3}	SPF_{H4}
electric energy input heat sink auxiliary	kWh	-	-	-	x
electric energy input backup heater	kWh	-	-	x	x
electric energy input heat source auxiliary	kWh	-	x	x	x
electric energy input - total	kWh	x	x	x	x
energy output heat	kWh	x	x	x	x
energy output DHW	kWh	x	x	x	x
supply temperature (heat sink)	°C	x	x	x	x
return temperature (heat sink)	°C	x	x	x	x
supply temperature (heat source)	°C	x	x	x	x
return temperature (heat source)	°C	x	x	x	x
outdoor temperature	°C	x	x	x	x
outdoor humidity	%	x	x	x	x
indoor temperature	°C	x	x	x	x

4.2 Additional measurements

There are also parameters that can be measured that are not necessary for the calculation of SPF, but can be usable for other purposes, for example in an energy balance over the heat pump system or for information purposes. The storage losses of the storage tank can also be calculated by using extra measurements. Examples of extra measuring points are displayed in Table 9 below.

Table 9. Optional measurements

		SPF_{H1}	SPF_{H2}	SPF_{H3}	SPF_{H4}
energy output heat source	kWh	x	x	x	x
energy output into DHW storage	kWh	x	x	x	x
pressure difference, heat source	Pa	x	x	x	x
pressure difference, heat sink	Pa	x	x	x	x

4.3 Data acquisition system

The data must be recorded with a system that interfaces the sensors to a data acquisition system that can handle the necessary number of inputs from the entire sensor set.

5 Studied methods for field measurement

The relevant methods for field measurements that are studied in this project are three Nordtest methods (NT VVS) and one SP method:

- Large heat pumps - Field testing and presentation of performance (NT-VVS076)
- Refrigeration and heat pump equipment - General conditions regarding field testing and presentation of performance (NT-VVS115)
- Refrigeration and heat pump equipment - Check-ups and performance data inferred from measurements in the refrigerant system (NT-VVS116)
- Prestandaprovning av luft/luft värmepumpar i fält (SP metod nr 1721)

5.1 NT VVS methods

The NT VVS methods intend to cover the need of capacity- and functional controls and measurements for heat pumps in field applications in four different levels.
The methods states recommendations of how the measurements of temperature, flowrates, pressures and pressures differences shall be performed. In appendix estimations of measured uncertainties are given for all measured quantities with examples. The stated uncertainties for measurement given are:

- Level 1 < 5% capacity measurement
- Level 2 < 10% capacity measurement
- Level 3 < 15% capacity control

Table 10. Example of maximum permissible deviation from the mean value. Taken from the NT VVS 115-method.

Temperature, flowrate	maximum permissible deviation from the mean value (±)	
	Level 1	Level 2 and 3
Temperature of heat transfer medium, cold side	0.5 K	1 K
Flowrate of heat transfer medium, cold side	5%	10%
Temperature of heat transfer medium, hot side	1 K	2 K
Flowrate of heat transfer medium, hot side	5%	10%

The system boundaries are specified in each method. The measurements can either be carried out for the single heat pump or for the larger system, the plant.

Method NT-VVS 076 recommends that operating conditions are those for which the heat pump performance data has been guaranteed. NT-VVS 115 and NT-VVS 116 do not have recommendations. The thermal power output is decided by measuring the flow rate and temperature rise of the hot side heat transfer medium. Thermal power input is determined by measuring the flow rate and the temperature drop of the cold side heat transfer medium. Heat meters can be used. In method 116 also refrigeration condensing and evaporating pressures and temperatures are measured.

If possible the plant/ heat pump must have operated under stable conditions, within the limits of stated maximum deviations, for at least 30 minutes before the measurements starts. The measurement period is at least 30 minutes and readings are taken at a maximum interval of 3 minutes.

If the heat pump operates during defrost conditions the measurements shall be carried out with defrosted heat exchanger surfaces, during the most stable 30 minute period possible. The performance test in NT-VVS 115 and NT-VVS 116 is carried out when the heat pump has attained regular frosting-defrosting sequence starting at least 10 minutes after a terminated defrost cycle. In method NT-VVS 076, the defrosting function is checked concerning its influence on heat pump performance during one complete frosting- and defrosting cycle.

Measuring instruments must have a certificate of calibration traceable to a national or international primary standard that is not older than 1 year at the moment of testing.

Equations for calculating COP and SPF are given. The SPF equations include also any supplementary heating and states that standby losses must be concerned.

5.2 SP method nr 1721

SP method nr 1721 is a field measurement method for field testing of electrically driven air to air heat pumps in heating or cooling mode. The method includes heating capacity, electric power input and coefficient of performance. Instructions of how the measurements shall be performed are stated. If the test is conducted in accordance with the measuring requirements of the method, the coefficient of performance can be determined with an uncertainty of measurement lower than 10%. The method is validated in a combination of laboratory and field measurements.

The system boundaries are specified in the method. The measurements can either be carried out for the single heat pump, the heat pump system or for the entire heat system, see Figure 5.

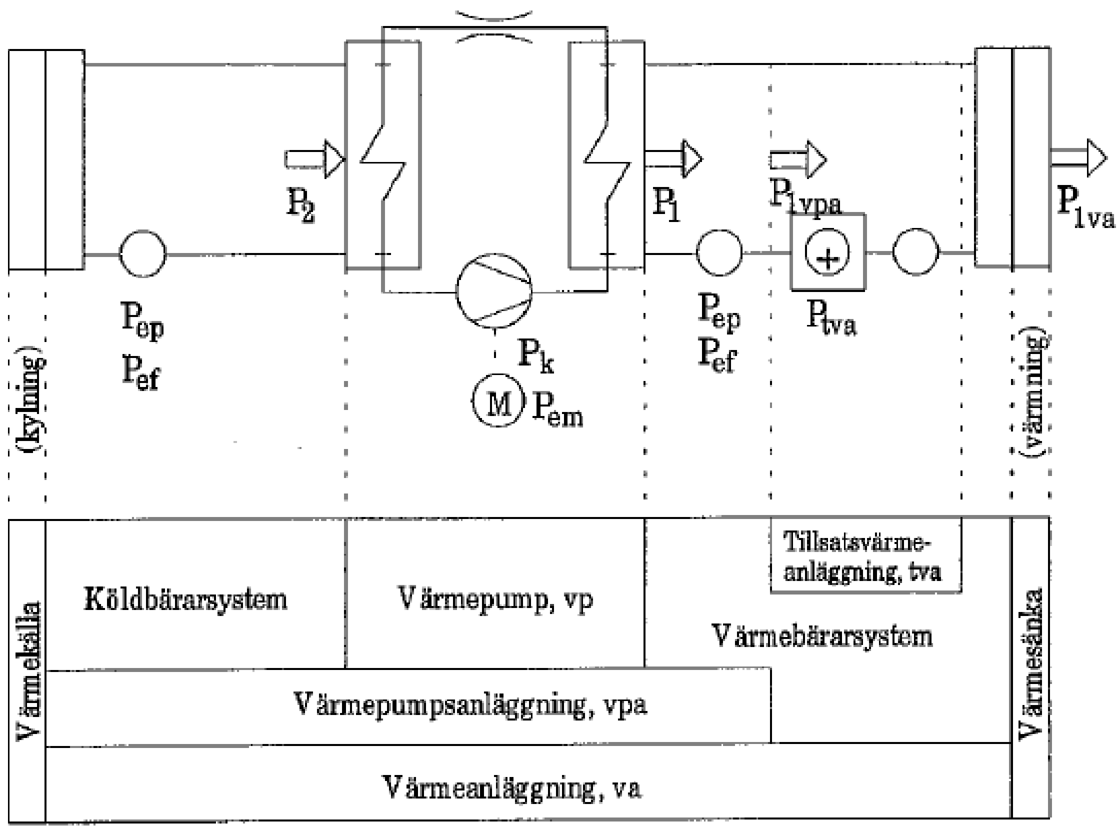


Figure 5. The figure shows the boundaries of the system for measuring the heat factors.

No examples or recommendations of operating points for the tests are stated in this method.

The total electricity consumption during the test is measured by attaching an electrical power meter or an integrated electrical energy meter to the supply cable of the heat pump.

The emitted heat effect is decided by measurements in the circulation flow. A volume- or a mass flow meter (installed according to the manufacturer's instructions) is used to measure the air flows in the heat transfer medium circuit. To minimize effects at the air flow, the meter is not allowed to affect the static pressure at the outflow of the heat pump more than ±3Pa. Therefore it is often necessary to include an extra fan.

The temperatures that shall be measured are: incoming cooling medium temperature, incoming heating medium and leaving heat transfer medium temperature. The temperature of the incoming cooling medium is measured by one sensor placed in the centre of the air intake. The temperature of the incoming heating medium is measured by at least four temperature sensors evenly spread over the air intake. The variation between the highest and lowest temperature indication shall be lower than 1 K. The temperature of the leaving heat transfer medium circuit is measured by at least four sensors evenly spread out at a point where the air is mixed. The mixing device is not allowed to affect the static pressure of the outflow of the heat pump more than ±3Pa, whereupon it is often necessary to include an extra fan. Heat exchange between the mixing device and the surroundings shall be taken into account. The variation between the highest and lowest temperature indication shall be lower than 1 K.

The data collection starts when "the plant" has operated at least five minutes at steady state conditions, within the required permissible deviations, see Table 11. The stability is controlled by continuous measuring at intervals shorter than 1/5 of the stability period, maximum one minute interval.

Table 11. Required permissible deviations for data collection in SP Method 1721.

Temperature, flow	Maximum permissible deviations from mean value
t_{vbin}	± 1K
t_{vbut}	± 1 K
q_{vvb}, q_{mvb}	± 5%

The sampling period shall be at least 10 minutes and the collection of data shall be either continuous register or measuring by intervals more frequent than 1/5 of the measuring period (<2min). The operation shall be stable also during the measurement period.

When the heat pump operates during conditions where frosting occurs, the capacity test is performed after a defrost period at the most stable 10-minutes period possible (at least five minutes after the defrost period).

6 Studied methods for calculation of SPF

The matrix below (Table 12) is a summary of the most important standards studied in the project. It is divided into different categories trying to sort out the content of the different standards. All AHRI standards mentioned above refers to ASHRAE standard 37 for the description the test method and requirements for testing. The purpose of the AHRI standards is to provide test and rating requirements, requirements for operating and the like for different kinds of heat pumps. The standards EN 255-3, prEN 255-3, TS14825 and prEN14825 all refers to the standard EN 14511 for requirements to fulfil the test method. For data input to the calculations of the calculation method EuP Lot 10 and to some extent EuP Lot 1 and EN15316-4-2, one is referred to the test results from standard EN 14511.

The first category "type of standard" shows whether the standard describes a test method for laboratory tests, for field tests and if it includes a calculation model for the calculation of seasonal performance factor.

The second category "type of heat pump" describes what kind of heat pumps that is included in the standard or test method.
The third category "Operation" describes the type of operation that is treated by the standard. The different types of operation can be heating mode, cooling mode or production of domestic hot water. The column called "combined operating" refers to the simultaneous production of heating and/or cooling and the production of domestic hot water. The last column within this category "part load conditions" shows if the standard includes the operation of the heat pump in part load.

The intention of the fourth category "requirements" is to show whether the standard has any requirements of testing to reach accurate test results. Typical requirements could be that steady state has to be reached before the measurements are performed, requirements of maximum deviations from the stated measurements and a largest permissible uncertainty of measurements of the tests. The last column within this category shows whether the standard gives any recommendations of how the measurements shall be performed, such as the placement of sensors.

Table 12. Matrix of existing methods for testing and measurement and evaluation of SPF for heat pumps.

	Type of standard			Type of heat pump			Operation					Requirements				Aspects in capacity calculations					Calculations of	
	Laboratory tests	Field tests	Calculation model for SPF	ASHP	GSHP/WSHP*	AIR/AIR	Heating	Cooling	Domestic hot water	Combined operating	Part load conditions	Steady state	Permissible deviations	Uncertainty of measurements	Test set up/ performance of measurement	Pumps and fans included	Defrost period	Standby losses	On/off cycles capacity regulation	Other	COP/EER	SPF/SEER
NT VVS 076				x	x	x	x	x			x	x	x	x	x						x	x
NT VVS 115				x	x	x	x	x			x	x	x	x	x						x	x
NT VVS 116				x	x	x	x	x				Δ	Δ	x	x						x	
SP 1721						x	x	x				x	x	x	x						x	
ASHRAE standard 37	x			x	x	x	x	x				x	x	x	x		x			x		
AHRI 210/240	x					x	x	x			x	x	x	Δ	x				x		x	x
AHRI 870-2005	x				x		x	x			x	Δ	Δ	Δ	Δ						x	
AHRI 390-2003	x					x	x	x			x	Δ	Δ	Δ	Δ		x				x	
AHRI 320-1998	x				x*		x	x			x	Δ	Δ	Δ	Δ						x	
AHRI 325-1998	x				x		x	x			x	Δ	Δ	Δ	Δ	x					x	
AHRI 330-1998	x				x		x	x			x	Δ	Δ	Δ	Δ	x					x	
EN14511	x			x	x	x	x	x				x	x	x	x	x	x					
prEN14511	x			x	x	x	x	x				x	x	x	x	x	x					
EN 255-3	x			x	x				x			x	x	x	x	x	x	x			x	
prEN 255-3	x			x	x				x			Δ	Δ	Δ	Δ	x	x	x			x	
TS14825	x			x	x	x	x	x			x	Δ	Δ	Δ	Δ	x	x		x		x	
prEN14825	x		x	x	x	x	x	x			x	Δ	Δ	Δ	Δ	x	x		x		x	x
EN15316-4-2			x	x	x	x	x		x	x	x	α	α	α	α	α	α	α	x	α		x
EuP Lot 1	x		x	x	x		x				x	x	x	x	x	x	?		x		x	x
EuP Lot 10	x		x			x	x	x		x	x	Δ	Δ	Δ	Δ	x	x		x		x	x

The sign "Δ" means that the standard refers to another standard where the requirements are fulfilled.

The sign "α" means that the method is a calculation method that does not include requirements from a specified test method.

The fifth category "Aspects in capacity calculations" describes aspects that are taken into account in the capacity calculations. It describes whether liquid pumps and fans are included in the effective power absorbed by the unit. The "Defrost period" column describes whether the defrost periods are taken into account when measuring and calculating the capacity of the heat pump. The "standby losses" column means that standby losses are measured and taken into account when calculating the capacity of the heat pump. The NT-VVS 076 and NT-VVS 115 both mention that it is necessary to take standby losses into account when calculating the SPF, but there is no method of how to measure the losses. Both the standards for measuring the production of domestic hot water EN 255-3 and prEN 255-3 states methods of how to measure the standby losses, but the way of taking the standby losses into account when calculating the COP differs a lot between the standards. "On/off cycles and capacity regulation" shows whether the standard treats what kind of capacity regulation that is used by the heat pump. The last column "other" shows whether there are other important aspects apart from the earlier mentioned ones, which are taken into account in the capacity calculations. It shows that for some of the methods mentioned in the standard ASHRAE 37 adjustments of the line loss capacity and duct losses are made.

The last category "calculations of" describes the calculated outcome of the standard. The NT VVS standards provide simple equations of how to calculate SPF without a calculation model.

6.1 Other methods including calculation models

Besides the models mentioned above there are several other standards and models that can be used in order to find an appropriate model to calculate a seasonal performance factor. The ones studied in this project are shortly summarized in this chapter.

EN 15316-2-3 Heating systems in buildings – Method for calculation of system energy requirement and system efficiencies – Part 2-3: Space heating distribution systems
This method calculates the system thermal losses and the auxiliary energy demand of water based distribution system for heating circuits (primary and secondary), as well as the recoverable system thermal losses and the recoverable auxiliary energy. The calculations are related to a design effect and design heat load of the accounted zone (EN 12831). Correction factors are provided for a number of different conditions, these conditions can for example be corrections for the size of the building, for systems without outdoor temperature compensation, efficiency and part load. The method can be applied for any time step (hour, day, month or year).

EN 13790:2008, Energy performance of buildings – Calculation of energy use for space heating and cooling (ISO 13790:2008)
This standard provides a calculation method for the assessment of the annual energy use of buildings. Factors that are taken into account are for example the heat transfer by transmission and ventilation of the building when heated or cooled to constant internal temperature, contribution of internal and solar heat gains to the building energy balance and the annual energy use for heating and cooling.
There are two different main methods that are used by the standard, one where the heat balance is calculated during a sufficiently long time (one month or a season) and dynamic effects of the building are taken into account by an empirically determined gain and/or

loss utilization factor and one method where the heat balance is calculated over small time steps (typically one hour) and the heat stored in, and released from, the mass of the building is taken into account.

EN 12831 Heating systems in buildings – Method for calculation of the design heat load
This standard is used to calculate the design heat losses of a heated space; the result is then used to determine the design heat load at standard design conditions. The temperature distribution (air and design temperature) is assumed to be uniform. The climatic data that is used for the calculations are the external design temperature and the annual mean external temperature.

Factors taken into account are for example size of the building, type of building, activities inside the building, type of room, interior, building envelope and ventilation.

A number of standards/methods for the calculation of seasonal performance factor are investigated. Some of the methods only contain a calculation model while some of them also contain instructions of how to test the heat pumps. The calculation models that are studied in this project are prEN14825:2009 draft Nov 09, EN 15316-4-2:2008, EUP LOT 1 and EUP Lot 10.

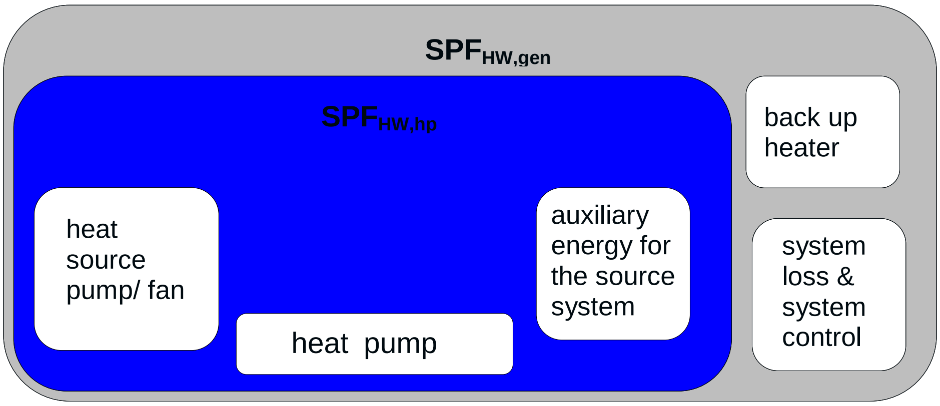
6.2 EN 15316-4-2:2008

Heating systems in buildings – method for calculation of system energy requirements and system efficiencies – Part 4-2: space heating generation systems, heat pump systems

15316-4-2 is a calculation model for the calculation of system energy requirements and system efficiencies. Input product data for the calculations, like heating capacity and COP are determined according to European or national test standards. The method treats calculations for space heating, production of sanitary hot water and combined operation of space heating and sanitary hot water production in either simultaneous or alternating operation. Presently there is no European standard for testing DHW production and space heating simultaneously; therefore a national standard shall be used instead. As an example in this standard calculations based on testing of a DHW cycle performed according to EN 255-3 during heating operation are done, see Annex D in EN 15316-4-2:2008.

<u>System boundaries</u>
The method takes into account different physical factors that can have impact on the SPF and required energy input. For example type of generator, type of heat pump, variation of heat source and sink temperature, effects of compressor working in part load (on-off, stepwise, variable speed units), and system thermal losses.

Losses due to ON/OFF cycling are considered small and negligible unless part load testing data or national values are available. If part load data is not available the stand-by auxiliary energy is considered enough for the degradation of COP in part load operation.



<u>Input to the calculations</u>
Two performance calculation methods for the generation subsystem are described corresponding to different applications (simplified or detailed estimation). The differences between the two methods are the required input data; the operating conditions taken into account and the calculation periods.

The simplified method
The considered calculation period is the heating season and the performance data is taken from tabulated values for fixed performance classes of the heat pump. Operating conditions are taken from typology of implementation characteristics, which means that they are not case specific. This method is in particular suitable when limited information of the generation subsystem exists.

The detailed method
This method is a temperature bin method where the specific operating conditions of each individual heat pump can be considered. The bins describe frequency of the outdoor temperature and the calculations are carried out with operating conditions for the heat pump that corresponds to the heat energy requirement of the space at each bin. The operating conditions of the bins are characterized by an operating point in the centre of each bin and in the calculations it is assumed that this point represents the operating conditions of the whole bin. The standard contains one example of climate; it represents the climate of Gelterkinden in Switzerland and span from -11°C-35°C with a resolution of one bin per K. Appendix A in EN 15316-4-2:2008 shows how to calculate bins using meteorological data for the actual spot. There are examples in the standard that uses only four bins, but with lower resolution, see figure 4 in EN 15316-4-2:2008. There are some criteria when choosing the bin resolution. The bins has to be evenly spread out over the operating range, operating points should be chosen at, or close to test points and the number of bins shall reflect the changes in heat source and sink temperatures. COP values and heat capacity can be interpolated from tested values to fit the bins.

The heat energy requirement of the distribution subsystem can be evaluated if the heat load for space heating and domestic hot water is known. The heat load for space heating is calculated based on cumulated heating degree hours which are defined by the difference between the outdoor air temperature and the indoor design temperature at the different bins. Analogously the DHW load depicted as constant daily profile can be cumulated.

Back up heaters can be accounted for, both for space heating and for sanitary hot water production. If no information about electrical back up heaters is given, an efficiency of 95% is used.

Input data for calculation with the bin method according to chapter 5.3.2 requires indoor design temperature, heat energy requirement of the space heating distribution subsystem according to EN 15316-2-3, type and controller setting of the heat emission system heat pump characteristics for heating capacity and COP according to test standards, results for part load operation according to prEN 14825, system configuration like back up heater calculated according to 15316-4-1 and installed heating buffer storage, power of auxiliary components (pumps etc.). It also requires input data for the DHW-production for example heat energy requirement of the distribution subsystem according to EN 15316-3-2 etc.

Output from the model

Two different seasonal performance factors can be calculated by using this model.

$SPF_{HW, gen}$ is the total seasonal performance factor of the generation subsystem. It includes the heat pump in space heating mode and production of sanitary hot water, the backup heater, the space heating distribution system and auxiliary energy.

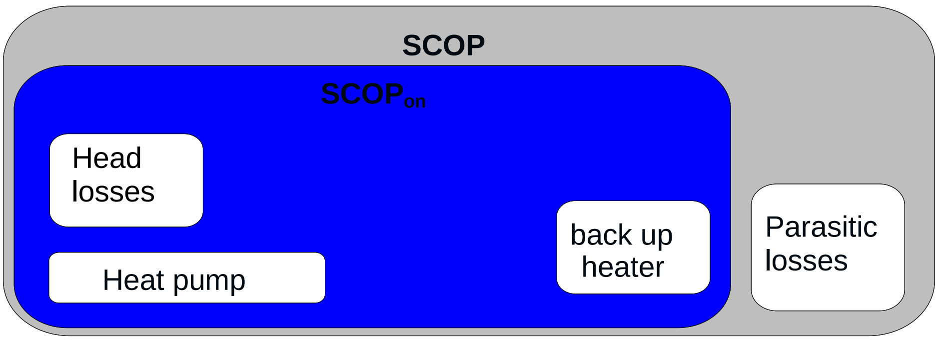
$SPF_{HW, hp}$ is the seasonal performance factor of the heat pump with regard to the heat produced by the heat pump. It includes the heat pump in space heating mode and production of sanitary hot water, the auxiliary energy input for the source system and the auxiliary energy for the heat pump in standby mode.

6.3 Ecodesign LOT 10

LOT 10 applies to “residential room conditioning” appliances (air conditioners and ventilation) with cooling power ≤12kW. It describes a calculation model for calculating the seasonal energy efficiency for operating in heating or cooling mode. This model will probably be replaced by prEN14825 within shortly.

System limits

The model can be used to calculate the seasonal performance factor for an air/air heat pump. The model does not include any losses from the house. To complete the heat demand of the building a backup heater with COP that equals to 1 is accounted for.



Input to the calculations

To use the calculation model provided by the excel sheet the load profile of the building, P*designh* has to be selected. There are nine different sizes to chose between from size 3XS to XXL that spans from 1.1 kW to 19.2 kW. The function of the heat pump is set to either “heat only” or “heat and cool” and the type of heat pump is set to “split” or “multi-split”.

The model is a bin method with three different climates for the heating season and one for the cooling season. A table declares the number of bin hours occurring at each bin temperature, T_j, for each specific climate. The lowest temperature for each climate respectively is declared the design temperature, T_{design}. The part load ratio (of the building), pl_j, is calculated from the equation below:

$$Pl_j = \frac{(T_j - 16)}{(T_{design} - 16)}$$

The reference annual heating demand, Q_{HE} is decided in kWh for each climate as a product of P*designh* and the number of full load heating hours that corresponds to each climate.
Load fractions fracA, fracC and fracW indicate the fraction of the total heating demand (load) occurring in a specific bin at a specific climate. The fractions are given by:

$$frac_j = \frac{n_j * pl_j}{\sum_{j=1}^{40} n_j * pl_j}$$

Input to the calculations is the COP and capacity of the heat pump at four-five different temperature levels +12°C, +7°C, +2°C, -7°C and -15°C(-15°C is only required for the colder climate). The heat pump should be tested at part load to deliver the required heat load of the building at each temperature level. At this point the paper version is not consistent. In one way it says that the capacity of the heat pump at each bin shall complete the energy demand of the building at the part load declared by the product of the annual reference heating demand, Q_{He}, and $frac_j$, but in one way it says that the energy demand is declared by the product of the part load ratio, Pl_j, and P*design.* However the excel sheet uses the first alternative and therefore care should be taken when deciding the operating points (the required effect at each temperature bin) for testing the heat pump. This alternative does not provide any effect balances. Since one house is chosen for the calculations the required effect at each outdoor temperature should be the same among the climates, but this is not the case.

In cases where the heating power supplied by the heat pump is not enough to cover the energy demand of the building in a specific bin, the difference is filled up by a backup heater with a declared capacity of COP=1. Deciding the part load from the product of Q_{He}, and $frac_j$, might result in an underestimated effect demand and therefore underestimate the required backup heating.

Instructions of how the heat pump shall be tested are given in the method for each type of operation respectively; fixed capacity units, staged capacity units and variable speed capacity units.

A degradation factor Cd, which is the efficiency loss per kW of output power when cycling the heat pump, is decided from a specific cycling test.

The energy consumption for the heat pump when operating in thermostat off mode, off mode and crankcase heater mode is decided in tests, but is only required for the calculation of SCOP.

The turndown ratio for heating, which is the lowest steady state over the maximum power and the binlimit, which is the lowest operating temperature of the heat pump, is used as input to both of the SCOP calculations.

<u>Output from the model</u>
This model is used to calculate two different seasonal performance factors:

COP_{ON} is a seasonal performance factor for the heat pump that includes electricity of the backup heater. COP_{ON} is calculated by the total electricity used by the heat pump and the backup heater over the total heat demand of the building.
(LhpC_tp*COPC_tp+resC_tp)/LhsysC_tp

SCOP is a seasonal performance factor which unlike COP_{ON}, also includes the electricity consumption of auxiliary energy for the heat pump operating in thermostat off mode, off mode and crankcase heater mode.

The energy of the backup heater is included in all seasonal performance factors that results from the excel-calculation sheet.

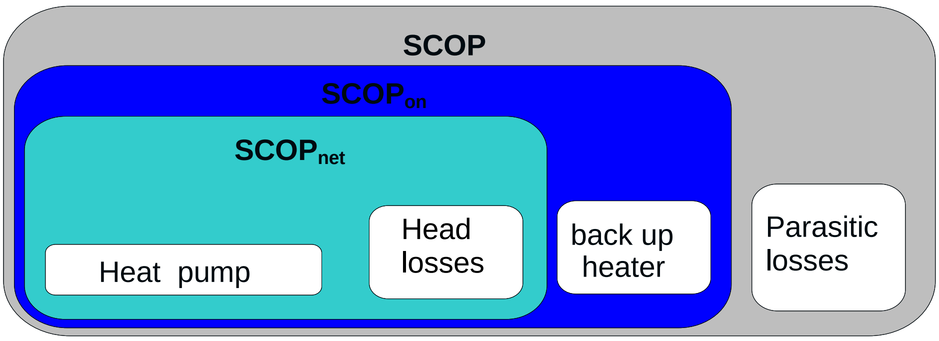
The annual electricity consumption split up in supplementary heating, heat pump operation and auxiliary heating is given from the calculations.

The annual carbon emission and label energy class is also result of the calculations.

6.4 PrEN14825

This is a standard under development that aims to cover the laboratory testing and a calculation model for SPF calculations for electric driven heat pumps. The heat pumps are tested at a number of different part load conditions (4-6) designed for heating or cooling the house to a set temperature of 16°C at different outdoor temperatures. Different test conditions are given for each type of heat pump.
This standard serves as an input for the calculation of the system energy efficiency in heating mode of specific heat pump systems in buildings, as stipulated in the standard EN15316-4-2:2008.

System limits
The model can be used to calculate the seasonal performance factor for air/air- ground source- and air source- heat pumps. The model does not include any losses from the house. To complete the heat demand of the building a backup heater with COP that equals to 1 is accounted for. The system boundary in SPF 4 applies. (Data is treated according to EN14511 where the effect of heat sink pumps and ventilation fans is corrected to overcome the pressure differences of the heat pump.)



Input to the calculations
The calculation of the seasonal performance (SPF or SEER) is performed using a temperature bin method where each bin represents one degree Celsius and the number of bin hours occurring at the corresponding temperature is given. The cooling season is represented by one climate that span from 17°C-40°C while the heating season is represented by three different climates: one colder, one average and one warmer, that

span from -30°C-15°C, see Table 29 and 30 in prEN 14825:2009 draft Nov 09. Each climate corresponds to one design temperature and one design heat load of the building.

The heating/cooling demand and the number of bin hours for the different climates are determined as templates, taking different aspects into account; the climate, type of building and building characteristics, set point and set back settings and internal gains. Those aspects also decide the number of hours in which the heat pump works in active mode, thermostat off mode, standby mode, crankcase heater mode or off mode. The electricity consumptions at the different modes are determined from tests. These effects are called the parasitic losses.

Input to the calculations is the COP and capacity of the heat pump tested at four-five different temperature levels +12°C, +7°C, +2°C, -7°C and -15°C (-15°C is only required for the colder climate). The heat pump shall be tested in equivalence with standard EN 14511:2007, with the same test methods, test set up, uncertainty of measurements and the way of evaluating data. The heat pump shall be tested at part load to deliver the required heat load of the building at each temperature level. Instructions of how the heat pump shall be tested by means of part load and type of operation; fixed capacity units, staged capacity units and variable speed capacity units, are given in this method.
The required part load for the building at the test points are given by:

$$Part\ load\ ratio = \frac{(T_j - 16)}{(T_{design} - 16)}$$

Where T_j is the outdoor (bin) temperature and T_{design} is the lower temperature limit of the selected climate.

If the declared capacities of a unit matching with the required heating/ cooling demand the corresponding COP/EER value is to be used. This may occur with staged capacity or variable speed capacity units. If the declared capacity is higher than the heating/cooling loads, the unit has to cycle on/off. Then a degradation factor (Cd (air/air or Water/air) or Cc (others)) has to be used to calculate the corresponding COP/EER values. Cd and Cc can be determined by testing; else a default value of 0.25 and 0.9 respectively is used.

The bivalent temperature, which is the lowest temperature when the heat pump can deliver 100% of the heat demand of the building, is necessary to use the excel sheet. The design heat demand of the building is a consequence of the stated bivalent temperature. The reference annual heating demand, kWh/a, is given by the product of the full load in heating P_{design} and the equivalent number of heating hours.

The operation limit of the heat pump is set to the lower temperature limit for which the heat pump can operate.

<u>Output from the model</u>
With above input the excel sheet gives two different SCOP: $SCOP_{NET}$ and $SCOP_{ON}$. $SCOP_{NET}$ is the seasonal performance factor for the heat pump, while $SCOP_{ON}$ also includes the electricity and heat delivered to the building from a backup heater.
The paper version of the standard also calculates a seasonal performance factor, SCOP that includes the parasitic losses of the heat pump. The effect from each operational mode is tested according to the standard while the corresponding operational hours for each mode respectively are found in a reference table.

6.5 EuP LOT 1 - Boiler testing and calculation method

This model is used to calculate the specific seasonal energy efficiency *etas* of a space heating boiler. The model contains possibilities to include several different types of space heating appliances in the efficiency calculations, such as boilers, heat pumps, electricity or solar systems. The types of heat pumps included in the model is air source and ground source heat pumps tested in either floor heating- or in radiator heating mode. The model only applies for space heating.

System limits

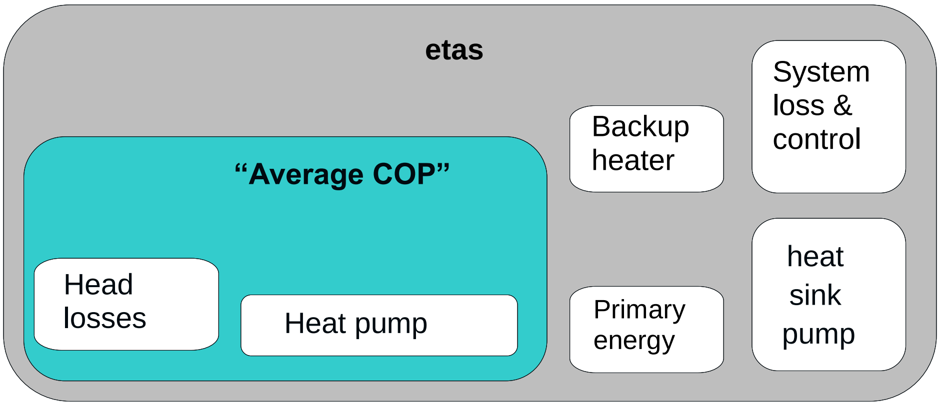
Heat pump data is taken from tests according to EN14511, therefore the head losses from heat source fans or liquid pumps are taken into account in the heat capacity and COP data. This model also includes the heat sink liquid pump.

The model takes into account the net space heating demand, Lh, of the house. The heat demand of the house is a consequence of the choice of the load profile and the so-called system losses Lsys. The size of Lsys depends on the characteristics of the boiler and the installation characteristics. The system losses include fluctuation losses, stratification losses, distribution losses, buffer losses and timer losses, which are set as a percentage that is depending on the heat demand.

The model also includes losses from control, auxiliary equipment and system buffer standing losses.

A back up heater is used to cover up the energy demand that the heat pump cannot deliver.

The electricity use in the model is accounted with the primary energy factor 2.5.



Input to the calculations

The test method for testing the heat pump refers to best testing practice e.g. EN 14511 (see document 7) except for some deviations. The test points are similar to the test points in EN14511:2007, but the temperatures of the return/feed temperature differs, se table IV.2 in the standard. In LOT 1 the temperature difference between T_{return} and T_{feed} gets larger the higher temperature of the T_{feed}. Only three test points are necessary to calculate the seasonal energy efficiency by using this model.

The calculation uses a temperature bin method to evaluate the seasonal energy efficiency, etas. There are three different climates to chose among, warmer (+2°C), average (-10°C) and colder (-22°C), see table I.1, LOT 10. Each bin describes the equivalent number of hours corresponding to the bin temperature with a resolution of one bin/K. Input data to the calculations can be either the test points given in this method or test points given in EN 14511.

The maximum heating capacity, Pmax, at the different climates is calculated from the heating capacity data obtained in the test. It is not possible to choose the size of the required heat load for the building, but is given by the model for each bin level based on the capacity of the heat pump. To meet the lower heat load requirements at the different bin levels, the heat pump is assumed to work in part load condition. The heat pump does not have to be tested in part load operation; instead the model uses a degradation factor, Cd, to calculate the COP when working in part load condition. Cd can either be obtained from tests or a default value, Cd=0.15, can be used.

For fixed capacity units the default is COPmin= 0.89*COP at power output Phpmin=0.5*Php.
For staged capacity units the default is COPmin= 0.975*COP at power output Phpmin=0.5*Php.
For variable capacity units the default is COPmin= COP at power output Phpmin=0.4*Php.

It is optional to choose whether the heat pump operates with night set back or not. The bin assumes constant night temperatures during night set back to +1°C, +6°C and 0°C for each climate respectively.

Other inputs to the calculations is type of heat pump, type of operation of the heat pump, type of control of the heat pump, type of heating (floor heating or radiator heating), minimum source operating temperature, the effect of auxiliary equipment and backup electricity heater.

Other possible energy sources can also be chosen, but this chapter only treats the heat pumps.

Output from the calculations
The model calculates the energy use and losses based upon constant fractions. The fraction of the energy use and the different losses is displayed by the model. A diagram shows the energy supply per temperature bin and how it is covered from different energy sources. The seasonal energy efficiency, etas, is calculated.

$Etas = Lh/Q_{tot}+cctrl$ *where* $Q_{tot}=Lh + L_{sys} + Q_{gen} + Q_{el}$

etas is the net space heating demand of the house over the sum of the generated heat of the system. Q_{tot} is the sum of the space heating demand (Lh), the losses from the heating system (L_{sys}), the primary energy losses of the energy input to the system (Q_{gen}) and the energy needed by the auxiliary equipment such as control and heat sink pumps (Q_{el}).

All electricity used by the heat pump and the backup heater is multiplied by a primary energy factor of 2.5. The model is not transparent. It is tricky to follow the outputs of the model since it consists from several excel-sheets and the information turns up all over. It is also difficult to understand all steps of the calculations. To be able to compare the results with field measurements and prEN14825 a value of SPF, the so called “average COP” (see the system boundaries) is calculated without the system losses. Average COP corresponds to $SCOP_{net}$ in prEN14825.

6.6 SP-method A3 528

SPA3 528 is a calculation program that is used to calculate the seasonal performance factor and energy saving over the year for houses having a defined heating requirement. It can be used for air/air heat pumps, air source heat pumps and ground source heat pumps. The heat loss from the house is defined in the program and given as the total loss factor, k-value, of the house [W/K]. The method can be used to calculate the energy requirement of a building with a k-value of either 109 W/K or 199W/K. A duration diagram of the outdoor temperature can be calculated from the mean annual temperature and together with the loss factor, the area under the duration curve gives the actual power requirement.

The heat pump is tested in accordance to EN 14511 at outdoor temperatures of -15°C, -7°C, +2°C and +7°C with an indoor temperature of +20°C. The heat pump is also tested in part load conditions according to CEN/TS 14825 at +7°C (75% and 50%) and at +2°C (50%). The lowest ambient temperature is assumed to be -15 °C and no heating is assumed to be required for ambient temperatures above +17 °C. The output data from the tests, thermal heat capacity and electrical input power, is used as input to the calculations.

7 Strengths and weaknesses with current methods

7.1 prEN14825

Strenghts

A strength of standard prEN14825 is that it includes all kinds of heat pumps (except exhaust air heat pumps). The model treats heat pumps both in heating and cooling operation. The fact that the heat pump is tested in exactly part load should result in more sufficient results compared to degradation coefficient etc. The model is foreseeable and quite easy to follow.

Weakness

The model is not completely clear with its definitions of part loads. The part load ratio for which the heat pump is to be tested is the part load energy demand of the building at the corresponding temperature bin. To perform the SPF calculations according to prEN14825 the heat pump is tested at a certain climate (A,W or C) and a certain heat load profile for the building. This means that the test data might not be suitable for another climate or another heat load.

It is also not completely transparent since it describes (ANNEX C) the reference heating/cooling demand and the number of hours in each operational mode (active mode, thermostat off mode, standby and crankcase heater mode) is decided from weighted climate, type of building, internal gains, set back setting and so on, but there is no reference that describes the calculations. Therefore it is not possible to recalculate the hours to fit specific needs. The climate hours that describes the temperature bins does not seem to be adjusted in any ways since it is the same hours that is used in Ecodesign LOT 1.

Another weakness is that the model does not include domestic hot water.

Possibility

The model could be developed so that it would be possible to decide the energy demand of the house. It could also be a possibility to fit the model to your own climate. Maybe the ground water temperature and thereby the bore hole temperature could be climate depending.

It should be obvious how interpolations or extrapolations of capacity and/or COP should be performed to avoid differences between users.

Risk

The performance of water/water heat pumps can be overestimated, especially at the cold climate, since they are only tested at +10°C at the cold side (in reality the ground water temperature can be lower than this). This can also be the case for other ground source heat pumps.

The degradation coefficient Cc might be a disadvantage for a ground source heat pump when default values are used. Cc =0.9 is a larger degradation of GSHP’s than what is shown in reality. There is a risk that the requirement of having heat pumps tested in part load might lead to extensive laboratory tests, which is costly. It is also difficult to get sufficient data from existing laboratory tests, since few heat pumps are tested in part loads.

7.2 EN 15316-4-2

Strength

This model is very wide and thorough in its content. It treats both room heating and tap water production. The model is adaptable to different climates and the resolution of the temperature bins can be chosen.

The model specifies the requirements and losses of the certain house and defines recoverable respectively unrecoverable energy.

It is not necessary to test the heat pump at the part loads, since there are default values that can be used.

The model can be used to calculate the SPF for the entire system with the building included or only for the heat pump.

Weakness

The strengths of this model could also turn out to be its weaknesses. The wideness of the model makes it complicated and twisty. There are too many aspects that are taken into account in the calculations. The standard refers to several other standards for calculations of losses and needs. The model requires large knowledge of the house.

The fact that default values can be used to calculate the operation in part load for the heat pumps can result in lower accurateness of the model.

The model does not treat operation in cooling mode.

Possibility

The model can be studied and give input to a new easier model.

Risk

There is a present danger of doing mistakes when using the model. The large amount of data that is taken into account will probably result in much estimation that will differ from case to case and will therefore result in incomparable outcome of the model. Also the same heat pump installation can probably give different results depending on the way it is calculated, (choosing method, input, accuracy and test points).

7.3 EuP LOT 1

In general, the Energy Using Products (E**u**P) Directive have broadened to include also Energy Related Products (E**r**P), but for the treatment in this report, we choose to use the term EuP, since heat pumps are energy using.

Strength

Test data from EN 14511 can be used in the calculations. The model provides default values to recalculate the test points to fit the part load of the heat pump for the different kinds of heat pumps (fixed capacity, staged capacity och variable capacity). The capacity and corresponding COP values are then interpolated between the temperature bins. However, the accurateness of the recalculation is unknown.

The model itself has suggested test points with a radiator curve (supply temperature) that is adjusted to the outdoor temperature. At colder outdoor temperatures the supply temperature is higher and at warmer outdoor temperatures the supply temperatures are lower.

The model can be used to calculate how to cover the energy need of the house by using different techniques, for example solar cells, heat pumps and fossil fuel. This is a good thought, but might not be interesting in this project (?).

Weakness
Unfortunately the model still contains bugs and technical mistakes in the equations and the way of thinking. It seems to be adjusted to boilers and bio boilers instead of heat pumps.

The model does not include a power balance, but is doing a temperature balance instead. This makes the distribution of the energy need and the required amounts of backup heat differ from the theoretical needed.

The model includes a decided fraction of heat loss that cannot be escaped from. For example if the heat pump does not use night set back a default penalty loss of 12% from the total delivered energy is subtracted. The losses from the apparatus and system operation are also decided in percentages.

At part load operation there is no change in the system flows. This does not seem right with controlled radiators. (Should the radiators be controlled or is it enough with a displacement/adjustment of the radiator curve?)

The night set back function uses the same night temperature all year around, which is not the case in reality.

It is not possible to choose the energy requirement of the house; instead the energy demand is an outcome of the capacity of the heat pump. If the heat pump is not monovalent also the fraction of backup heat is needed to decide the energy demand of the house.

GSHP's are treated unfairly when recalculating the operation data to part load operation. The ground source heat pumps are degraded by a factor 0.89 at 50 % of the delivered capacity. (The Cd factor, i.e. the on/off control, is overestimated for water borne systems)

Even though the program is transparent in the sense that all equations are reported in the model, it is very hard to understand and follow the calculations, and the program cannot be said to be transparent in the general sense. The interface of the program is not very friendly and can easily confuse the user. The model does not include tap water.

Possibility
Making the ground water and borehole temperature climate dependent might lead to results more sufficient to its actual installation spot.

Risk
The model is not adjusted to fit heat pumps and is disadvantaging heat pumps. Despite this the COP and capacity of water to water heat pumps can be overestimated since they are tested at +10°C at the cold side (this can also happen to ground source heat pumps, but probably not to the same extent).

7.4 EuP LOT 10

Strength
This model can be used both in heating and cooling mode and it has three different climates both for the cooling season and the heating season. The model has reference heating/cooling demands to choose between.

<u>Weakness</u>

The model takes only air to air heat pumps into account. In accordance to prEN14825 the test points for the heat pump has to be chosen specifically to fit the chosen climate and heat profile of the house.

In accordance to LOT 1 the model does not include an effect balance at each temperature bin. This results in that the heat demand of a house at a specific temperature bin is different at different climates and that the heat requirement of a backup heater is misleading.

The model does not seem to be entirely consistent, partly it is contradicting itself.

<u>Possibility</u>

To make the model usable at other spots it would be better to make it possible to use other climates. Now the model only provides a number of specified heat loads of the house. It would be useful to be able to freely choose the heat demand of the house. There is a risk though, that since the heat pump has to be tested in part load, it has to be tested at each specific heat requirement.

Other types of heat pumps could be included in the model. The model only provides the SPF (SCOP) with the backup heater included. For comparable reasons, it would be useful to include a SPF with backup heater excluded.

<u>Risk</u>

It is not obvious whether the excel model is compatible with the standard. There are also some calculations in the standard that seems to be incorrect.

8 Comparison of existing calculation methods and results from field measurements

8.1 Heat (and cooling-) demand of the house

This study is focused on heat pumps for indoor heating. The study is made in houses with different heat demand. The ground source heat pumps in this study are considered monovalent, but it is difficult to determine the actual energy demand of the house. When using the calculation models the required heat load of the house is decided by the capacity of the heat pump.

The studied air to air heat pump is not monovalent. The energy demand of the house with the heat pump installation was estimated in the field study. When using the calculation models the energy demand of the house were tried to be the same as in the field study.

8.2 Indoor climate

The indoor climate is expected to reach 20°C for all models. In the calculation models the heat pump is used to reach a temperature of 16°C. Internal gains are expected to contribute to the last temperature increase.

The actual indoor temperature has not been measured in the Fraunhofer field measurements. Thereby it is not possible to compare the real indoor temperatures with the temperatures estimated in the calculation models.

8.3 Outdoor climate

The outdoor climate follows the climate of the year. The calculation models use the same temperature climate when calculating SPF for the ground source heat pumps. The climate corresponds to a European average climate, Strasbourg, with the coldest temperature of -10°C.

The field measurements of the ground source heat pumps are carried out in Germany. The heat pumps installations used for the SPF calculations are spread over the country, from the Hamburg area in the north to Stuttgart in the south. In the calculation models the average climate is chosen as the climate mostly corresponding to the German.

The air to air heat pump installation is made in a climate that is similar to the “colder” climate. Therefore the colder climate is used in the calculation models when calculating SPF for the air to air heat pump.

8.4 Definition of SPF field measurement system boundaries

In the ongoing EU project “SEPEMO-Build (SEPEMO short)” four SPF’s with different system boundaries are defined. The definitions from the SEPEMO project have been used for calculating the SPF for the field measurements. The four defined SPF’s are:

SPF_1 includes only the heat pump unit itself. Thereby SPF_1 is identical to the average COP for the measured period.

$$SPF_{H1} = \frac{Q_{H\_hp} + Q_{W\_hp}}{E_{HW\_hp}}$$

SPF_2 consist of the heat pump unit and the equipment needed to make the heat source available the heat pump.

$$SPF_{H2} = \frac{Q_{H\_hp} + Q_{W\_hp}}{E_{S\_fan/pump} + E_{HW\_hp}}$$

SPF_3 represents the heat pump system SPF. SPF_3 includes the heat pump and the heat source pump as in SPF_2, but also the back up heater.

$$SPF_{H3} = \frac{Q_{H\_hp} + Q_{W\_hp} + Q_{HW\_bu}}{E_{S\_fan/pump} + E_{HW\_hp} + E_{HW\_bu}}$$

SPF_4 includes all parts relates to SPF_3, additionally SPF_4 also includes the distribution of the heat.

$$SPF_{H4} = \frac{Q_{H\_hp} + Q_{W\_hp} + Q_{HW\_bu} + Q_{DHW\_bu}}{E_{S\_fan/pump} + E_{HW\_hp} + E_{bt\_pump} + E_{HW\_bu} + E_{B\_fan/pump}}$$

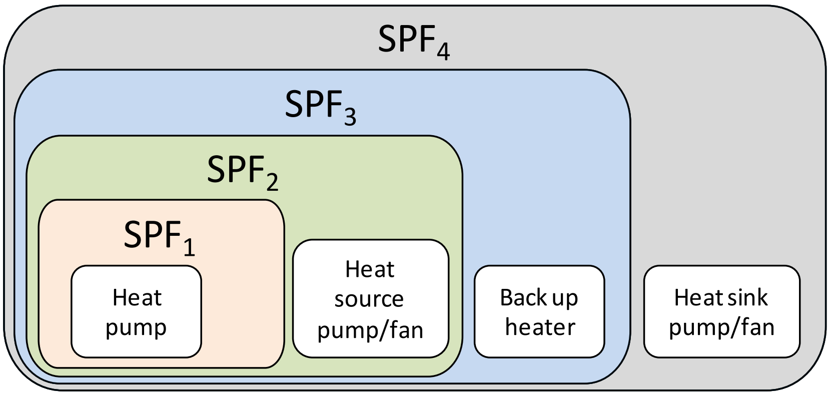


Figure 6 System boundaries for calculations of SPF

SPF_1 is normally measured on the brine/water sides of the evaporator/condenser, but it could also be measured directly in the refrigeration loop with e.g. the Climacheck equipment [10]. This requires measurement of the pressure and temperature of the refrigerant. This methodology is very efficient if the status/condition or diagnosis of the heat pump is to be evaluated, but generally in domestic heat pumps, the measurement is not easy to carry out since measurement sockets are not generally installed.

8.5 Calculation of SPF

In this study SPF is calculated for three of the four different system boundaries and categories by using data from the field measurements. The system boundaries used are SPF_2, SPF_3 and SPF_4 and are described in section 8.4. The categories are “heating only”, “heating and domestic hot water production” and “domestic hot water production”.

For facilities where the installed heat pump also is tested in a laboratory, the laboratory test results are used to calculate SPF by using the calculation models. This is the case for seven different ground source heat pumps and one air to air heat pump.

This chapter will explain how the calculations are performed and what assumptions are made for the different models.

<u>Field measurements</u>

Ground source heat pumps
All analyzed heat pump systems are installed in German single family houses with floor heating. The heat pump is more or less monovalent, only a very small amount of backup heat has been used during the year of measurements. The heat pumps in the study were all installed in new built houses during the years 2004-2008. The data used for the SPF calculations are based on field measurements carried out during one year, with one exception the SPF for site no. 1 is based on data measured from January to August.

The calculations of SPF's are based on the field measurements data from the Fraunhofer study. In the data we have received from the Fraunhofer study the total energy consumption for the heat pump system and its components is presented as well as the energy consumption divided into energy used for space heating and energy used for production of domestic hot water.

In this project we have not been able to evaluate exactly how these allocations have been made. For some of the studied installation sites a part (up to 20%) of the total electricity consumption has been allocated neither to space heating nor to the domestic hot water production. This is mainly the case for the electricity consumption. For the heat produced no energy gap is seen between the total energy production and the energy divided into space heating and domestic hot water.

The calculated SPF's in the study are based on the energy allocated to the space heating only, this in order to make the results comparable to the results from the calculation models in prEN14825 and Lot 1, which not include the production of domestic hot water.

Air to air heat pumps
The field measurement of the air to air heat pumps is carried out in single family houses located in the Borås area of Sweden. All houses in the study have electricity driven radiators for back up heating. The field measurements are based on SP method 1721. From the field measurements SPF_2 and SPF_3 has been calculated as described below.

The electricity consumed by the heat pump, W_{HP}, is measured continually while the produced space heating is measured at five "performance tests" done at different outdoor temperatures. During the performance tests the heating capacity of the heat pump is measured during stable conditions and is thereby not including any defrost period. Therefore the calculated COP for each test point is based on data from only a part of the operating cycle.

The total amount of heat produced during the total measuring period needs to be calculated based on the five performance tests. The calculations are made as follows:

$$Q_{HP\_year} = \sum W_{HP\_month} * COP_{average\_month}$$

COP is calculated from the performance tests at made at different outdoor temperatures. During the performance test both the electricity consumed (W_{HP}) and the produced heat (Q_{HP}) is measured and COP can be calculated as:

$$COP = \frac{Q_{HP\_test}}{W_{HP\_test}}$$

From the five performance tests the COP for the heat pump can be expressed as a function of the outdoor temperature. From this function an average COP for each month is calculated based on the average temperature for the month. Knowing the electricity produced each month by the heat pump the SPF_2 can be calculated:

$$SPF_2 = \frac{Q_{HP\_year}}{W_{HP\_year}} = \frac{\sum W_{HP\_month} * COP_{average\_month}}{W_{HP\_year}}$$

The electricity consumed by the backup heaters is also measured, thereby SPF_3 is calculated as:

$$SPF_3 = \frac{Q_{HP\_year}}{W_{HP\_year} + W_{backup\_year}} = \frac{\sum W_{HP\_month} * COP_{average\_month}}{W_{HP\_year} + W_{backup\_year}}$$

Due to lack of data from laboratory testing of the heat pump models included in the field measurement, this study has only been able to compare the SPF values from the field measurements with the SPF calculated with the calculation models prEN14825 and Lot 10 for one of the tested heat pumps.

prEN14825

Ground source heat pumps
When using prEN14825, data according to Table 13 has to be filled in. The chosen climate, “average” gives that Tdesign is -10°C. Tbivalent is the outdoor temperature where the capacity of heat pump covers the heat demand of the house. It is set to -10°C, to make the heat pump monovalent, like in the field study. TOL, the operation limit temperature, is set to -25°C. This temperature declares where the heat pump no longer can operate. The model calculates Pdesign as a result of Tbivalent and is the heat demand of the house at Tdesign.

Table 13. Input data for the prEN14825 calculation model.

T design	-10	°C
T bivalent	-10	°C
T OL	-25,00	°C
P design	8,81	kW

The test conditions for the heat pumps were taken from Table 20 in the standard, brine to water heat pump, average climate and low temperature application. The unit is assumed to be a fixed capacity unit with fixed outlet temperature. The heat pumps in the study where all tested in full load according to EN 14511. For the part load conditions the COP was calculated by using equation 12 in the standard. The test point used for the calculations was the 30°C/35°C point from EN 14511 laboratory data. The capacity and COP at Tbivalent and TOL is set to the maximum, while the COP for the delivered capacity at the different outdoor temperatures is calculated by using equations from the standard prEN14825. The default degradation factor where Cc=0.9 is used.

Air to air heat pumps
The data for SPF calculations regarding air to air heat pumps are taken from the field measurements. There are no laboratory data available for the heat pumps tested in the field study.

The colder climate is chosen for the calculations, since this climate is similar to the climate where the field installation is. The bivalent operation point of the heat pump is calculated by using SPA3528, which is another model for the calculation of SPF. The bivalent point is 0°C. The operation limit point is set to -20°C.

At -7°C the heat pump operates in full load to deliver heat to the house. At +2°C and at +7°C the heat pump operates in part load. COP for part load operation is interpolated by using linear interpolation between existing test points. At +2°C the interpolation is made between full load operation and operation at 47% part load, at +7°C the interpolation is made between part load operation at 50% and 57% of the heat pump capacity. At +12°C the required heat load is so small that the heat pump is assumed to cycle on/off. The capacity of this point is calculated by using equation 11 in the standard. The COP for the bivalent point is interpolated from test points in full load operation at +2°C and -7°C.

<u>Lot 1</u>
In Lot 1 there are some general inputs that has to be filled in into the excel sheet. The following inputs are used:

- Reduced setback: Yes
- Radiator (with setback): No
- Floor heat (24h): Yes
- Control: 4 – Weather ctrl BT
- Pump: 3 fixed speed
- Pump timer: 24h
- Buffer: No
- Tmino: -25°C

The only heat generator in use is heat pump. No back up heater is included in the calculations.

The default degradation factor, Cd= 0.15, is used. Default is also used for hpaux (=30W) and hpsb (=10W). The test conditions are taken from the reference test conditions in table V.3. in the standard. The test point used for the calculations was the 30°C/35°C point from EN 14511 laboratory data. The model recalculates the test data to fit with the test conditions of Lot 1 (table V.2.).

Data for part load operation is calculated from equations of "option B" at page 27 in the standard, where COP_{min}= 0.89*COP at power output Phpmin=0.5*Php for a fixed capacity unit.

From Lot 1 two different results are obtained, "etas" and "average COP". Etas are calculated by involving the primary energy factor of 2.5 which makes it difficult to compare with other calculated SPF. However, "average COP" corresponds to SPF 1.

<u>Lot 10</u>

Air to air heat pumps
The design load of the house is chosen to 8,5kW, which is the design load that best corresponds to the size of the house in the field measurement. The house in the field is installed in a climate, similar to "colder" climate, therefore "colder" is chosen. The test

points for the calculation are given in a table at page 24 in LOT 10 Annex II. The heat pump is tested according to EN 14511 and CEN/TS 14825 for part load conditions.

The heat pump is a variable capacity heat pump, but since the heat pump is not tested at exactly the required heat effect (within ±3%), the calculations of COP has to be performed in accordance with a staged capacity unit.

At -15°C and -7°C the delivered capacity from the heat pump is lower than the house requires; capacity and COP data are taken from operation in full load at these outdoor temperatures. An exception from the standard is made, since the standard proposes a recalculation of the COP at those points. The recalculation does not seem to make sense and is therefore ignored.

At +2°C and +7°C the heat delivered from the heat pump exceeds the required heat from the house and is therefore operated in part load. COP for part load operation is interpolated by using the equation for staged capacity units at page 26 in the standard. At +2°C the interpolation is made between full load operation and operation at 47% part load, at +7°C the interpolation is made between part load operation at 57% and 44% of the heat pump capacity.

The heat pump is not tested at +12°C. Full load operation at +12°C is extrapolated from test data at +7°C and +2°C. 50% part load is extrapolated from 50% part load operation at +7°C and +2°C. COP for the required effect is extrapolated by using this data. Each extrapolated COP value is corrected with a degradation factor of 0.975.

Default values are used for the degradation factor (Cd=0.1), turndown ratio heating (=25%), thermostat off mode (50W), crankcase heater mode (=10W) and off mode (=10W). The bin limit is set to -20°C.

8.6 Analysis of the results

The results from the SPF calculations of the different heat pump installations in field is compared with the results obtained from the laboratory data used in calculation models.

Ground source heat pumps
Most of the heat pumps installed in field operates both in floor heating mode and produces domestic hot water. The measurements include both kind of operations and the results are presented in Table 14and Figure 7 below. SPF for domestic hot water production is always lower compared to operation in heating mode. The energy balances is not 100% complete for the field measurement, which is quite common in field measurements, since heat losses are present, but cannot be measured directly as they can be in the laboratory.

Table 14 The table shows two different SPF from the field measurements in two different levels. SPF for heating and DHW (domestic hot water) is lower than SPF for heating only. This is because COP for domestic hot water production is lower than COP for heating

Results field measurements				
	Heating and DHW	Heating and DHW	Heating only	Heating only
	SPF 1	**SPF 3**	**SPF 1**	**SPF 3**
site 3	3,70	3,46	4,66	
site 6			3,86	3,43
site 8	4,13	3,02	4,71	
site 9	3,97	3,64	4,53	
site 11	3,62	3,32	4,71	4,56
site 13	2,71	2,55	3,99	3,83
site 14	4,14	3,55	5,43	5,16

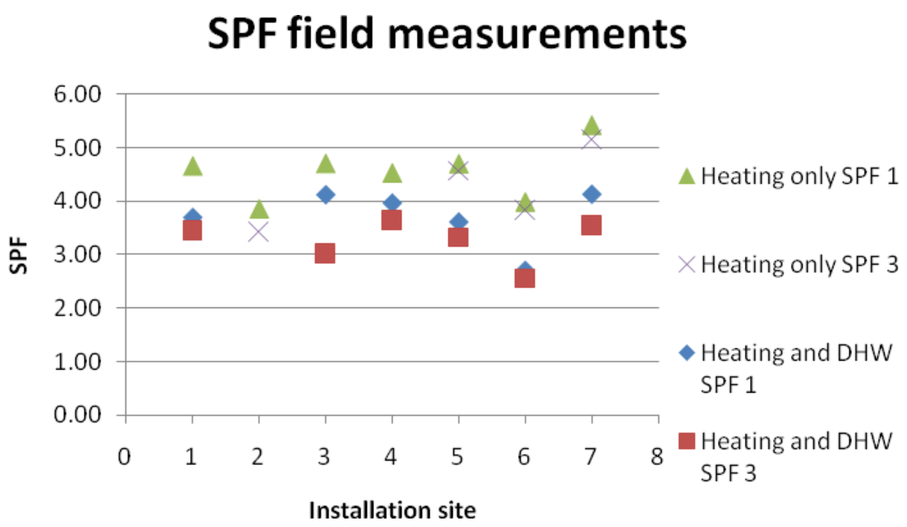


Figure 7 The figure show SPF results from two different SPF, “heat only” and “heat and DHW” (domestic hot water heating) at two different levels, “SPF 1” and “SPF3”, from field testing.

The conditions for measurements in a laboratory and in field differ with respect to various factors e.g. the boundary conditions. SPF_1 in field measurements includes the electrical energy from the heat source brine pump, while “average COP” and “$SCOP_{net}$” only includes the head losses. This could make the electrical energy use a little larger for the field measurements, but on the other hand “average COP” and “$SCOP_{net}$” also contain head losses for the heat sink side which SPF_1 does not. The electrical energy from the heat sink pump for SPF_1 is included in SPF_3.

Table 15. The table shows the results from using Lot 1. Average COP is comparable with SPF 1 from the field measurements. Pdesign shows the maximum capacity needed for the house

Results Lot 1			
	avg COP	etas	Pdesign
site 3	3,57	1,05	7,7
site 6	3,49	1,03	7,6
site 8	3,49	1,02	5,9
site 9	3,49	1,03	7,6
site 11	3,83	1,12	7,2
site 13	3,88	1,12	5,8
site 14	3,88	1,12	5,8

Table 16. The table shows results from using prEN14825. SCOPnet is comparable with SPF 1 from the field measurements. Pdesign shows the maximum capacity needed for the house.

Results prEN14825			
	SCOPon	SCOPnet	Pdesign
site 3	3,66	3,66	8,81
site 6	3,58	3,58	8,7
site 8	3,6	3,6	7,17
site 9	3,58	3,58	8,7
site 11	3,96	3,96	9,64
site 13	4,02	4,02	8,01
site 14	4,02	4,02	8,01

Since the ground source heat pumps in this study is considered monovalent, the comparison of the results are mainly done for SPF_1 from the field measurements and SPF that corresponds to SPF_1 from the calculation models, “average COP” from Lot 1 and “SCOPnet” from prEN14825. The results are presented in Figure 8 below.

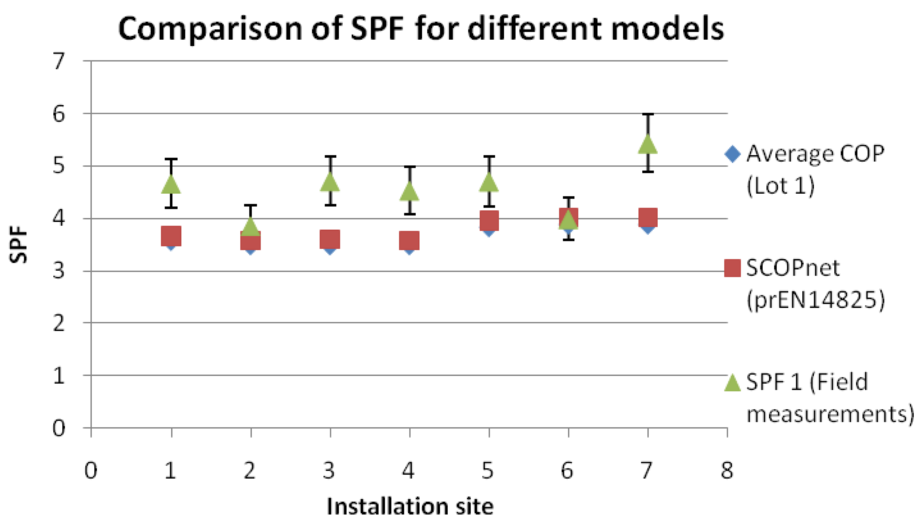


Figure 8 The figure shows a trend that SPF_1 is higher compared to “average COP” and “SCOPnet”. Field measurements imply a higher uncertainty compared to measurements in a laboratory. The bars of error show an error of ±10% to cover the margins of error.

There are two main differences between “average COP” and “$SCOP_{net}$“:

- There are differences in degradation for part load operation
- Lot 1 does not make an capacity balance of the heating demand of the house at each outdoor temperature.

The last factor results in that the design capacity, P_{design}, turns out to be larger for the house when using $SCOP_{net}$ compared to “average COP”. The result show that P_{design} for “average COP” is approximately 13-28% lower compared to “average COP” and SPF is approximately 3-4% lower. The degradation of COP is a little bit tougher when using Lot 1 compared to using prEN14825. The comparison is illustrated in Figure 9 below.

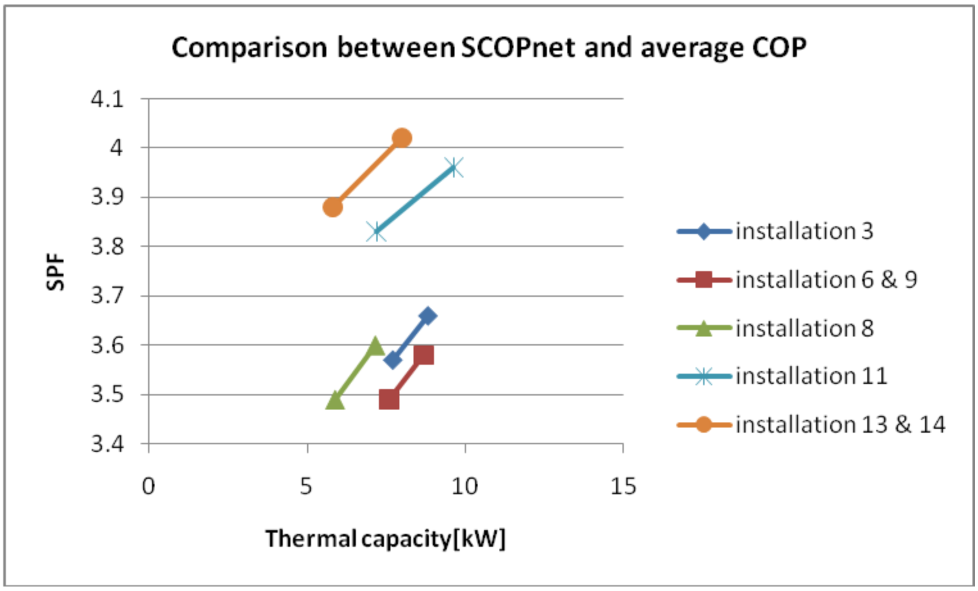


Figure 9. The figure illustrates the differences in design capacity when using EuP Lot 1 and prEN14825. The lower value corresponds to Lot 1 and the higher value corresponds to prEN14825.

Air to air heat pump
Laboratory test data was available for one of the air to air heat pumps that were studied in field. SPF from the field study and SPF from the calculation models are presented in Table 10 below. SCOPnet is the SPF for the heat pump that corresponds to SPF_1. SCOPon is SPF for the heat pump with the backup heater included and corresponds to SPF_2. SCOP is SPF for the heat pump with both backup heater and parasitic losses included. There are some problems by comparing the laboratory test data and the data from field testing, since the field tests do not include the defrosting periods. Therefore SPF from field testing might turn out a little higher than in reality.

Table 17. The table shows a comparison of result for the air to air heat pump.

Results field measurements			
	SCOPnet	**SCOPon**	**SCOP**
prEN 14825	2,52	1,96	
Field measurement	2,4	2,1	
Lot 10		2,15	2,12

8.7 Conclusions from comparisons

Some of the field installations show different SPF_1 despite that the same heat pump model is installed. This can be an indication of how important the sizing of the heat pump is. An oversized heat pump results in for example more part load operation and causes standby losses.

The calculation models show that there can be benefits when installing a heat pump where the bivalent temperature is higher than the lowest operation temperature of the year, even though backup heating is necessary.

9 Requirements for a new calculation model to evaluate SPF from lab measurements

The requirements on a new calculation model differs defpending on the aim of the model. In general, three different uses can be identified:
Based on lab data understand the consequences of technology choice in comparison with competing heating technologies
To understand the consequences of correct sizing of the heat pump
To make a correct dimensioning of the heat pump in a specific house

It should also be possible to study three modes of operation, DHW production, heating or combined DHW production and heating.

Based on the models, it should be possible to make comparisons of e.g. LCC and environmental performance of different systems

What should be included/ not included in the model?

- It should be possible to decide the energy demand of the house in the model, either by given reference loads, or by choosing a specific energy demand of the house. This should be separated into space heating and domestic hot water. When the model itself calculates the losses of the house it can be misleading and not sufficient for the actual house. This can be one boundary of the project. Alternatively, typical houses are used in typical climates, both preset in the model.

- To take into account for the climate at the installation, generally accepted spot climate data, for example Meteonorm data [9], Should be a part of the model.

- The dynamics of the house can be a part of the model. The perceived temperature of the house is not fully consistent with the actual outdoor temperature. At colder temperature dips of for example -15°C, the house will not experience the real outdoor temperature, but experiences a temperature of -12°C instead (due to internal heat gains). Even the irradiance of the sun differs between the seasons (and different spots). The energy demand of the house is affected from those variances over the year, why it might be an idea to calculate the SPF over monthly periods. Also the use of a fictive outdoor temperature would be an alternative. The climate data can be adjusted (flattened out) depending on a number of inputs, but a temperature dip is still needed in order to make a proper effect dimensioning (this is dimensioning the entire system such as deep wells etc.).
  In a serious effort to evaluate dynamics, other factors have to e incorporated in a model, such as form factor, impact of building weight, window area compared to wall area, placement of windows, etc., which make such a model very complex.

- For ground source heat pumps, the temperature of the ground is varying during the year. The model should include a correction for this. This could be expressed as a function where the ground source temperature is a function of the outdoor temperature over the year.

- The model should contain a radiator heat curve where requisite supply temperature is calculated, an example of this can be found in the thesis of Fredrik

Karlsson [8]. At a colder outdoor temperature, the supply temperature should peak; this makes the test scheme tables in EN 14511 deficient. Also other heat distribution systems, such as under floor heating, and mixed systems should be included in the model.

- Part load performance of the heat pump must be properly taken into account, and be based on relevant testing standards.

Night set back is a choice in some calculation models; this is not relevant for heat pumps and should not be a part in a new calculation model.

- Back up heaters is sometimes necessary to complete the energy demand of the house. Back up heaters should be included in the calculation model. Supplementary heating should be possible to choose between different sources of supplementary heat, e.g. electricity, solar or biomass heating.

- The possibility to include the production of domestic hot water to the SPF calculations is also a necessity in future calculation models. It should also be described how this shall be measured in tests alternatively, how the amount of produced domestic hot water shall be estimated. Today there are two main ways how to do the measurements, including the losses or not (one can measure the amount of energy that is obtained by tappings or the amount of tap water the heat pump is producing). A lot of work has already been done in this respect in the IEA HPP Annex 28 [13]. Also, there is a CEN standard on the way on how to treat DHW production. This standard however does not take into consideration combined heating and DHW production.

- Accumulators should be possible to include in the model.

- A model must contain clear system boundaries for what is to be included in the calculations and how measurements are performed. As a basis, the system boundaries presented in the SEPEMO project [12] is recommended.

- The model must be transparent so it is possible to follow and understand the calculations. The studied models all contain parts that are more or less transparent. For example how the estimation of the number of equivalent heating hours is performed is not shown in any method.

An outcome of the results should be to see that a properly sized heat pump is the best alternative to install. An oversized heat pump will result in unnecessary on/off cycling losses and an undersized heat pump will result in unnecessary high back-up heating.

For the calculation, either BIN methods or hour by hour calculations could be used. The existing calculation models based on heat pump performance testing according to standards are all using BIN models. Therefore, to keep a clear connection to existing test standards, it is the easiest to base a new model on BIN models. A hybrid model using chronological BIN's could also be an interesting option to look into.

The drawback with this approach might be that dynamic effects, especially in cases with large or well stratified accumulators are not treated in a way that the full potential of these units are revealed.
In the proposed IEA Annex, a thorough investigation of the positive and negative effects of these approaches should be performed.

10 Conclusions

For a new calculation method to better represent real SPF values there is a need to rely on consistent sets of performance data acquired from lab testing. These lab testings guarantee consistency, repeatability and reliability.

If the objective is to give better values for individual houses, more details on the building envelope, climate data etc. must be provided for the specific setup.

If the objective is to give reliable values for typical conditions, type houses in type climates should be used, but with better details than currently used in existing models.

A new model should include combined DHW and heating to the full extent.

Other key numbers, such as energy performance, energy savings, environmental performance and life cycle cost should be developed in a harmonized way. These key numbers act as a complement to SPF values.

System boundaries should be transparent and comparable with other heating technologies. The use of more than one system boundary allows analyzing parasitic losses from pumps, fans and piping work. The use of different system boundaries also allows to communicate what parts of a heat pump system that working properly or not satisfying in the final installation.

It is important to not only act as a national project in the case of SPF, since much of the activities are on an European or even global level, so the results from this project will be very valuable input to the international work within IEA.

11 Further work

The results from this project will be fed into the IEA Annex on SPF, and further development of a calculation method can be proposed to relevant stakeholders from that Annex.

12 Publications from this project

Within this project, SP have presented results in the form of an article to the Swdish magazine KYLA., “Jämförelse av metoder och fältmätningar för utvärdering av årsvärmefaktor (SPF) “. The article was planned to be published in issue 3, 2010.
An abstract has been submitted to the forthcoming IEA heat pump conference in 2011.

13 References

1. Meeting notes SPF Annex Chicago 2009-01-27
2. Meeting notes Paris: HPP SPF Annex 2010-03-05
3. A common method for testing and rating of residential HP and AC seasonal performance, draft legal text IEA HPP Annex
4. Ekonomisk och driftserfarenhetsmässig utvärdering av bergvärmepumpar, Utarbetad av Per Levin, Projektengagemang Energi och klimatanalys Danderyd, februari, 2008
5. Two large field-teste of new heat pumps in Germany, Miara, M. 9th IEA heat pump conference, Zyrich, 2008
6. Årsmätning på fem bergvärmeanläggningar i Sjuhärad, Energimyndigheten, 2005
7. Fältmätning på Luft/Luft värmepumpar i svenska enfamiljshus, SP, 2009, to be published
8. Capacity Control of Residential Heat Pump Heating Systems, Fredrik Karlsson, Building Services Engineering, Department of Energy and Environment, Chalmers University of Technology, Göteborg, Sweden 2007
9. Meteonorm, Global Meteorological Database for Engineers, Planners and Education, http://www.meteonorm.com/pages/en/meteonorm.php
10. Climacheck website, accessed 2010-04-21, http://www.climacheck.com/
11. Prestandaprovning av luft/luft Värmepumpar i fält, SP metod nr 1721, Fahlén, P., Kjellgren, C., SP AP 1994:51, Energiteknik, Borås 1994
12. Seasonal Performance factor and Monitoring for heat pump systems in the building sector (SEPEMO-Build), IEE/08/776/SI2.529222, http://www.sepemo.eu/
13. Final Report IEA HPP Annex 28, Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating, HPP-AN28-1, ISBN: 91-85533-30-0

Appendix 1. References for field measurements, presented in RIS-format.

TY - BOOK
A1 - Abrahamsson, Thore
T1 - Skövdebadet - solenergiuppvärmning? : alternativstudie avseende uppvärmning genom värmepump, solfångare och värmelager i utebassäng
AU - Jonson, Sten
AU - Norin, Fredrik
Y1 - 1979
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningen, 0346-5616 ; 1979:71
SN - 91-540-3035-8
ER -

TY - BOOK
A1 - Andersson, Per-Åke
T1 - SIMSYS : simuleringsprogram för värmecentraler med ny energiteknik
AU - Askling, Åke
AU - Dalenbäck, Jan-Olof
Y1 - 1986
CY - Göteborg
T3 - Intern skrift / Chalmers tekniska högskola, Avdelningen för installationsteknik, 99-0354505-3 ; 24
T3 - Rapport / Chalmers tekniska högskola, Avdelningen för installationsteknik, 0280-2899 ; 1986:2
ER -

TY - BOOK
A1 - Backman, Anders
T1 - Värmeåtervinning ur avloppsvatten med värmepump för 400 lägenheter i Falun : projektering
Y1 - 1983
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1983:103
SN - 91-540-3991-6
ER -

TY - BOOK
A1 - Björk, Erik
T1 - Praktisk provning av vattenburet värmesystem med värmepump och konvektorer/radiatorer
AU - Wiklund, Sören
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:131
SN - 91-540-3356-X
ER -

TY - BOOK
A1 - Bokalders, Varis

T1 - Energisnåla hus : [30 hus med energisnåla lösningar: solfångare, värmeåtervinning, växthus, braskamin, värmepump, passiva solhus, värmelager]
Y1 - 1981
KW - Uppvärmning (byggnader)
KW - Energieffektiva byggnader
CY - Västerås
PB - Ica bokförl.
SN - 91-534-0568-4
ER -

TY - BOOK
A1 - Boklund, Tord
T1 - Värmepump för luft-vattensystem med mekanisk vinds- och krypgrundsventilation som värmekälla : enplansbyggnad
Y1 - 1983
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1983:53
SN - 91-540-3940-1
ER -

TY - BOOK
A1 - Buresten, Rune
T1 - Solvärme med värmepump som komplement till en oljeeldad värmecentral i Göteborg
AU - Gunnarsson, Ingemar
Y1 - 1983
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1983:26
SN - 91-540-3891-X
ER -

TY - BOOK
A1 - Claesson, Staffan
T1 - Lågenergihus i Högås : hydrologiska förutsättningar för värmepump med en mindre tjärn som värmemagasin
AU - Sandén, Kent
Y1 - 1976
N1 - Examensarbete
T3 - Examensarbete / Institutionen för vattenbyggnad, Chalmers tekniska högskola, 99-0211330-3 ; 1976:4
ER -

TY - BOOK
T1 - Energi ur grundvatten : inventering av stora grundvattenmagasin för energiutvinning med värmepump : slutrapport mars 1980
Y1 - 1980
ER -

TY - BOOK
A1 - Eriksson, Sven-Olof
T1 - Stockholmsprojektet : energibehov i flerbostadshus med solvägg, tung stomme och värmepump : kv Konsolen
Y1 - 1993
KW - Flerbostadshus

KW - Energieffektiva byggnader
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - R / Byggforskningsrådet, 1103-6346 ; 1993:44
SN - 91-540-5590-3
ER -

TY - BOOK
A1 - Eriksson, Sven-Olof
T1 - Stockholmsprojektet, kv Konsolen : energiteknisk utvärdering av flerbostadshus med solvägg, tung stomme och värmepump
Y1 - 1993
N1 - Lic.-avh. Stockholm : Tekn. högsk.
CY - [Stockholm]
T3 - Arbetsrapport / Institutionen för byggnadsteknik, Kungliga Tekniska högskolan, 0349-6562 ; 1993:1
ER -

TY - BOOK
A1 - Faarinen, Jukka
T1 - PoD anläggning av öppet absorptionssystem.
Y1 - 1993
N1 - Examensarbete
CY - Luleå
T3 - Examensarbete / Tekniska högskolan i Luleå, Ingenjörslinjen 80 poäng, 1102-4070 ; 1993:22I
ER -

TY - BOOK
A1 - Hammarsten, Patrik
T1 - Experimentell studie av R22/R142b som arbetsmedium i en värmepump.
Y1 - 1992
N1 - Examensarbete
CY - Göteborg
ER -

TY - BOOK
A1 - Hill, Anders
T1 - Återvinning av värme med värmepump från lokaler där spillvärme alstras
AU - Matsson, Lars Olof
AU - Ryberg, Anders
Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1981:145
SN - 91-540-3628-3
ER -

TY - BOOK
A1 - Holmgren, Mats
T1 - Användning av värmepump för återvinning av värme vid godssvalning
AU - Stigmarker, Håkan
Y1 - 1984
CY - Jönköping

PB - Svenska gjuteriföreningen
ER -

TY - BOOK
A1 - Hult, Johan
T1 - Jämförelse mellan konventionell gaspanna och värmepump med inbyggd gaspanna
AU - Jonasson, Jens
Y1 - 2000
CY - Malmö
PB - SGC
T3 - Rapport SGC, 1102-7371 ; 113
ER -

TY - BOOK
A1 - Jeal, David
T1 - Erfarenheter av luft-vatten värmepump med djurvärme som energikälla : uppvärmning av personalutrymme vid Ökna lantbruksskola
AU - Norrfors, Mats
Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1981:137
SN - 91-540-3612-7
ER -

TY - BOOK
A1 - Jelbring, Hans
T1 - Prov med värmepump och sjöförlagd värmeväxlare vid Kungliga slottet
Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1981:43
SN - 91-540-3498-1
ER -

TY - BOOK
A1 - Jensen, Lars
T1 - Uteluft och mark som värmekälla för värmepump i kv Bobinen i Malmö : utvärdering
AU - Wetterstad, Lennart
Y1 - 1985
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1985:89
SN - 91-540-4430-8
ER -

TY - BOOK
A1 - Johansson, Lars
T1 - Energibehov vid torkning : jämförelse mellan mekanisk värmepump och öppet absorptionssystem för torkskåp inom hushåll : CFD-analys avabsorbator
Y1 - 1998
CY - Luleå

T3 - Examensarbete / Luleå tekniska universitet, Civilingenjörsprogrammet, 1402-1617 ; 1998:020
ER -

TY - BOOK
A1 - Johansson, Mats
T1 - Datorprogram för studium av temperatur- och frysningsförlopp i mark kring kylrör till värmepump
AU - Westman, Kristina
Y1 - 1979
CY - Lund
T3 - Rapport BKL, 0281-6318 ; 1979:9
ER -

TY - BOOK
A1 - Kahsay, Samson
T1 - Processintegration av en kompressions-/absorptions-värmepump (KAVP) vid Värö bruk
AU - Wikensten, Bo
Y1 - 1999
N1 - Examensarbete
T3 - Examensarbete / Chalmers tekniska högskola, Institutionen för värmeteknik och maskinlära, 99-1473418-9 ; 99:4
ER -

TY - BOOK
A1 - Karlsson, Hanna
T1 - Utredning om ökade fjärrleveranser från Shell Raffinaderi AB genom installation av värmepump
Y1 - 2002
CY - Göteborg
PB - Chalmers tekniska högsk.
T3 - Examensarbete (Institutionen för kemiteknik och miljövetenskap/Värmeteknik och maskinlära, Chalmers tekniska högskola); 02:12
ER -

TY - BOOK
A1 - Lagerkvist, Knut-Olof
T1 - Mätning och utvärdering av värmepump med avloppsvatten som värmekälla i Falun
Y1 - 1987
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1987:42
SN - 91-540-4726-9
ER -

TY - BOOK
A1 - Landé, Johan
T1 - Praktiska prov med olika köldmedier [i] brine-vatten värmepump
Y1 - 1992
CY - Stockholm
PB - Institutionen för mekanisk värmeteori och kylteknik, KTH
T3 - Trita-REFR, 1102-0245 ; 92:8
ER -

TY - BOOK
A1 - Ljungqvist, Jarl
T1 - Värmepump med frånluft som värmekälla för tappvatten i befintligt flerbostadshus i Göteborg
AU - Ohlsson, Kjell
Y1 - 1982
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:126
SN - 91-540-3818-9
ER -

TY - BOOK
A1 - Mogensen, Palne
T1 - Fullskaleförsök med berg som värmekälla för värmepump i Järfälla : mätning och utvärdering
Y1 - 1985
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1985:123
SN - 91-540-4470-7
ER -

TY - BOOK
A1 - Mårtensson, Alf-Göran Mårtensson
T1 - Redovisning av försök beträffande värmeåtervinning från djurstall med värmepump placerad i bostadshuset samt nedsmutsningens inverkan på lamellbatteriers funktion.
Y1 - 1983
CY - Lund
T3 - Intern stencil / Sveriges lantbruksuniversitet, Institutionen för lantbrukets byggnadsteknik (LBT) och konsulentavdelningen/teknik, 99-0412863-4 ; 6
ER -

TY - BOOK
A1 - Nilsson, Christer
T1 - Torkning av spannmål med hjälp av värmepump och balktork
Y1 - 1990
CY - Lund
PB - LBT
T3 - Sammanställt / Sveriges lantbruksuniversitet, Institutionen för lantbrukets byggnadsteknik (LBT), 1101-5845 ; 3
ER -

TY - BOOK
A1 - Nordström, Hans
T1 - Mätning, modellbygge, simulering och reglering tillämpat på en anläggning med värmepump
AU - Svensson, Lennart
Y1 - 1977
CY - Lund
ER -

TY - BOOK
A1 - Nordström, Hans

T1 - Mätning, modellbygge, simulering och reglering tillämpat på en anläggning med värmepump
AU - Svensson, Lennart
Y1 - 1977
CY - Lund
T3 - Rapport BKL, 0281-6318 ; 1977:9
ER -

TY - BOOK
A1 - Norén, Bjarne
T1 - Studie av kombinationen gasturbin med värmeväxlare och värmepump
Y1 - 1981
CY - Lund
T3 - Technical report from the Division of Heat and Power Engineering, 0280-9931 ; 5106
ER -

TY - CHAP
A1 - Norgren, Mårten
T1 - Värmepump i Bodens reningsverk betalar sig på tre år
Y1 - 1982
JF - Kommun-aktuellt
VL - 1982: nr 8, s. 15
PB - Kommun-aktuellt
SN - 0347-5484
KW - Uppvärmning (byggnader)-- Sverige -- Norrbotten
KW - Värmepumpar-- Sverige -- Norrbotten
KW - Boden
ER -

TY - BOOK
A1 - Nyman, Bertil
T1 - Värmebalans i småhus med kombinerat uppvärmningssystem - solfångare, värmepump, värmeväxlare samt tidsstyrd värme och ventilation : [licentiatavhandling]
Y1 - 1982
CY - Stockholm
PB - KTH
ER -

TY - BOOK
T1 - Oskadliggörande av läckande CFC-köldmedier från värmepump- och kylanläggningar : förstudie
AU - Jacobson, Sten Olle
AU - Keck, Karl-Erik
AU - Jacobson, Lars
Y1 - 1988
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1988:71
SN - 91-540-4932-6
ER -

TY - BOOK
A1 - Persson, Sven-Erik
T1 - Värmepump med ismaskin för 43 lägenheter i Sälen : förstudie

Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1981:49
SN - 91-540-3479-5
ER -

TY - BOOK
A1 - Sjöbris, Karin
T1 - Värmeåtervinnings- och rökgasreningsanläggning : kondensation av rökgaser i kombination med värmepump : förstudie
Y1 - 1987
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1987:118
SN - 91-540-4836-2
ER -

TY - BOOK
A1 - Stock, Christina
T1 - Kriterier för dimensionering av bergborrande brunnar för energiuttag med värmepump.
Y1 - 1984
T3 - Examensarbete / Institutionen för vattenbyggnad, Kungl. Tekniska högskolan, 99-0535463-8 ; 291
ER -

TY - BOOK
T1 - Stora värmepumpsboken : nyttig kunskap, vägledning och inspiration när du funderar på att köpa värmepump
Y1 - 2009
CY - Arvika
PB - Thermia Värme AB
SN - 91-631-7643-2
ER -

TY - BOOK
A1 - Svensson, Torbjörn
T1 - Värmeåtervinning ur avloppsvatten med värmepump för fjärrvärmeanslutning i Enköping : förstudie
AU - Wahlberg, Herje
AU - Wahlberg, Olof
Y1 - 1983
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1983:58
SN - 91-540-3913-4
ER -

TY - BOOK
A1 - Svärd, Ann-Christine
T1 - Bestämning av värmefaktorn för en värmepump vid Östra Sjukhuset, Göteborg
Y1 - 1979
N1 - Examensarbete
CY - Göteborg

T3 - Examensarbete / Chalmers tekniska högskola, Avdelningen för installationsteknik, 99-0354504-5 ; 4
ER -

TY - BOOK
A1 - Tengblad, Niklas
T1 - Resultat från prov med liten luft-luft värmepump med propan/gasol som köldmedium
Y1 - 1993
CY - Stockholm
PB - Avdelningen för tillämpad termodynamik och kylteknik, Institutionen för energiteknik, Kungliga Tekniska högskolan
T3 - Trita-REFR, 1102-0245 ; 93:12
ER -

TY - BOOK
A1 - Jagenäs, Arne
T1 - Mätning av ljudnivå från värmepump.
Y1 - 1981
T3 - Rapport / Chalmers tekniska högskola, Avdelningen för byggnadsakustik. S, 0282-3446 ; 81:03
ER -

TY - BOOK
A1 - Jansson, Lennart
T1 - Värmepump med uteluft som värmekälla : projektering av värmepumpsystem för befintligt flerbostadshus i Söderköping
AU - Strindehag, Ove
Y1 - 1982
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:124
SN - 91-540-3814-6
ER -

TY - BOOK
A1 - Jilar, Torbjörn
T1 - Fjärås centrum - solvärmesystem med värmepump : utvärderingsresultat
Y1 - 1987
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1987:11
SN - 91-540-4692-0
ER -

TY - BOOK
A1 - Jilar, Torbjörn
T1 - Solvärmeteknik i stor skala : Ingelstad - en värmecentral utan värmepump
Y1 - 1984
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1984:103
SN - 91-540-4171-6
ER -

TY - BOOK

A1 - Johansson, Jan
T1 - Naturvärmesystem för biltestningsanläggning i Slagnäs : ground heat storage sytem for the car test plant at Slagnäs
AU - Josephsson, Börje
Y1 - 1995
N1 - Examensarbete
CY - Luleå
T3 - Examensarbete / Tekniska högskolan i Luleå, 0349-6023 ; 1995:139E
ER -

TY - BOOK
A1 - Jonsson, Eric
T1 - Fullskaleprov med solfångare i kombination med ytjordvärmesystem för uppvärmning av småhus : [ett energisnålt alternativ för uppvärmning av småhus] : [beskrivning av ett kombinationssystem] : [utvärdering och rapport]
Y1 - 1983
CY - [Eskilstuna]
PB - [Eskilstuna kommun]
ER -

TY - BOOK
A1 - Jönsson, Sven-Erik
T1 - Värmepump för spillvärme till fjärrvärmenät : teknisk och ekonomisk utredning gällande spillvärme från Domsjö sulfitfabrik till Örnsköldsviks fjärrvärmenät : förstudie
AU - Sandberg, Edvard
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:72
SN - 91-540-3278-4
ER -

TY - BOOK
A1 - Kangro, Andrus
T1 - Värmeåtervinning med värmepump i djurstallar : mätning och utvärdering av anläggning i Klovarp
Y1 - 1982
CY - Lund
T3 - Specialmeddelande / Sveriges lantbruksuniversitet, Institutionen för lantbrukets byggnadsteknik (LBT), 0348-0593 ; 115
SN - 91-576-1167-X
ER -

TY - BOOK
A1 - Larsen, Johan
T1 - Uppvärmningsalternativ med värmepump för industri
AU - Usinovski, Dime
Y1 - 2003
CY - Göteborg
PB - Chalmers Lindholmen högsk.
ER -

TY - BOOK
A1 - Lemmeke, Leif

T1 - Renvatten som värmekälla till värmepump : en studie i
Åkarp
AU - Persson, Ulf
AU - Luterkort, Staffan
Y1 - 1984
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1984:75
SN - 91-540-4125-2
ER -

TY - BOOK
A1 - Lindahl, Anders
T1 - Värmepump för utvinning av havsvattenvärme : förstudie
beträffande havsvattenvärmeväxlare
AU - Stenström, Börje
AU - Öst, Staffan
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:123
SN - 91-540-3337-3
ER -

TY - BOOK
A1 - Nilsson, Emma
T1 - Småhus med solceller och värmepump : utvärdering av
Ekosolkonceptet
Y1 - 2008
CY - Göteborg
PB - Chalmers tekniska högskola
ER -

TY - BOOK
A1 - Stråhlin, Fredrik
T1 - Optimering av värmeoproduktion : en studie vid Birka
Energi AB
Y1 - 1999
N1 - Examensarbete
CY - Luleå
T3 - Examensarbete / Luleå tekniska universitet,
Civilingenjörsprogrammet, 1402-1617 ; 1999:281
UR - http://epubl.luth.se/1402-1617/1999/281
ER -

TY - BOOK
A1 - Westerlund, Lars
T1 - Öppna absorptionssystem : absorbatorers egenskaper
Y1 - 1989
N1 - Examensarbete
CY - Luleå
T3 - Teknisk rapport / Tekniska högskolan i Luleå, 0349-3571
; 1989:17T
ER -

TY - BOOK
T1 - Årsmötet 1975. D. 2, Krångede och Gammelänge
kraftstationer - ökning av turbineffekten

Y1 - 1975
CY - Stockholm
PB - Svenska kraftverksfören.
ER -

TY - BOOK
A1 - Collin, Mats
T1 - Ångmotordriven värmepumpsprocess för bostadsbebyggelse : förstudie och jämförelse med dieselmotordriven värmepump
AU - Palmgren, Mikael
Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1981:133
SN - 91-540-3604-6
ER -

TY - BOOK
A1 - Dahlberg, Marcus
T1 - Marknadspotential för en kemisk kyl/värmepump
Y1 - 2002
CY - Stockholm
T3 - Trita-IEO. EX, 1403-7777 ; 2002:26
ER -

TY - BOOK
A1 - Franck, Per-Åke
T1 - Värmepump med vertikalt jordvärmesystem och vindkonvektorer
AU - Modin, Björn
Y1 - 1981
CY - Göteborg
T3 - Rapport / Jordvärmegruppen, Chalmers tekniska högskola, 0349-9723 ; 2
ER -

TY - BOOK
A1 - Luterkort, Staffan
T1 - Gasmotordriven värmepump för fjärrvärmenätet i Oxie : förstudie
AU - Lemmeke, Leif
AU - Torisson, Tord
Y1 - 1984
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1984:128
SN - 91-540-4237-2
ER -

TY - BOOK
A1 - Mårtensson, Alf-Göran
T1 - Energibesparing med värmepump - stalluft som värmekälla
Y1 - 1978
KW - Värmepumpar
KW - Stallbyggnader
KW - Energisparande
CY - Uppsala
PB - Sveriges lantbruksuniv.

SN - 91-7088-931-7
ER -

TY - BOOK
A1 - Saue, Olav Jarle S.
T1 - Solinfångande bassäng, pulserande värmelager och värmepump : diskussion av driftstrategier
Y1 - 1982
CY - Lund
T3 - Technical report from the Division of Heat and Power Engineering, 0280-9931 ; 5118
ER -

TY - BOOK
T1 - Skarpnäcks trädgårdsstadsförening 80 år : från kakelugn till värmepump : 1927-2007
AU - Barkman, Clas
KW - Skarpnäcks trädgårdsstadsförening
KW - Stockholm-- historia -- Skarpnäck -- 1900-talet -- sekelskiftet 2000
CY - Johannneshov
PB - Skarpnäcks trädgårdsstadsförening
ER -

TY - BOOK
A1 - Sönsteröd, Gunnar
T1 - Värmepump och oljepanna i bivalent drift : fältmätningar på typfall
Y1 - 1988
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1988:3
SN - 91-540-4846-X
ER -

TY - BOOK
A1 - Agerstrand, Torgny
T1 - Projekt Bergvärme : energiuttag med värmepump ur bergborrade hål : förstudie
AU - Ericsson, Lars O.
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:172
SN - 91-540-3418-3
ER -

TY - BOOK
A1 - Backman, Lennart
T1 - Spillvärme från mejeri till fjärrvärmenät via värmepump : alternativprojektering i Östersund
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:46
SN - 91-540-3228-8
ER -

TY - BOOK
A1 - Berntsson, John
T1 - Konstruktion av slavenhet för styrning av värmepump
AU - Bäckhage, Eric
Y1 - 2006
N1 - Examensarbete
CY - Göteborg
PB - Chalmers tekniska högskola
ER -

TY - BOOK
A1 - Bärring, Mats
T1 - Återvinning av värmeförluster i fjärrvärmekulvertar med värmepump : kulverttyper och systemoptimering
AU - Rosenberg, Mats
Y1 - 1986
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1986:53
SN - 91-540-4571-1
ER -

TY - BOOK
A1 - Cedheim, Lars
T1 - Nedbrytning av glykoler i värmepump och solvärmesystem : förstudie
Y1 - 1986
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1986:4
SN - 91-540-4506-1
ER -

TY - BOOK
A1 - Cedheim, Lars
T1 - Nedbrytning av glykoler i värmepump och solvärmesystem : laboratorieundersökning
AU - Blume, Ulla
AU - Lundgren, Björn
Y1 - 1988
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1988:107
SN - 91-540-4980-6
ER -

TY - BOOK
A1 - Elmroth, Arne
T1 - SPARSAM : fem energisnåla småhus med glasveranda och värmepump
AU - Granberg, Gunnar
Y1 - 1987
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1987:47
SN - 91-540-4736-6
ER -

TY - BOOK
A1 - Grafström, Hans
T1 - Värmepump vid friluftsbad med havsvatten som värmekälla : projektering, uppföljning och utvärdering
AU - Lagergren, Staffan
Y1 - 1982
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:45
SN - 91-540-3692-5
ER -

TY - BOOK
A1 - Hammerbo, Erik
T1 - En femte värmepump vid Rya värmepumpverk, Göteborg
AU - Johansson, Björn
Y1 - 1994
N1 - Examensarbete
T3 - Examensarbete / Chalmers tekniska högskola, Institutionen för installationsteknik, 1400-9552 ; 104
ER -

TY - BOOK
A1 - Hultmark, Göran
T1 - Grundvattenvärme med värmepump vid strandbadet i Karlskoga : utvärdering
Y1 - 1985
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1985:141
SN - 91-540-4479-0
ER -

TY - BOOK
A1 - Jacobsson, Carl
T1 - Värmeutvinning ur havsvik med värmepump för villauppvärmning
AU - Jönsson, Sven
Y1 - 1981
N1 - Examensarbete
CY - Göteborg
T3 - Examensarbete / Institutionen för vattenbyggnad, Chalmers tekniska högskola, 99-0211330-3 ; 1981:2
ER -

TY - BOOK
A1 - Johansson, Anders
T1 - Förvärmning av uteluft till värmepump med markkollektor : datormodell
Y1 - 1990
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1990:68
SN - 91-540-5246-7
ER -

TY - BOOK
A1 - Lindvall, Anna

T1 - Strategier vid val av värmepump till småhus : en energijämförelse mellan varvtalsreglerade och on-/offreglerade värmepumpar
Y1 - 2007
CY - Helsingborg
PB - LTH Ingenjörshögskolan vid Campus Helsingborg, Lunds universitet
T3 - Examensarbete (LTH Ingenjörshögskolan vid Campus Helsingborg), 1651-2197 ; THID-07/5069
ER -

TY - BOOK
A1 - Margen, Peter H.
T1 - Optimering av solvärmecentraler med respektive utan värmepump : VVS 78
Y1 - 1978
CY - Nyköping
PB - Studsvik energiteknik
ER -

TY - BOOK
A1 - Olsson, Ulf
T1 - Dieselmotordriven värmepump för gruppbebyggelse och större fastigheter : förstudie i Fisksätra
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:45
SN - 91-540-3226-1
ER -

TY - BOOK
A1 - Rosengren, Bengt
T1 - Förvärmning med värmepump av dricksvatten till mjölkkor : Preheating of drinking water to dairy cows by means of heat pump
Y1 - 1991
CY - Lund
T3 - Specialmeddelande / Sveriges lantbruksuniversitet, Institutionen för lantbrukets byggnadsteknik (LBT), 0348-0593 ; 179
ER -

TY - BOOK
A1 - Schroeder, Kjell
T1 - Sjövärmeprojekt Motala Väster : utvärdering och analys av värmekälla och värmepump
AU - Svensson, Torbjörn
Y1 - 1989
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1989:58
SN - 91-540-5019-7
ER -

TY - BOOK
T1 - Solenergi och värmepump värmer kontor och industrilokaler.

Y1 - 1983
T3 - SV-rapport, 99-0308735-7 ; 1983:1
ER -

TY - BOOK
A1 - Stillesjö, Staffan
T1 - Värmepump eller direktel för lågenergihus i grupp : förstudie
Y1 - 1984
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1984:187
SN - 91-540-4281-X
ER -

TY - BOOK
T1 - Termokemisk lagring i kombination med dieseldriven värmepump : fullskaleprov i Televerkets arbetslokal i Jakobsberg
AU - Bakken, Kjell
Y1 - 1982
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:102
SN - 91-540-3782-4
ER -

TY - BOOK
T1 - Värmeåtervinning med värmepump från Scaniarinken i Södertälje
AU - Hagstedt, Bertil
Y1 - 1982
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:123
SN - 91-540-3812-X
ER -

TY - BOOK
A1 - Wilén, Peter
T1 - Grundvatten som värmekälla för husuppvärmning med värmepump : litteraturstudie, system och ekonomi
Y1 - 1981
CY - Göteborg
T3 - Rapport / Jordvärmegruppen, Chalmers tekniska högskola, 0349-9723 ; 1
ER -

TY - BOOK
T1 - Energi ur grundvatten : inventering av stora grundvattenmagasin för energiutvinning med värmepump. Kartbilaga till slutrapport : länsvis redovisning av tätorternas och tätortsregionernas energiresurser i stora grundvattenmagasin
Y1 - 1980
ER -

TY - BOOK

A1 - Jacobson, Lars
T1 - Jordvärmesystem med värmepump i befintlig och ny bebyggelse : förprojektering av sju objekt i Västsverige
Y1 - 1982
KW - Geotermisk energi
KW - Värmepumpar
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:112
SN - 91-540-3793-X
ER -

TY - BOOK
A1 - Larsson, Lars I.
T1 - Kraftvärmeverk och värmepump för utnyttjning av rökgasens kondensationsvärme
Y1 - 1980
CY - Lund
T3 - Technical report from the Division of Heat and Power Engineering, 0280-9931 ; 5101
ER -

TY - BOOK
A1 - Löving, Ronald
T1 - Elmotordriven värmepump för luft/vatten i Backatorp, Göteborg
AU - Bäckström, Bernt
Y1 - 1983
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1983:87
SN - 91-540-3988-6
ER -

TY - BOOK
A1 - Möller, Kjell
T1 - Gastät lagring och torkning av spannmål med värmepump : Airtight storage and drying of grain using a heat pump
AU - Nilsson, Christer
Y1 - 1988
CY - Lund
PB - LBT
SN - 91-576-3577-3
ER -

TY - BOOK
A1 - Andersson, Frank
T1 - Fjärås centrum : projektering av solvärmesystem med värmepump
AU - Länsberg, Mats
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:8
SN - 91-540-3170-2
ER -

TY - BOOK

A1 - Areskoug, Mats
T1 - Miljöfysik
Y1 - 1997
N1 - Prel.uppl.
KW - Miljöfysik
CY - Malmö
PB - Lärarhögskolan
ER -

TY - BOOK
A1 - Areskoug, Mats
T1 - Miljöfysik
Y1 - 1998
CY - Malmö
PB - Malmö högskola, Lärarutbildningen
ER -

TY - BOOK
A1 - Axby, Fredrik
T1 - Rökgaskondensering med värmepump = Flue gas condensing with heat pump
AU - Pettersson, Camilla
Y1 - 2004
KW - Fjärrvärme
KW - Värmeteknik
CY - Stockholm
PB - Värmeforsk
T3 - Värmeforsk, 0282-3772 ; 891
ER -

TY - BOOK
A1 - Backman, Anders
T1 - Värmeåtervinning ur avloppsvatten via värmepump : förstudie i Gävle
AU - Hallenberg, Jonas
AU - Bustad, Tohmmy
Y1 - 1983
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1983:44
SN - 91-540-3922-3
ER -

TY - BOOK
A1 - Berendson, Jaak
T1 - Värmepump Motala Väster : korrosionsuppföljning
Y1 - 1991
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1991:50
SN - 91-540-5374-9
ER -

TY - BOOK
A1 - Bergström, Ulf
T1 - Direktverkande värmepump för småhus : utvärdering av funktion och energibesparing genom fältmätningar
AU - Fehrm, Mats

AU - Mattsson, Per Olof
Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:18
SN - 91-540-3641-0
ER -

TY - BOOK
A1 - Bovin, Jan
T1 - Stirlingmotordriven värmepump : förstudie avseende planerat experimentbygge i Kågeröd
Y1 - 1990
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1990:25
SN - 91-540-5180-0
ER -

TY - BOOK
A1 - Bäckström, Bernt
T1 - Värmepump med energistapel : förprojektering av befintligt mindre flerbostadshus i Göteborg
AU - Hallén, Tomas
AU - Samuelsson, Torbjörn
Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1981:140
SN - 91-540-3618-6
ER -

TY - BOOK
A1 - Carlsson, Jan
T1 - Värmepump typ Stirling : examensarbete
AU - Åkersten, Lars-Erik
Y1 - 1981
CY - Göteborg
T3 - Rapport / Institutionen för energiteknik, Ångteknik med reaktorteknologi, Chalmers tekniska högskola. T, 0280-9249 ; 93
ER -

TY - BOOK
A1 - Edman, Jan
T1 - Isbana som värmekälla för värmepump : förstudie i Hörby
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:157
SN - 91-540-3389-6
ER -

TY - BOOK
T1 - Energiutvinning ur ytvatten via värmepump : förprojektering i Borlänge
AU - Backman, Anders
Y1 - 1980

CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:109
SN - 91-540-3326-8
ER -

TY - BOOK
A1 - Enström, Henrik
T1 - Louddens värmepump : utvärdering av avloppsvatten/fjärrvärme-värmepump
AU - Solin, Lars
Y1 - 1987
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1987:16
SN - 91-540-4679-3
ER -

TY - BOOK
A1 - Enström, Henrik
T1 - Uppsala värmepump : utvärdering av avloppsvatten/fjärrvärme, värmepump
AU - Solin, Lars
Y1 - 1988
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1988:97
SN - 91-540-4960-1
ER -

TY - BOOK
A1 - Enström, Henrik
T1 - Visby värmepump : utvärdering
AU - Nilsson, Madelaine
Y1 - 1989
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1989:29
SN - 91-540-5020-0
ER -

TY - BOOK
A1 - Fransson, Kenneth
T1 - Energiåtervinning i animalieproduktionen med värmepump : värmekälla mjölk och stalluft
Y1 - 1982
N1 - Examensarbete
CY - Uppsala
T3 - Rapport / Sveriges lantbruksuniversitet, Institutionen för arbetsmetodik och teknik, 0347-9773 ; 77
SN - 91-576-1314-1
ER -

TY - BOOK
T1 - Från kakelugn till värmepump : en presentation av värmepumpinstallationerna i Forsmarks bruk
Y1 - 1987
CY - Vällingby

PB - Vattenfall
ER -

TY - BOOK
T1 - Från S- till F-system med värmepump : ombyggnad av självdragssystem till mekanisk frånluftsventilation med värmepump : ett praktikfall
AU - Erikson, Bengt E.
Y1 - 1985
CY - Gävle
PB - Statens inst. för byggnadsforskning
T3 - Meddelande / Statens institut för byggnadsforskning, 0347-4348 ; 85:1
SN - 91-540-9219-1
ER -

TY - BOOK
A1 - Grafström, Hans
T1 - Värmepump för uppvärmning av friluftsbad : förstudie i Sölvesborg
AU - Lagergren, Staffan
Y1 - 1980
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1980:44
SN - 91-540-3224-5
ER -

TY - BOOK
A1 - Gustafsson, Gösta
T1 - Avfuktning med värmepump i växthus : Dehumidification with heat pump in greenhouses
AU - Nimmermark, Sven
Y1 - 1991
CY - Lund
T3 - Rapport / Sveriges lantbruksuniversitet, Institutionen för lantbrukets byggnadsteknik (LBT), 0348-0259 ; 74
ER -

TY - BOOK
T1 - Gårdsvärme : ved, halm, torv, värmepump
AU - Liedholm, Hans
AU - Leuchovius, Torbjörn
Y1 - 1983
CY - Stockholm
PB - LT
SN - 91-36-02108-3
ER -

TY - BOOK
A1 - Hallén, Tomas
T1 - Västra klinikerna i Jönköping : grundvatten som värmekälla för värmepump
AU - Edberg, Bo
Y1 - 1985
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1985:88

SN - 91-540-4428-6
ER -

TY - BOOK
A1 - Hallenberg, Jonas
T1 - Avloppsvatten som värmekälla till värmepump : utvärdering av installation i Boden
AU - Wahlberg, Herje
Y1 - 1982
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:125
SN - 91-540-3816-2
ER -

TY - BOOK
A1 - Hallenberg, Jonas
T1 - Sjövatten som värmekälla till värmepump : erfarenheter från uppvärmning av två småhus i Falun
AU - Norbäck, Kjell
Y1 - 1982
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1982:46
SN - 91-540-3694-1
ER -

TY - BOOK
A1 - Henningsson, Lars
T1 - Värmepump : installationsförslag och marknadsundersökning
Y1 - 1982
CY - Luleå
T3 - Examensarbete / Tekniska högskolan i Luleå, 0349-6023 ; 1982:067E
ER -

TY - BOOK
A1 - Henriksson, Jan
T1 - Bygg din egen värmepump
AU - Nilsson, Hans
Y1 - 1983
CY - Ljungby
PB - Colego
ER -

TY - BOOK
A1 - Hill, Anders
T1 - Ammoniak som köldmedie i värmepump
Y1 - 1989
CY - Stockholm :
PB - Statens råd för byggnadsforskning ;
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1989:103
SN - 91-540-5120-7
ER -

TY - BOOK
A1 - Hirschfeldt, Claes-Henrik

T1 - Utvärdering av en öppen kemisk värmepump
AU - Olofsson, Richard
Y1 - 1995
CY - Luleå
T3 - Examensarbete / Tekniska högskolan i Luleå, Ingenjörslinjen 80 poäng, 1102-4070 ; 1995:44I
ER -

TY - BOOK
A1 - Härröd, Peter
T1 - Energibesparing med värmepump i banddiskmaskin
AU - Jorlöv, Andreas
Y1 - 2000
N1 - Examensarbete
CY - Göteborg
PB - Chalmers Lindholmen högsk., Chalmers tekniska högsk.
ER -

TY - BOOK
A1 - Högström, Carina
T1 - Värmepump med propan som köldmedium
Y1 - 1999
N1 - Lic.-avh. Stockholm : Tekn. högsk.
CY - Stockholm
T3 - Trita-REFR, 1102-0245 ; 99:26
ER -

TY - BOOK
A1 - Karlsson, Erik
T1 - Värmepump 1, Findus i Bjuv
AU - Taberman, Marie
Y1 - 2003
CY - Göteborg
PB - Chalmers Lindholmen högsk., Chalmers tekniska högsk.
ER -

TY - BOOK
A1 - Lemmeke, Leif
T1 - Storskalig värmeförsörjning med värmepump : principförslag med kombinerat utnyttjande av yt- och grundvatten som värmekälla
Y1 - 1981
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1981:126
SN - 91-540-3597-X
ER -

TY - BOOK
A1 - Lindelöf, Jan
T1 - Effektiv effektivare värmepump : idéhandbok om värmepumpar
AU - Sandström, Bengt
AU - Sandquist, Hans
Y1 - 1995
CY - Stockholm
PB - Stockholm energi
SN - 91-630-3394-1

ER -

TY - BOOK
A1 - Lundén, Roger
T1 - Värmepump med effektutjämningssystem : en studie av ett system där värmepump kombineras med värmeackumulering i jord
Y1 - 1977
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningen, 0346-5616 ; 1977:85
SN - 91-540-2784-5
ER -

TY - BOOK
A1 - Lundqvist, Per G.
T1 - Stirling värmepump : konstruktion, byggande och prestandaprovning : slutrapport
Y1 - 1996
CY - Stockholm
PB - KTH, Inst för energiteknik
T3 - Trita-REFR, 1102-0245 ; 96:20
ER -

TY - BOOK
A1 - Mårtensson, Hans
T1 - Värmepump i villan
Y1 - 2007
KW - Värmepumpar
CY - Västerås
PB - Ica
SN - 978-91-534-2873-2 (inb.)
ER -

TY - BOOK
T1 - Oljebrännare vs värmepump
AU - Andersson, Ola
Y1 - 1997
N1 - Examensarbete
CY - Göteborg
PB - Ingenjörs- och sjöbefälsskolan vid Chalmers tekniska högsk.
ER -

TY - BOOK
A1 - Persson, Lars
T1 - Pelletseldning eller värmepump? : historik, nutid, framtid
Y1 - 2005
CY - Delsbo
PB - GDE-Net
ER -

TY - BOOK
T1 - Processmodell för värmepump
AU - Abrahamsson, Lars
Y1 - 1986
CY - Luleå

T3 - Teknisk rapport / Tekniska högskolan i Luleå, 0349-3571 ; 1986:012T
ER -

TY - BOOK
A1 - Ransmark, Sven-Erik
T1 - Värmepump med Lorenzprocess
Y1 - 1985
T3 - Institutionen för värme- och kraftteknik, Tekniska högskolan i Lund, 0282-1990 ; 3128
ER -

TY - BOOK
A1 - Renntun, Mats
T1 - Värmepump och värmelagring
Y1 - 1978
CY - Lund
T3 - Technical report from the Division of Heat and Power Engineering, 0280-9931 ; 5078
ER -

TY - BOOK
A1 - Rönnelid, Mats
T1 - Solfångare och värmepump : utvärdering av ett värmesystem i Uppsala
AU - Tepe, Rainer
Y1 - 2004
KW - Solfångare
KW - Solenergi
KW - Geotermisk energi
KW - Uppvärmning (byggnader)
CY - Borlänge
PB - EKOS, Högskolan i Dalarna
ER -

TY - BOOK
A1 - Saros, Georg
T1 - Att välja värmepump : handledning för att välja värmepump och att räkna på ekonomin
AU - Pettersson, Anders
Y1 - 1998
CY - Stockholm
PB - Konsumentverket :
SN - 91-7398-636-4
ER -

TY - BOOK
A1 - Tepe, Rainer
T1 - Solfångare och värmepump : marknadsöversikt och resultat av en preliminär simuleringsstudie av värmesystem med solfångare och bergvärmepump
AU - Rönnelid, Mats
Y1 - 2002
KW - Solfångare
KW - Solenergi
KW - Solenergi
CY - Borlänge
PB - SERC, Högskolan i Dalarna

T3 - Centrum för solenergiforskning, Högskolan Dalarna, 1401-7555 ; 75
ER -

TY - BOOK
A1 - Tepe, Rainer
T1 - Solfångare och värmepump : marknadsöversikt och resultat av en preliminär simuleringsstudie av värmesystem med solfångare och bergvärmepump
AU - Rönnelid, Mats
Y1 - 2002
KW - Solfångare
KW - Solenergi
KW - Geotermisk energi
KW - Uppvärmning (byggnader)
CY - Borlänge
PB - EKOS, Högskolan i Dalarna
T3 - EKOS publikation, 1650-1497 ; 2002:2
ER -

TY - BOOK
T1 - Värmepump : ett tekniskt system i utveckling
Y1 - 1988
CY - Stockholm
PB - Styr. för teknisk utveckling
T3 - STU-information, 0347-8645 ; 671
SN - 91-7850-243-8
ER -

TY - BOOK
T1 - Värmepump för avloppsvatten i Sala.
Y1 - 1982
CY - Vällingby
T3 - SV-rapport, 99-0308735-7 ; 1982:3
ER -

TY - BOOK
T1 - Värmepump för avloppsvatten i Sundsvall.
Y1 - 1982
CY - Vällingby
T3 - SV-rapport, 99-0308735-7 ; 1982:4
ER -

TY - CHAP
A1 - Waldenstad, Bengt
T1 - Testas i Härnösand : Stirlingmotor driver värmepump
AU - Österberg, Roine
Y1 - 1985
JF - VVS & energi
VL - 1985:5, s. 35-36
PB - VVS & energi
SN - 0280-9524
KW - Stirlingmotorer-- Sverige -- Ångermanland
KW - Värmepumpar-- Sverige -- Ångermanland
KW - Härnösand
ER -

TY - BOOK

A1 - Wirén, Mikael
T1 - Avancerad värmepump för enbostadshus :
marknadsförutsättningar i Sverige, Norge, USA och Kanada
Y1 - 1988
CY - Stockholm
PB - Statens råd för byggnadsforskning :
T3 - Rapport / Byggforskningsrådet, 0349-3296 ; 1988:48
SN - 91-540-4891-5
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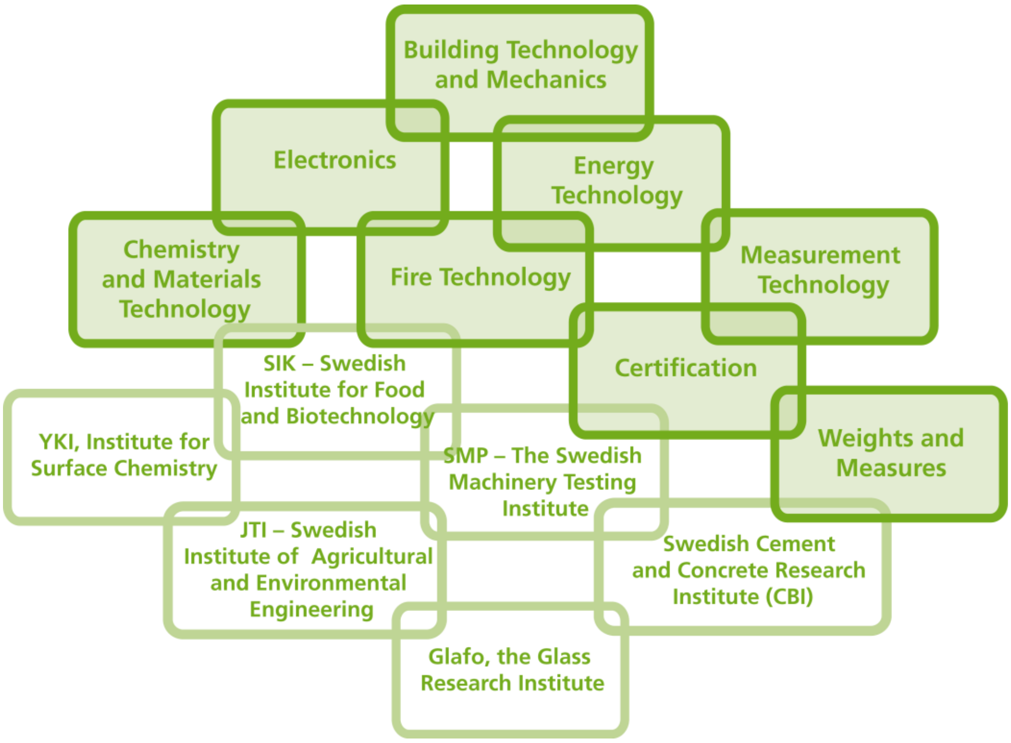

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Appendix 2 – KTHs rapport

KTH Industrial Engineering
and Management

Effsys2-Projekt 9

KTH-unik del:

Metod för beräkning av SPF för värmepumpsbaserade kombisystem uppvärmningssystem

Joachim Claesson, KTH-Energiteknik

Sammanfattning

Projektet har syftat till att identifiera de behov för vidareutveckling och homogenisering av beräkningsalgoritmer för att bestämma värmepumpsbaserade uppvärmningssystems effektivitet per årsbasis i småhus och flerfamiljshus. Enkät har genomförts vilket visar att alla program inte kan hantera mer komplexa uppvärmningssystem som t.ex. solfångare tillsammans med värmepump. Skillnaden på årsvärmefaktorn mellan de fiktiva fastigheterna är inte speciellt besvärande, med beaktande att en leverantör räknade avfrostning dubbelt. Dock pekar det på vikten av riktig och rak kommunikation kring den information som utbytes mellan kund och säljare.

Som andra del i projektet har metodik för beräkning av byggnaders energiprestanda tagits fram med beaktande på de internationella och svenska standarder som finns. Strukturen hänvisar i stor utsträckning till standarder där så tillämpligt och i vissa fall hänvisas till det arbete som genomfördes av värmepumpbranschen i samband med effsys1 och programvaran PRESTIGE.

Till sist beskrivs de nu gällande rekommendationer från Boverket gällande miljöbedömning av byggnaders energianvändning, där i huvudsak kontrakterad energis belastning enligt uppgift av leverantör rekommenderas användas. Detta är en viss skillnad från tidigare då framförallt el räknades som marginal-el, vilket nu längre inte är fallet.

Innehåll

Beräkningsprogram

Flera program finns, i princip alla tillverkare har ett eget. Inom KLIMAT21 och effsys arbetades ett branschgemensamt verktyg fram, PRESTIGE (Forsén, 2004) (Forsén, 2004), där byggnadsägaren anger vilken total energianvändning som byggnaden har, baserat på fakturor eller annan information som finns tillgänglig. Internvärme från personer och hushållsel antas via ett extra temperaturtillskott. Utifrån klimatdata fördelas sedan byggnadens energianvändning för alla timmar på året. Tappvarmvattenproduktion tas också hands om.

En del av projektet var att identifiera delar som tillgängliga dimensioneringsprogram inte har implementerat, skillnader mellan leverantörernas beräkningsresultat och deras begränsningar.

För att åstadkomma detta har två enkäter skickats ut till deltagande parter, där implementering av vissa utvalda delar i programmet anges, och där några byggnader med specificerad värmepumps-prestanda simulerats i respektive program och uppskattad energianvändning rapporterats in.

Enkät 1, Begränsningar hos programmen

I denna enkät efterfrågades huruvida programmen hade möjlighet att beräkna eller presentera specifik information eller specifik systemlösning. Enkäten var indelad på följande delar

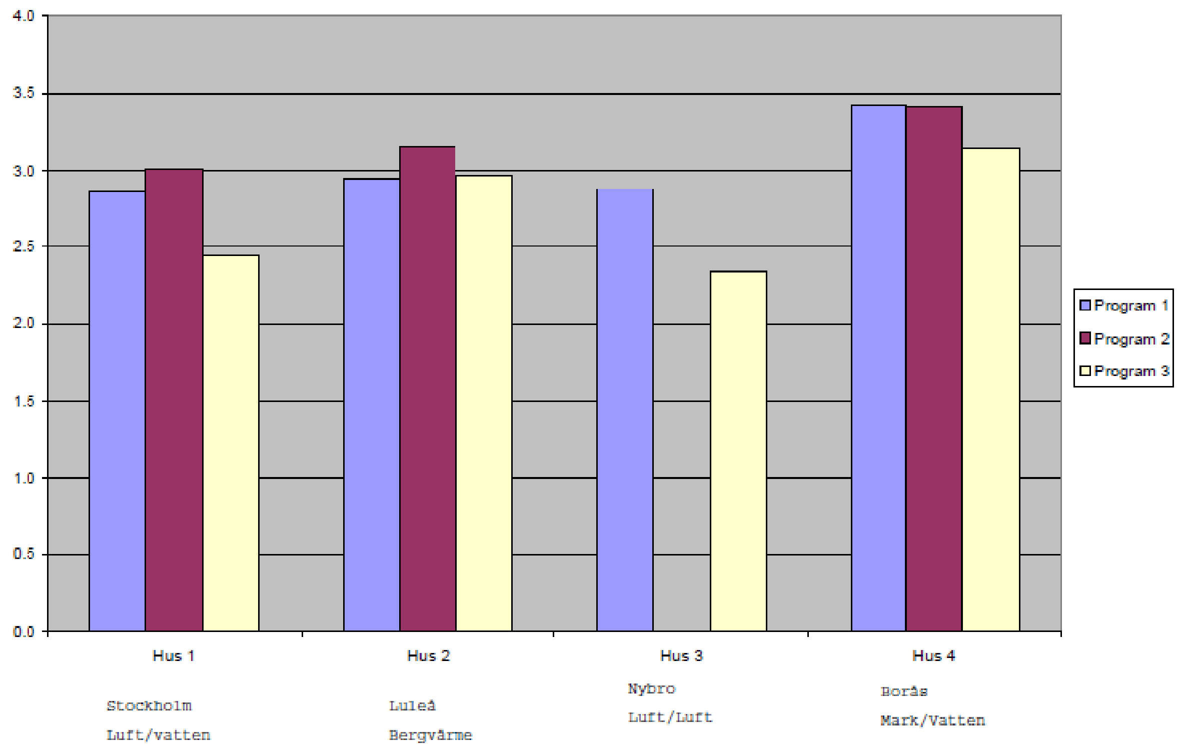
- Huset
- Värmesystemet
- Värmegenererande komponenter
- Klimat
- Resultat

Hela enkäten kan ses i ”Appendix 1 – Enkät 1”. Det kan noteras att en programvara beräknar byggnadens energianvändning, med övriga anger användningen eller enligt standard/norm. Alla fyra programmen anser att inre värmegenereringen i byggnaden är enligt schablon (t.ex. 3 °C tillskott), ingen faktiskt anpassning till byggnadens faktiska användning görs. Alla leverantörer har inte alla typer av värmepumpar, vilket avspeglar sig i vilka typer som programmen kan hantera, vidare har ingen av de som svarat annan möjlighet än att använda el eller panna som komplementsystem. Alla program som svarat har använder faktisk tappvarmvattenanvändning som läggs till totala energianvändningen. Ingen räknar med någon tappningsprofil. Inget av programmen miljövärderar det föreslagna energisystemets energianvändning. Uppvärmningskostnad ges av alla och pay-back ges av åtminstone ett program.

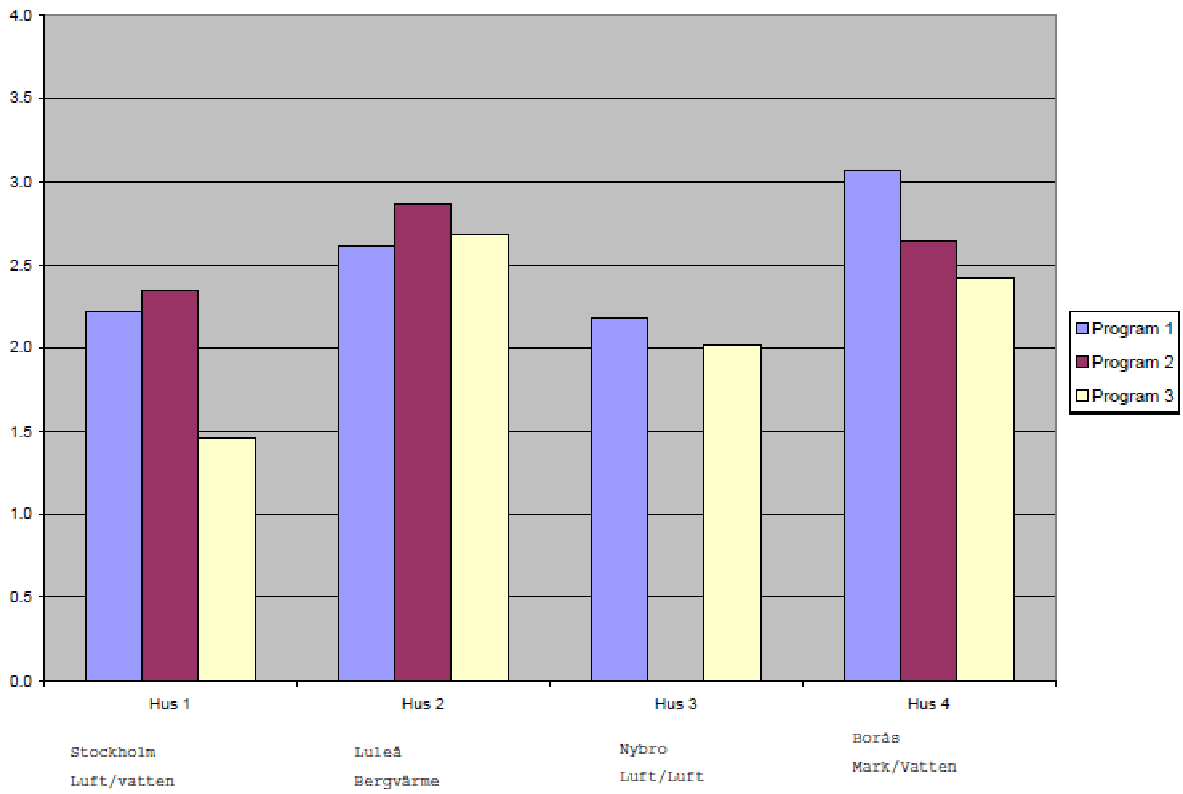
Enkät 2, Jämförelse av beräknad SPF

Fyra olika hus har simulerats av olika tillverkarna i sina egna beräkningsprogram. Nedan presenteras resultaten. Hela enkäten kan ses i ”Appendix 2 – Enkät 2”.

Figur 1 visar värmepumpsenhetens årsvärmefaktor (SPF). Det är tydligt att viss variation mellan de olika programmen råder. Det skall noteras att Program 3 tog hänsyn till avfrostning två gånger, då prestandafilen redan innehåller den informationen. Avvikelsen är alltså inte så stor som Figur 1 antyder. Det är ingen egentlig systematik kring skillnaderna förutom att just Program 3 ligger i underkant på alla fyra husen. Avvikelsen är i storleksordningen mindre än 10 %.



Figur 1: SPF för värmepumpsenheten.



Figur 2: SPF för värmesystemet

Figur 2 visar hur SPF för de olika husens värmesystem, inklusive tillsatsenergi, varierar mellan de olika programmen. Eftersom Program 3 räknade med avfrostning två gånger är skillnaden nu ännu större. Noterbart är den stora skillnaden för hus 4, med en ytjordsvärmepump. I övrigt är det inga signifikanta skillnader. Motsvarande information syns i Figur 30, men i form av köpt energi. Skillnaden är upp till cirka två tusen kronor, så skillnaden kan ha betydelse för utfallet av en installation.

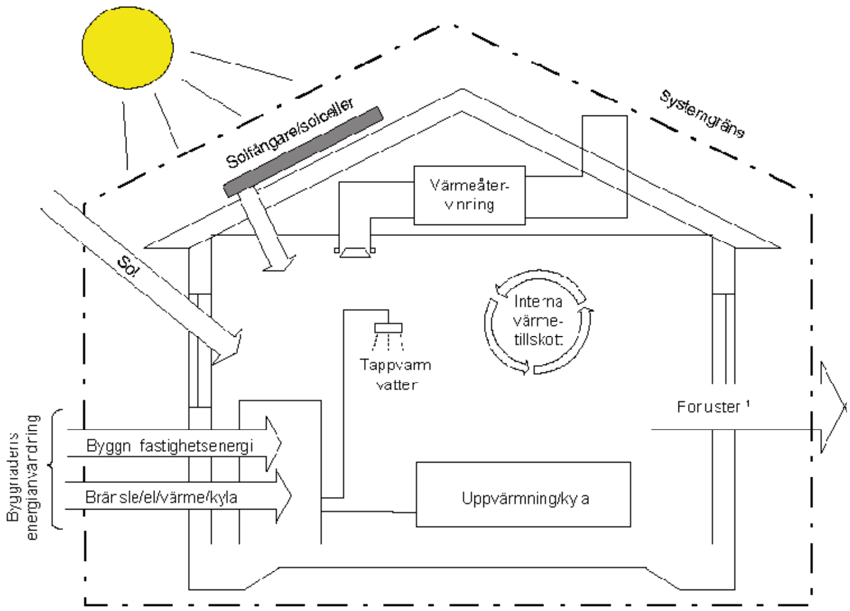
Diskussion

Det förekommer skillnader, både i hur indata hanteras, vilka uppvärmningssystem, samt hur data presenteras mellan programmen. Det är kanske naturligt att saluförs inte en viss typ av värmepump hanterar inte programmet detta. Klart är att egentligen inget av programmen kan hantera mer komplicerande sammansatta system. Det finns leverantörer som kombinerar värmepump med solpaneler, vilka kan hantera just denna systemlösning. Men överlag så har leverantörerna fokuserat på det produktutbud som de erbjuder.

När det gäller skillnader i SPF mellan programmen är det inte stor skillnad. Det ska sägas att det statistiska underlaget är väl tunt för att dra några generella slutsatser. Det finns uppenbarligen ett behov att kanske inte tvinga alla tillverkare att använda samma kod, men det kan finnas behovs att presentera ett ramverk för hur prestandan av energisystem i byggnader kan uppskattas, då det inte finns en sådan i dagsläget. Ett sådant presenteras i efterföljande kapitel.

Byggnadens energianvändning

Syftet med projektet är att basera uppvärmningssystemets effektivitet på internationella och nationella standarder. Eftersom årsvärmefaktorn för ett uppvärmningssystem beror på energianvändningen som byggnaden har, är det lämpligt att starta med att bestämma vad som egentligen avser en byggnads energianvändning. Figur 3 visar Boverkets syn på energiflödet i en byggnad. En byggnads energianvändning är enligt denna syn köpt energi för värme, kyla, fastighetsel, men inte hushållsel (alternativt verksamhetsel för lokaler). Solenergi räknas inte in som i byggnadens energianvändning, eftersom den inte är ”köpt”. På motsvarande sätt räknas inte tillförd värmeenergi till värmepumpens kalla sida in i byggnadens energianvändning eftersom den inte heller är köpt. Båda dessa anses vara ”gratis”.



Figur 3: Energiflöden i en byggnad (Boverket, 2010).

För att kunna uppskatta en byggnads energianvändning behövs ganska detaljerad information kring husets beskaffenhet, t.ex.:

1. Energianvändning i dagsläget
2. Fördelning av uppvärmningsenergi
    a. Tappvarmvatten
        i. Brukarprofil
        ii. Använd mängd
    b. Ventilation
        i. Flöden
        ii. Återvinning
    c. Transmissionförluster
        i. Fönster, U-värde
        ii. Väggar/Tak
        iii. Köldbryggor
        iv. Grund/Källare
3. Solinsläpp

    a. Orientering av byggnad
    b. Placering och storlek av fönster
    c. Fönster, SHGC-värde
    d. Solavskärmning
4. Hushållsel
5. Fastighetsel
6. Personvärme
7. Klimat
8. Värmedistributionssystem
    a. Typ
    b. Temperaturnivåer

Utan detaljerad information kring dessa delar kan egentligen ingen tillförlitlig förutsägelse ges kring vilken effektivitet ett nytt värmesystem kommer att få. Det är inte troligt att den typiske villaägaren har sådan detaljerad kunskap om sin byggnad.

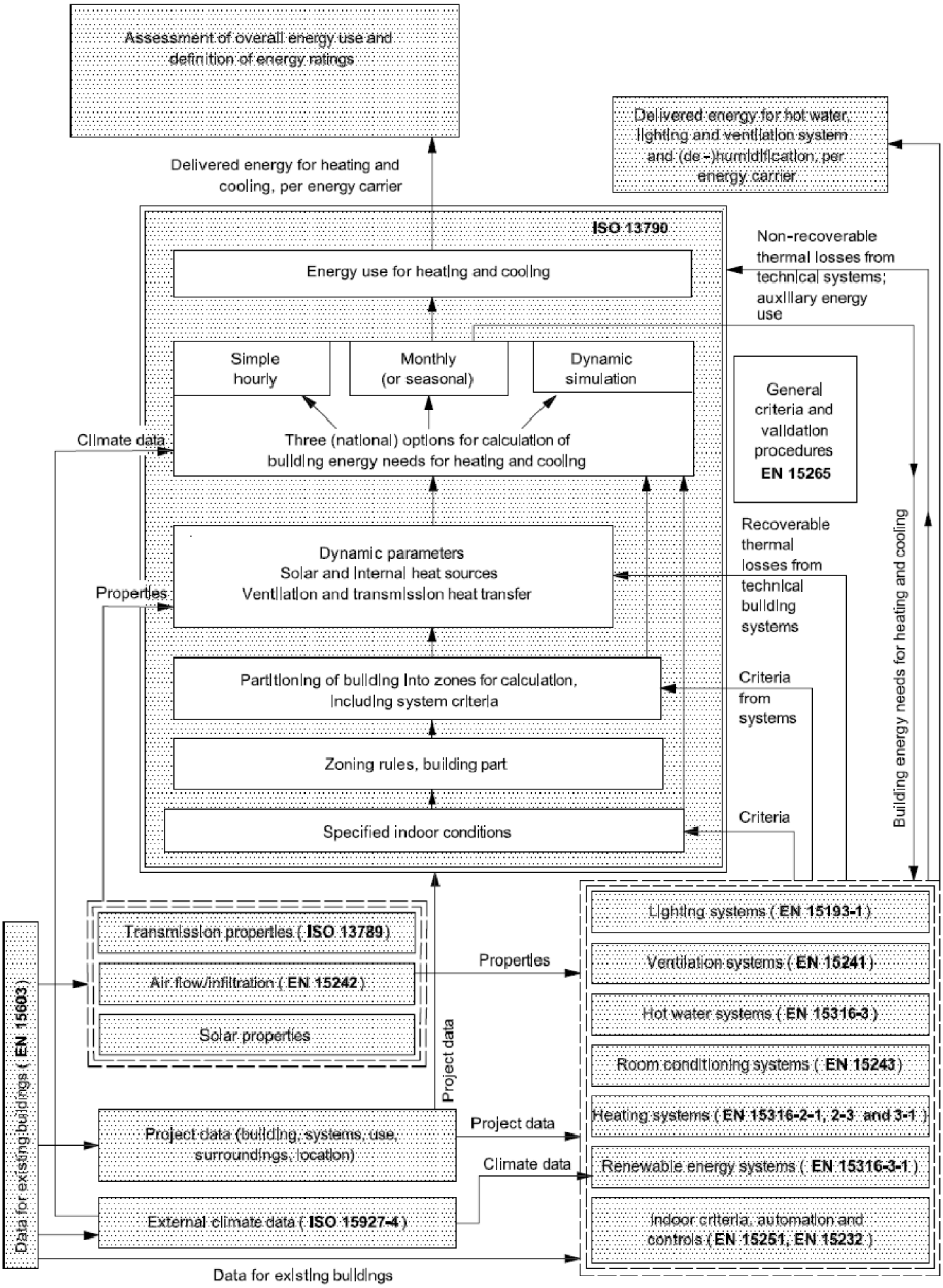
Viss hjälp finns/kommer att finnas genom energideklarationen av byggnaden, då en energiexpert har bedömt delar av dessa i byggnaderna. Denna är dock av varierande kvalité, men troligt är ändå att experten är mer skickad att göra denna bedömning än gängse villaägare.

SS-EN ISO 13790:2008

Hur energianvändningen skall bedömas finns beskrivet i (CEN SS-EN ISO 13790, 2008), se Figur 4 för en schematisk bild över beräkningsstegen. Som kan ses i figuren tillåts tre olika tidsupplösningar, timvis enkel, dynamisk, samt månatligt/säsong.

För ett energisystem som består av flera värmeproducerande komponenter, där även styrsystemet väljer mellan hur och i vilken ordning värmegeneratorerna (pannor, värmepumpar, solfångare) skall användas, är inte månatliga eller årsviktade beräkningsstrukturer tillräckligt för att bedöma systemets effektivitet. Det är därför rekommenderat att beräkningsstrukturen sker med timsteg. Det är inte rimligt att komplexa system skall handräknas utan beräknas på dator, vilket då innebär att skälet till att använda månatliga fördelningsprocedurer inte längre är viktig.

Figur 4 visar också hur den förhåller sig till andra standards. Av primärt intresse är att bedöma hur värmeproducerande komponenterna interagerar med varandra och med byggnaden. Utvärdering av de flesta tänkbara värmeavgivande komponenter finns beskrivna i standarder, främst av intresse är SS-EN 15316.



Figur 4: Beräkningsgång enligt (CEN SS-EN ISO 13790, 2008) för bedömning av en byggnads energianvändning.

Standarder för utvärdering av komponenters prestanda

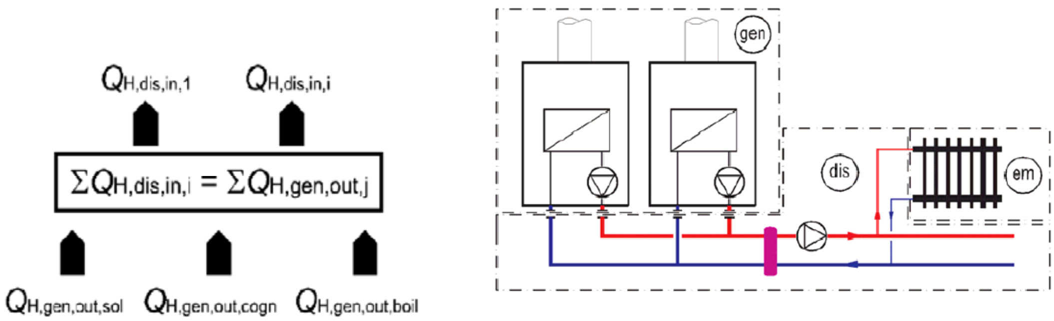
I en byggnad finns ett behov av i huvudsak värme, vilket skall tillföras byggnaden med det installerade energisystemet. Utformningen av detta system kan variera på oändligt många sätt, även identiska system kan ha olika prestanda i samma byggnad pga. t.ex. :

- implementeringen av kontrollsystemet skiljer sig,
- att användarprofilen skiljer sig, osv.

Syftet med projektet var att skapa enhetligt modell för beräkning av värmepumpsbaserade uppvärmningssystem. Systemet kan bestå av många olika komponenter som ska interagera, förhoppningsvis på bästa sätt.

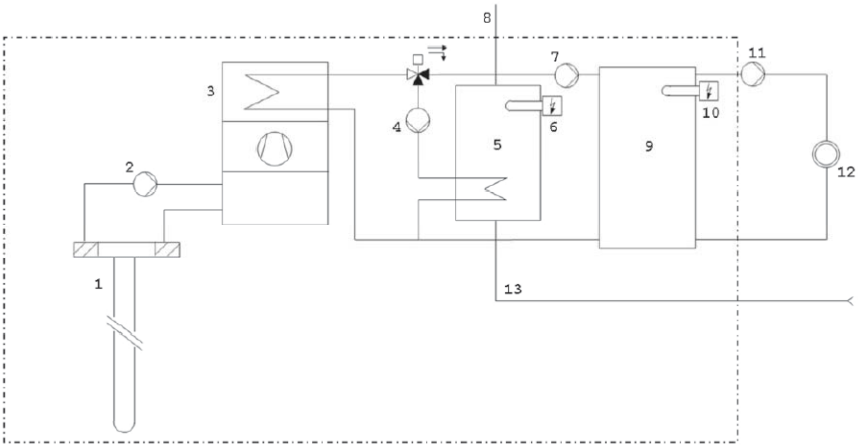
En användbar serie av standards för beräkning av värmesystems prestanda är SS-EN 15316-1 – SS-EN 15316-4 (CEN SS-EN 15316-X-X, 2008). Denna serie av standard beskriver metod för beräkning av energibehov och energieffektivitet för värmedistributionssystem (del 2-1 2-3), tappvarmvatten (del 3-1, 3-2, 3-3), värmegenerarande system (pannor 4-1, värmepumpar 4-2, solfångare 4-3, CHP 4-4, Fjärrvärme 4-5, PV-system 4-6, biomassbaserade 4-7).

Fördelningen av hur dessa värmegenererande system förser distributionssystemen med värmeenergi illustreras i Figur 5. Det är här upp till tillverkaren att definiera hur dessa skall stegas in.



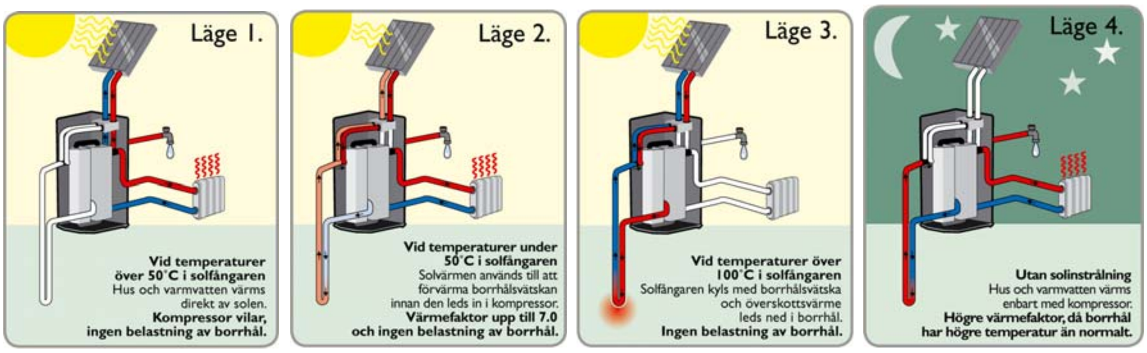
Figur 5: Summering av värmegenererande komponenters bidrag till distributionssystemen. (CEN SS-EN 15316-X-X, 2008).

Ovanstående är relativt självklart, då egentligen alla kommersiella värmepumpsystem på marknaden idag är ett kombinationssystem, där värmepumpen är baseffekt och el-patron är toppeffekt, se Figur 6



Figur 6: Enkelt värmepumpsystem enligt SS-EN 15316-4-2 (CEN SS-EN 15316-X-X, 2008).

Ett lite mer komplicerat system kan ses i Figur 7 där två värmegenererande (sol, värmepump) komponenter i systemet har fyra identifierade driftslägen. Båda systemens prestanda beror till stor del vilka temperaturer som de jobbar med.



Figur 7: Solpanelkopplad värmepumpsystem (EVI Heat, 2010).

Det är därför viktigt att vid beräkning av prestanda för respektive komponent dess omgivande tillstånd är känt eller kan beräknas. I SS-EN 15316-4-2, som gäller för värmepumpar, hänvisas till SS-EN 14511, som gäller testförfarande för on/off kapacitetsreglarede värmepumpar och chillers. För kapacitetsreglerade värmepumpar finns för närvarande ingen standard, även om det finns ett utkast (prEN 14825).

Tabell 1: Testpunkter för vätske/vätskekopplad värmepump (CEN SS-EN 14511-1--4, 2007).

		Outdoor heat exchanger		Indoor heat exchanger	
		Inlet temperature °C	Outlet temperature °C	Inlet temperature °C	Outlet temperature °C
Standard rating conditions	Water	10	7 [a]	40	45
	Brine	0	-3 [a]	40	45
	Water (for floor heating or similar application)	10	7 [a]	30	35
	Brine (for floor heating or similar application)	0	-3 [a]	30	35
Application rating conditions	Water	15	[b]	[b]	45
	Brine	5	[b]	[b]	45
	Brine (for floor heating or similar application)	5	[b]	[b]	35
	Brine	- 5	[b]	[b]	45
	Brine	0	[b]	[b]	55
	Water	10	[b]	[b]	55

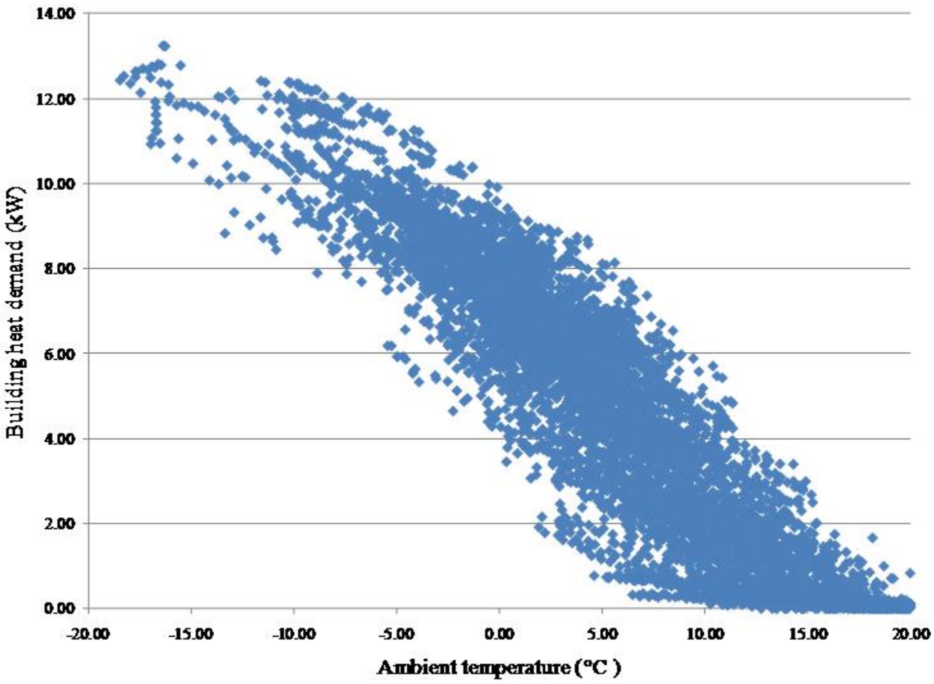
[a] For units designed for heating and cooling mode, the flow rate obtained during the test at standard rating conditions in cooling mode (see Table 8) is used.
[b] The test is performed at the flow rate obtained during the test at the corresponding standard rating conditions.

Tabell 2: Testpunkter enligt prEN 14825 (CEN TC 113 WG 7, 2009).

	A		Outdoor heat exchanger		Indoor heat exchanger	
			Ground Water	Brine	Medium temperature application Inlet / outlet temperatures (°C)	
	Part load ratio	Part load ratio %	Inlet / outlet temperatures (°C)	Inlet / outlet temperatures (°C)	Fixed outlet	Variable outlet
A	(-7-16)/(-TDesign - 16)	88%	10 / *	0/*	** / 45	**/43
B	(+2-16)/(TDesign - 16)	54%	10 / *	0 / *	**/ 45	**/35
C	(+7-16)/(TDesign - 16)	35%	10 / *	0 / *	** / 45	**/33
D	(+12-16)/(TDesign - 16)	15%	10 / *	0 / *	** / 45	**/29

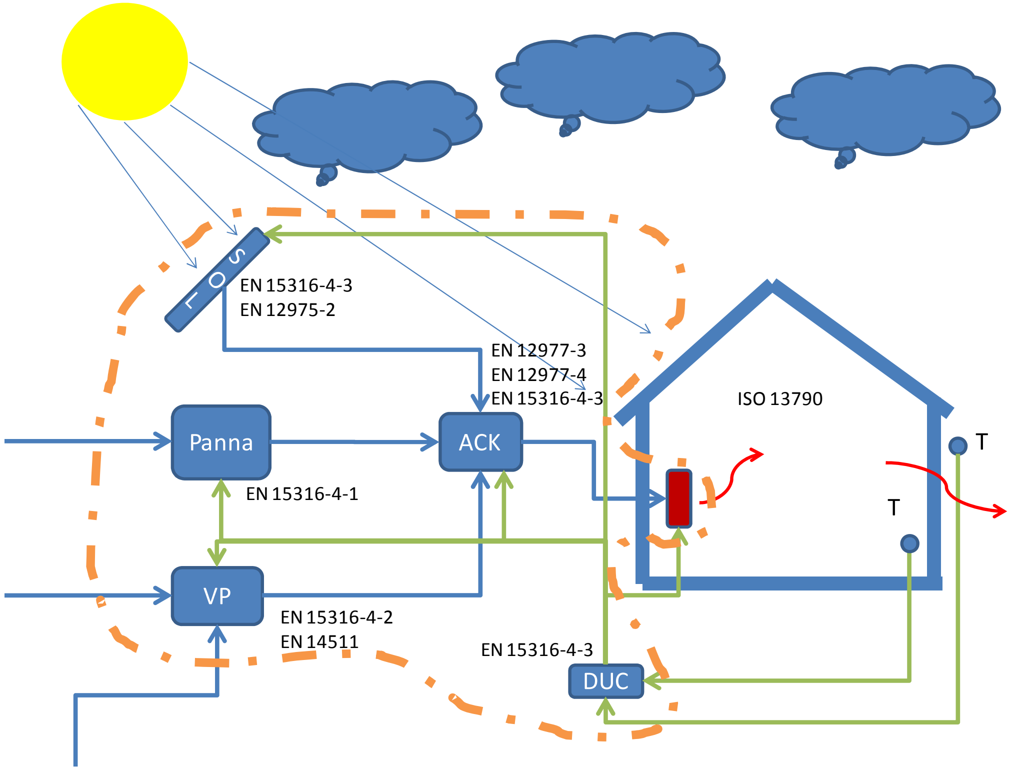
Metoduppbyggnad

Eftersom standardiserade metoder bör användas i största möjliga mån så baseras den föreslagna metoden till stora delar på europeiska standarder. Det är lämpligt att tidssteget som används vid beräkning av värmesystem är en timme. De flesta tidskonstanter som finns i systemet är mindre än så, förutom byggnaden i sig själv och marken som kan användas som värmekälla. Att inte den s.k. bin-metodiken förespråkas beror på att simuleringar av byggnadssystem indikerar att effektbehovet vid en given ute-temperatur inte enbart beror rådande ute-temperatur utan också på byggnaden och byggnadssystemens historik, se också slutrapport i effsys2 ”Projekt 9 - Dynamiska värmepumpsystem med kapacitetsreglering”. Figur 8 illustrerar detta. Speciellt besvärande kan detta bli då kombinationen värmepump och solpaneler ska installeras, då bådas prestanda beror starkt på des arbetstemperatur.



Figur 8: Byggnadens effektbehov från simulering i TRNSYS (från projekt P5).

Föreslagen metod baseras enkelt uttryckt på byggnadsnivå av ”svarta lådor” som representerar de olika komponenter i byggnaden, se Figur 9. Hur varje låda skall beräknas bestäms av tillämplig standard. Beräkningsstrukturen följer i princip energisystemet, med i motsatt riktning jämfört med värmen, dvs. beräkningen börjar vid huset och arbetar sig bakåt i energiflödet.



Figur 9: Metoduppbyggnad för beräkning av en byggnads energisystems prestanda.

Genom att studera Figur 9 identifieras de komponenter eller indata som behövs för att kvantifiera systemets prestanda:

1. Klimat
    a. Temperatur
    b. Relativ fuktighet
    c. Solstrålning, direkt
    d. Solstrålning, diffus
    e. Solstrålning, reflekterat
2. Byggnadens effektbehov
3. Värmekällan (jord, berg, sjö, osv.)
4. Ackumulatortank
5. Solfångare
6. Värmepump
7. Panna
8. Kontrollenheten
9. Driftscenarior

Klimat

I PRESTIGE (Forsén, 2004) används klimat från år 1996. Dessa klimatdata har branschen en gång enats om att de är rimliga att använda för beräkning av värmepumpsystem. Det kan nämnas att extremår för intervallet 1990 till 2000 enligt METEONORM ger god överreststämmelse för åtminstone Stockholm, övriga orter återstår att validera. I de klimatfiler som PRESTIGE använder ingår dock inte solstrålning eller relativ fuktighet, vilket innebär att dessa data behöver kompletteras.

Detta kan lämpligen göras med t.ex. METOENORM. Dimensionerande ute-temperaturen beräknas enligt PRESTIGE (Forsén, 2004)

$$DUT = EUT_6 + \frac{3}{1 - \frac{1}{e^{144/\tau_b}}}$$

där EUT_6 är extremtemperaturen på orten och τ_b är byggnadens tidskonstant. Tre definierade byggnadskaraktäristiker är kopplade till byggnadens tidskonstant, se Tabell 3.

Tabell 3: Byggnadskaraktäristik och byggnadens tidskonstant (Forsén, 2004).

Byggnadskaraktäristik	Byggnadens tidskonstant, τ_b
Lätt	24
Medel	95
Tung	150

Det finns en nyare definition av dimensionerande utetemperatur, DVUT, vilket finns beskriven i SS-EN ISO 15927-5. SMHI har beräknat DVUT enligt standard för byggnader med tidskonstanter mellan 1 till 12 dygn och finns redovisade av Boverket (Boverket, 2009). DUT enligt PRESTIGE är kallare än motsvarande DVUT, 0.2°C för lätt byggnad och ca 1.1°C för tung byggnad.

Byggnadens effektbehov och energibehov

Byggnadens effektbehov är egentligen den mest komplicerade delen av hela metodiken, i och med att dess tidskonstant är helt skild från övriga systems tidskonstanter. Jag anser ändå att det är mer rättvist för kreativa systemlösningar om dess interaktion kan hanteras på ett bra sätt, än att byggnadens dynamiska respons prioriteras. Bekvämast är om byggnaden kan implementeras i beräkningsprogrammet och då beräknas enligt gängse standards SS-EN ISO 13790 (CEN SS-EN ISO 13790, 2008). Detta är tyvärr relativt komplicerat och det finns omfattande dedikerade programpaket som gör enbart detta.

Det skulle vara önskvärt att byggnadens effektbehov kan laddas in i beräkningsstrukturen. På så sätt kan byggnadens dynamik hanteras på korrekt sätt i program som är utformat för detta, vilket sedan används i beräkningsalgoritmen för att bestämma energisystemets prestanda.

Några lämpliga programvaror kan vara t.ex. IDA ICE (Equa, 2010), DesignBuilder (DesignBuilder Software Ltd, 2010) eller IES VE (Integrated Environment Solutions, 2010). Det finns en mängd program tillgängliga för detta syfte, ovanstående program skall inte ses som en rekommendation utan snarare att det är program som vi har viss erfarenhet kring och vet att den kan göra just det som efterfrågas.

Dessa program kan vara komplicerade att använda och det kan finnas behov att förenkla metodiken. Den förenkling som här föreslås baseras på den som används av PRESTIGE (Forsén, 2004) och i standard för beräkning av dimensionerande värmeeffekt SS-EN 12831: (CEN SS-EN 12831, 2003).

En byggnads värmeeffektbehov beror på i huvudsak tre byggnadskaraktäristiska egenskaper:

- Isoleringsförmåg (U-värde)
- Täthet (infiltration)

- Ventilationsbehov

I ett kallt klimat är alla dessa termer värmeförluster ut ur byggnaden. Dessa kompenseras av i huvudsak tre olika tillförsel av energi:

- Intern värmegenerering
- Värmestrålning (SHGC-värde)
- Värmesystemet

Dessa tre tillför i huvudsak värme till byggnaden. Skillnaden mellan summa av de första tre och summan av de sista tre lagras eller avges av byggnadens termiska massa innanför isoleringen. Under antagande att temperaturen är konstant i byggnaden hela året (Forsén, 2004) kan lagringseffekten försummas och byggnadens uppvärmningsbehov via värmesystemet kan beräknas enligt:

$$\dot{\mathrm{Q}}_{\mathrm{VS}} = \dot{\mathrm{Q}}_{\mathrm{TR}} + \dot{\mathrm{Q}}_{\mathrm{INF}} + \dot{\mathrm{Q}}_{\mathrm{VENT}} - \dot{\mathrm{Q}}_{\mathrm{INT}} - \dot{\mathrm{Q}}_{\mathrm{SOL}} \qquad \text{(kW)}$$

Eftersom transmissionsförlust, infiltrationsförlust, samt ventilationsförlust är alla beroende på temperaturdifferensen mellan inne och ute[1], kan dessa summeras ihop enligt:

$$\dot{Q}_{VS} = \left(\sum UA + \rho \cdot c_p \cdot \dot{V}_{INF} + \rho \cdot c_p \cdot \dot{V}_{VENT}\right) \cdot (t_{INNE} - t_{UTE}) - \dot{Q}_{INT} - \dot{Q}_{SOL}$$

Om ingen bra uppskattning på solinstrålning eller framförallt internvärmen kan erhållas kan t.ex. skrift från Boverket eller SVEBY-programmet användas[2] (Boverket, 2007) (Levin, 2009). PRESTIGE (Forsén, 2004) satte helt enkelt bidraget av intern värmegenerering och solinstrålning till 3 K. Solinstrålning avses här strålning som inte tillförs byggnaden via värmesystemet. Solvärme som tillförs byggnaden via t.ex. solfångare ingår än så länge i Q_{VS}.

Om dessutom inverkan av termiska drivkrafter på ventilationen, vindens inverkan på infiltrationen och små förändringar i byggnadens UA-värde försummas kan parentesen ovan ansättas vara konstant.

$$\Lambda = \sum \mathrm{UA} + \rho \cdot \mathrm{c_p} \cdot \dot{\mathrm{V}}_{\mathrm{INF}} + \rho \cdot \mathrm{c_p} \cdot \dot{\mathrm{V}}_{\mathrm{VENT}} \approx \text{konstant} \qquad \text{(kW/K)}$$

vilket ger

$$\dot{\mathrm{Q}}_{\mathrm{VS}} = \Lambda \cdot (\mathrm{t}_{\mathrm{INNE}} - \mathrm{t}_{\mathrm{UTE}}) - \dot{\mathrm{Q}}_{\mathrm{INT}} - \dot{\mathrm{Q}}_{\mathrm{SOL}} \qquad \text{(kW)}$$

Finns ventilationsflöden uppmätta t.ex. genom utförd OVK kan dessa värden användas för att mer noggrant uppskatta byggnadens UA-värde. Typiska data för ventilationsflöden för bostäder är i medeltal för svenska bostäder enligt Tabell 4[3].

---

[1] Finns det värmeåtervinningssystem på avluften kommer dock inte ventilationsförlusten bero på samma temperaturdifferens som de övriga två, varpå uttrycket får anpassas

[2] Andra alternativa källor inkluderar (Adalberth, o.a., 2009) och (ATON Teknikkonsult AB, 2007).

[3] Mer detaljerad information hur ventilation och infiltration ska beräknas ges i SS-EN 15242:2007 (CEN SS-EN 15242, 2007).

Tabell 4: Typiska ventilationsflöden för bostäder i Sverige.

Ventilationssystem	Småhus (l/s·m²) (Boverket, 2007)	Flerfamiljshus (l/s·m²) (ATON Teknikkonsult AB, 2007)
Självdrag	0.25	0.33
Frånluft	0.24	0.39
Till och frånluft	0.30	0.40

Med ovanstående information tillsammans med byggnadens totala energianvändning kan värdet på Λ alternativt på $\sum UA$ uppskattas genom passningsräkning.

Innan detta kan göras måste dock varmvattenbehovet i byggnaden dras bort från byggandens totala energibehov. Det antas här att hushållselen sorteras in under $\dot{Q}_{INT}$.

För en byggnad där energianvändningen för ett år är känt anpassas nu byggnadens UA-värde, så att beräknad årsenergianvändning överrensstämmer med den kända användningen. Detta görs genom att byggnadens effektbehov beräknas för varje timme på året, vars summa då blir årsenergi-användningen.

Byggnadens uppvärmningsbehov vid dimensionerande utetemperatur beräknas sedan enligt

$$\dot{\mathrm{Q}}_{\mathrm{B}} = \Lambda \cdot (\mathrm{t}_{\mathrm{INNE}} - \mathrm{t}_{\mathrm{DUT}}) \qquad \text{(kW)}$$

Notera att detta inte normal är samma som värmesystemets behov, p.g.a. internvärme.

Många system stänger av värmegeneringen då dygnsmedeltemperaturen överstiger en viss nivå. Denna kan vara ställbar av användaren. Prestige (Forsén, 2004) använder $t_{\text{gräns}} = 11$ °C som värmegräns, vilket bör används om ingen annan information finns[4]. Byggandens värmeeffektbehov är för varje tidpunkt alltså

$$\dot{Q}_B(\tau) = \begin{cases} \Lambda \cdot (t_{INNE} - t_{ute}) & \text{för } \bar{\mathrm{t}}_{\mathrm{dag}} \leq \mathrm{t}_{\text{gräns}} \\ 0 & \text{för } \bar{\mathrm{t}}_{\mathrm{dag}} > \mathrm{t}_{\text{gräns}} \end{cases}$$

Personvärme

Enligt (CEN SS-EN ISO 13790, 2008), paragraf 10.4 vilket anger att nationellt annex kan användas. Sveby-programmet kan antas vara ett sådant och de anger rekommenderade värden enligt Tabell 5 tillsammans med Tabell 6.

Tabell 5: Personvärme enligt Sveby-programmet (Levin, 2009)

Rekommenderad personvärme	
Effekt per person	80 W
Närvaro per dygn	14 tim

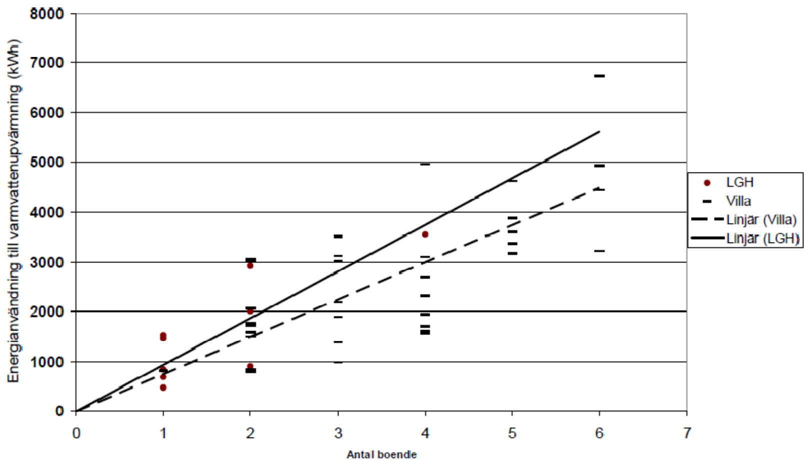
[4] Se t.ex. (Abel, o.a., 2008) för ytterligare information.

Tabell 6: Typiskt antal boende i lägenheter (Levin, 2009).

LGH typ	1 rkv	1 rk	2 rk	3 rk	4 rk	5 rk	6+ rk
Antal boende	1.42	1.42	1.63	2.18	2.79	3.51	3.51

Tappvarmvatten

I sällsynta fall vet byggnadsägaren sin varmvattenanvändning men mer ofta är denna användning okänd. Det finns ganska ny data som utifrån mätningar på faktisk användning kan uppskatta varmvattenanvändningen i både fastigheter och småhus (Energimyndigheten, 2009)[5]. Andelen varmvatten uppmättes i denna rapport till 33 % för småhus och 32 % för lägenheter av total vattenanvändning. Energianvändningen för varmvatten uppskattas i rapporten enligt Figur 10.



Figur 10: Varmvattenanvändning per hushåll (Energimyndigheten, 2009).

Baserat på Energimyndighetens data (Energimyndigheten, 2009) erhålls följande två ekvationer:

$$Q_{VV,LGH} = 979 \cdot n \cdot m \qquad \text{(kWh/år)}$$

för lägenheter och för småhus

$$\mathrm{Q_{VV,VILLA}} = 781 \cdot \mathrm{n} \cdot \mathrm{m} \qquad \text{(kWh/år)}$$

där n är antal boende i bostadsmodulen och m är antal bostadsmoduler (t.ex. antal lägenheter i en fastighet).

Ytterligare information kring uppskattningar av varmvattenanvändning kan hämtas i (ATON Teknikkonsult AB, 2007) och (Boverket, 2007).

Detaljerade instruktioner för hur beräkning kring energianvändning, tappmönster och förluster i distributionssystemet för tappvarmvatten finns beskrivet i standards SS-EN 15316-3-3 (CEN SS-EN 15316-3-3, 2007), SS-EN 15316-3-1 (CEN SS-EN 15316-3-1, 2007), SS-EN 15316-3-2 (CEN SS-EN 15316-3-2, 2007).

[5] Kallvattentemperaturen är antagen i medel till 8.6 °C för småhus och 11 °C för lägenheter. Motsvarande varmvattentemperaturer är 52 °C för småhus och 57 °C för lägenheter (Energimyndigheten, 2009).

För framförallt värmepumpar är det viktigt att inte bara veta vilken energianvändning per årsbasis, utan även när den sker på året eftersom köpt energi att tillverka varmvattnet beror på utetemperaturen. PRESTIGE använder varmvattenanvändningen per dygn och tillverkar den vid rådande utetemperatur. Varmvattenanvändning för byggnaderna är enligt (Energimyndigheten, 2009):

- 58 liter/(dygn·person) för lägenheter, motsvarande 2.68 kWh/(dygn·person)
- 42 liter/(dygn·person) för småhus., motsvarande 2.14 kWh/(dygn·person)

Tappmönster finns för typiska byggnader i standard (CEN SS-EN 15316-3-1, 2007) men även exempel finns i Energimyndighetens rapport (Energimyndigheten, 2009). Det föreslås här två olika alternativ:

1. Dygnets varmvattenproduktion antas ske under hela dygnet, där aktuellt dygns medeltemperatur används.
2. Energimyndighetens tappvarmvattenprofil och tappvattenproduktion sker efter faktiskt behov enligt styrenheten.

Metod 1 är enklare att implementera medans Metod 2 ger möjlighet till bättre passning till faktiskt beteende hos systemet vilket även medger ”smart” styrning. Metod 2 är lämplig att använda för de system där produktion av tappvarmvatten sker samtidigt/parallellt med värmeavgivning till byggnadens värmesystem. I båda metoderna används aktuell timdata, t.ex. brinetemperatur eller utetemperatur för bestämning av värmeavgivande komponentens (t.ex. en värmepump eller en solpanel) prestanda. Det är lämpligt att använda temperaturdata på varmvattensidan enligt rapport från Energimyndigheten (Energimyndigheten, 2009), se även fotnot 5 ovan. Värmepumpars prestanda skall vara testad enligt standard (CEN SS-EN 255-3, 1997). Standarden anger endast en standardpunkt vilket kanske inte är representativt för hela årets driftpunkter. Det är därför inte möjligt interpolera mellan flera punkter och om inte fler punkter utöver SS-EN 255-3 finns tillgängliga skall det antas att värmepumpen arbetar med konstant Carnot-verkningsgrad[6] för andra driftpunkter (CEN SS-EN 15316-4-2, 2008), se vidare under avsnitt ”Värmepumpar”.

System med samtidig värme och tappvarmvattenproduktion skall beräknas enligt Metod 2 ovan om validerade testdata finns, annars används Metod 1 ovan.

Det skall noteras att data enligt Energimyndighetens mätningar inte inkluderar förluster i lager och distributionssystem. Dessa måste alltså läggas till ovan angivna energimängder.

Standard (CEN SS-EN 15316-3-2, 2007) beskriver tillvägagångssätt för varmvattendistribution i byggnader och distributionssystemets förluster. Denna standard är alltså nödvändig att beakta vid nya byggnader. Distributionsförluster av tappvarmvatten kan vara betydande i byggnader, framförallt för flerfamiljshus där VVC används, d.v.s. cirkulation för varmhållning av varmvattnet.

Värmeförlusterna[7] från distributionssystemet utan VVC är enligt (CEN SS-EN 15316-3-2, 2007)

$$Q = \frac{1000 \cdot 4.2 \cdot V_{rör} \cdot (T_{vv} - T_{INNE}) \cdot \sqrt{n \cdot m \cdot 11^2}}{3600} \qquad \text{(kWh/dag)}$$

[6] Se avsnitt Värmepumpar, sidan28, för beskrivning av metodiken.

[7] Standarden anger att om inga nationella annex finns skall det anses att ingen av värmeförlusten kan återvinnas till uppvärmning.

där sista termen baseras på antal tappningar enligt (CEN SS-EN 15316-3-1, 2007). T_{INNE} repressenterar omgivande temperatur kring varmvattenrören, vilket ofta är innetemperaturen. $V_{rör}$ är volymen av vätskan (i m³) stående i rören. Standarden antar alltså att hela rörets volym sjunker till omgivningstemperaturen. Typiska rörlängder finns att använda finns i (CEN SS-EN 15316-3-2, 2007).

För system med VVC kan 40 W/m rör användas enligt (CEN SS-EN 15316-3-2, 2007) vid avsaknad av nationella värden. Mer detaljerad modell kan också användas, se (CEN SS-EN 15316-3-2, 2007). För VVC-system skall även pumpens energianvändning inkluderas, enligt

$$W_{VVC} = P_{pump} \cdot t_{op,pump} \qquad \text{(kWh/dag)}$$

där P_{pump} är pumpens märkeffekt (i kW) och $t_{op,pump}$ är drifttiden per dygn.

Hushållsel

Om inte schablonen 3 °C vill användas finns det uppgifter publicerade av Boverket kring typisk hushållselanvändning. För småhus rekommenderar (Boverket, 2007)

$$\mathrm{Q_{INT}} = (2500 + 800 \cdot \mathrm{n}) \cdot 0.7 \qquad \text{(kWh/år)}$$

Boverket anger alltså att 70% av hushållselen kan användas för uppvärmning om värmebehov föreligger (se även (Levin, 2009)).

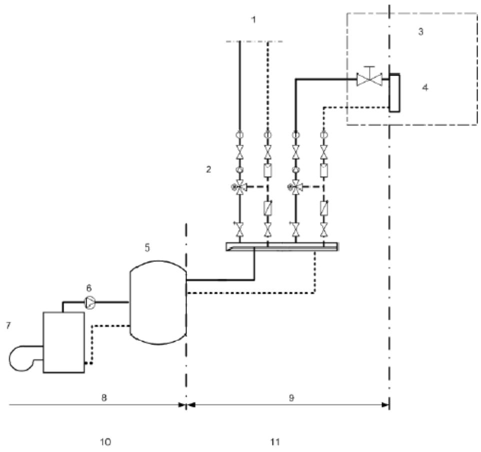
För flerbostadshus rekommenderar Boverket att hushållselen kan uppskattas enligt (Boverket, 2007)

$$Q_{INT} = 1040 \cdot m + 300 \cdot n \cdot m + Q_{TVÄTT} \cdot n \cdot m + Q_{MAT} \cdot m \qquad \text{(kWh/år)}$$

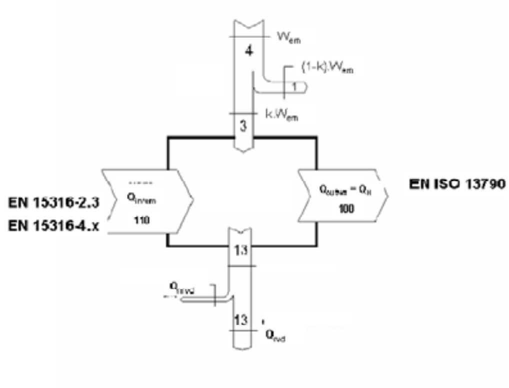
där $Q_{TVÄTT} \approx 160 \frac{kWh}{person}$ och $Q_{MAT} \approx 526 \rightarrow 730 \frac{kWh}{LGH}$

Värmesystemet

Värmeeffekten som byggnaden behöver förses till byggnaden antingen via ventilationsluften, men vanligast i svenska byggnader är att värmen i huvudsak tillförs via ett radiatorsystem. Används mekanisk till-luft kan det finnas två värmesystem, ett radiatorsystem och ett ventilationsvärme-system. Värmesystemet består i huvudsak av två delar, ett distributionssystem och ett emissions-system. Energiförluster för dessa beskrivs i standard SS-EN-15316-2-3 (CEN SS-EN 15316-2-3, 2007) samt SS-EN-15316-2-1 (CEN SS-EN 15316-2-1, 2007). Olika delar av värmesystemet illustreras i Figur 11 och systemgränsen för värmeemissionssystemet kas ses i Figur 12.



Figur 11: Olika delar av värmesystemet. (CEN SS-EN 15316-2-3, 2007)

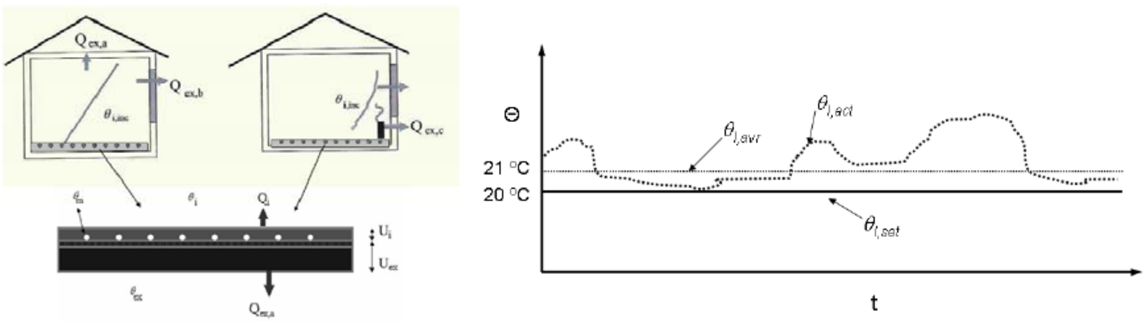


Figur 12: Systemgräns för värmeemissionssystemet (CEN SS-EN 15316-2-1, 2007).

SS-EN-15316-2-1 beskriver i huvudsak den extra energi som behöver tillföras emissionssystemet p.g.a.:

- Ej konstant temperatur i byggnadens alla delar
- Värmeavgivande enheter inbyggda i byggnadsstrukturen
- Kontrollnoggrannhet på styrning av temperaturnivåer i byggnaden

Exempel på dessa extra förluster illustreras i Figur 13.



Figur 13: Extra energi pga. styrning, placering av emissionskomponenter och ojämn temperaturfördelning (CEN SS-EN 15316-2-1, 2007).

Dessa förluster är redan inbakade i den angivna energianvändningen som byggnadsägaren anger, men föreslås ändå inkluderas i beräkning av det nya energisystemets energiprestanda främst ur konservativt perspektiv, alternativt görs beräkningen iterativt med befintligt värmesystem först. Byggnadens värmebehov fås genom att jämföra befintligt systems simulerade energianvändning med känd användning. Genom att inkludera denna del kommer det troliga utfallet vara i överkant (utan iteration) mot verkligt utfall. Energi per tidsenhet (timme) beräknas enligt (CEN SS-EN 15316-2-1, 2007)

$$\mathrm{Q_{VS,IN}} = \mathrm{Q_{VS}} - \mathrm{k} \cdot \mathrm{W_{VS}} + \mathrm{Q_{VS_{LOSS}}} \qquad \text{(kWh)}$$

Den totala energi som behöver tillföras värmesystemet ($Q_{VS,IN}$) är högre än husets energianvändning (Q_{VS}), på grund av avvikelser från idealt värmesystem. Viss del av pumparbete och fläktarbete i värmesystemet kan återvinnas ($k \cdot W_{VS}$) enligt SS-EN-15316-2-1. Ingen vägledning av storleken på återvinningsgraden (k) ges i standarden och sättes därför till noll (k=0), dvs ingen återvinning av denna energi antas. Förlusterna som beror på avvikelser från idealt ($Q_{VS_{LOSS}}$) värmesystem beräknas enligt (CEN SS-EN 15316-2-1, 2007)

$$\mathrm{Q_{VS_{LOSS}}} = \left(\frac{1.03}{\eta_{\mathrm{em}}} - 1\right) \cdot \mathrm{Q_{VS}} \qquad \text{(kWh)}$$

där

$$\eta_{em} = \frac{1}{[4-(\eta_{str}+\eta_{ctr}+\eta_{emb})]}$$

vilket tar hänsyn till stratifiering av temperaturen inne, inbyggda värmeelement i byggnadsstrukturen, samt p.g.a. reglerutrustning. Inverkan av kontrollenheten för rumstemperaturen, föreslås enligt Tabell 7 för ej inbyggda radiatorer:

Tabell 7: Effektivitet av rumstemperaturreglering (CEN SS-EN 15316-2-1, 2007), ej inbyggda radiatorer.

Typ av reglering	η_{ctr}
Framledningsreglering	0.80
En central rumskontroll	0.88
P-reglering (2K)	0.93
P-reglering (1K)	0.95
PI-reglering	0.97
State-of-art	0.99

Approximativa värden för övertemperaturer föreslås enligt

$$\eta_{str} = \frac{1.948-0.0016\cdot\overline{\Delta T_{rad}}}{2}$$

men mer information kan fås i (CEN SS-EN 15316-2-1, 2007). $\cdot\overline{\Delta T_{rad}}$ representerar medelövertemperaturen på radiatorsystemet i förhållande till rumstemperaturen[8].

För golvvärmesystem eller andra inbyggda system hänvisas till (CEN SS-EN 15316-2-1, 2007).

[8] Om t.ex. radiatortemperaturen är vid en given tidpunkt 33 och 41, är $\overline{\Delta T_{rad}} = \frac{(41-20)+(33-20)}{2} = 17$°C.

Värmesystemets ”radiatorkurva”, d.v.s. hur framledningstemperaturen och returtemperaturen ändras med utetemperaturen behövs för att dels avgöra hur förlusterna ovan kan bedömas men framförallt för att bestämma de värmeavgivande komponenternas effektivitet. I föreliggande rapport hänvisas till metodiken beskriven inom ramen för PRESTIGE (Forsén, 2004), i vilket radiatorkurvan för befintligt system uppskattas genom initialt bestämma konstanterna K_1, K_2, som beräknas vid dimensionerande ute-temperatur enligt

$$K_1 = \left. \frac{\left(\frac{t_{fram} - t_{retur}}{ln\left(\frac{t_{fram} - t_{inne}}{t_{retur} - t_{inne}}\right)} \right)^{1+n}}{t_{fram} - t_{retur}} \right|_{DUT}$$

$$K_2 = \left. \frac{t_{inne} - t_{ute}}{t_{fram} - t_{retur}} \right|_{DUT}$$

Utöver dessa konstanter använder PRESTIGE två hjälpvariabler enligt

$$\alpha = \mathrm{K}_2 \cdot \mathrm{e}^{\frac{\ln\left(\frac{\mathrm{K}_1 \cdot (\mathrm{t}_{\mathrm{inne}} - \mathrm{t}_{\mathrm{ute}})}{\mathrm{K}_2}\right)}{1+\mathrm{n}}}$$

och

$$\beta = e^{\frac{t_{inne} - t_{ute}}{\alpha}}$$

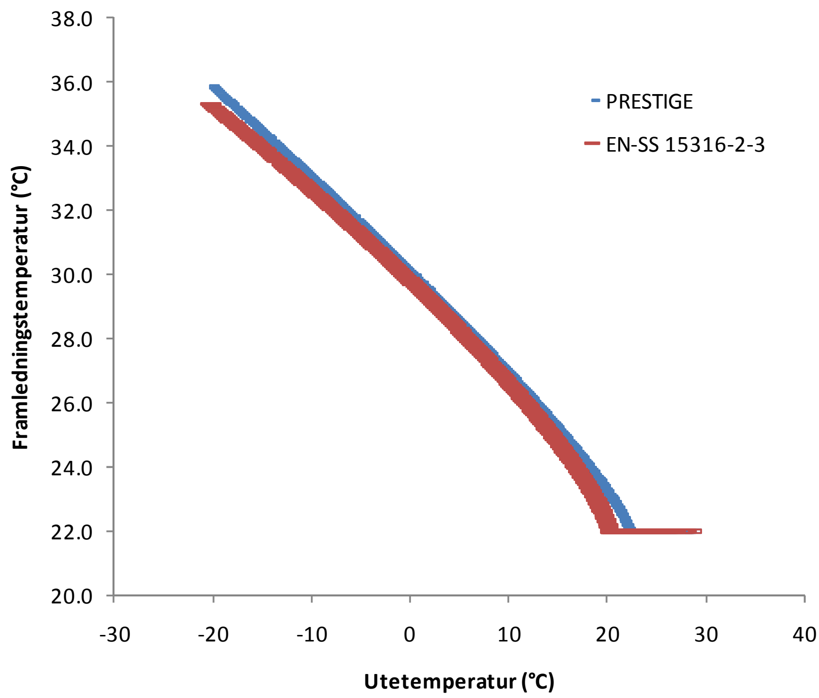
Från dessa konstanter och hjälpvariabler beräknas framlednings och returledningstemperaturen för värmeemissionssystemet enligt

$$\mathrm{t}_{\mathrm{fram}} = \frac{\beta \cdot (\mathrm{t}_{\mathrm{inne}} - \mathrm{t}_{\mathrm{ute}}) + \mathrm{K}_2 \cdot \mathrm{t}_{\mathrm{inne}} \cdot (\beta - 1)}{\mathrm{K}_2 \cdot (\beta - 1)} \qquad (°\mathrm{C})$$

och

$$\mathrm{t}_{\mathrm{retur}} = \frac{(\mathrm{t}_{\mathrm{inne}} - \mathrm{t}_{\mathrm{ute}}) + \mathrm{K}_2 \cdot \mathrm{t}_{\mathrm{inne}} \cdot (\beta - 1)}{\mathrm{K}_2 \cdot (\beta - 1)} \qquad (°\mathrm{C})$$

Det finns även beskrivet i standard (CEN SS-EN 15316-2-3, 2007) hur radiatorkurvan kan beräknas. En jämförelse för ett lågtempererat värmesystem visas i Figur 14, där det framgår att föreslagen procedur ger något högre framledningstemperaturer.



Figur 14: Jämförelse mellan framledning beräknad enligt PRESTIGE och EN-SS 15316-2-3.

Distributionssystemets förluster och drivenergi beräknas enligt standard (CEN SS-EN 15316-2-3, 2007), där enbart distributionssystemets drivenergi skall tas hänsyn till. Tryckfall i t.ex. pannor eller värmepumpar skall beaktas separat, se respektive generatorenhet. I befintliga hus där värmesystemet inte skall ändras, d.v.s. cirkulationspumpen inte byts ut mot ny mer effektiv kan drivenergiinverkan anses inte påverka utfallet av systemets prestanda. För system där t.ex. frekvensstyrning av cirkulationspumpen avses att installeras, så som finns integrerat i vissa värmepumpar, behöver drivenergin tas med, då frekvensstyrningen påverkar värmesystemets prestanda. Det föreslås därför att drivenergin för alltid beaktas.

Proceduren följer standard (CEN SS-EN 15316-2-3, 2007) enligt förenklad metod (Annex A) för byggnader där ingen eller bristande information kring värmesystemet finns. Metoden är alltså att betrakta som en miniminivå. Den detaljerade beräkningsmetodiken enligt standard kan också användas om mer information finns att tillgå, se (CEN SS-EN 15316-2-3, 2007).

Första steget i att bestämma pumpenergi är att uppskatta tryckförlusterna i distributions- och emissionssystemet.

$$\Delta p_{VS} = \frac{1+f_{comp}}{1000} \cdot R \cdot L + 2 + \Delta p_{FH} + \Delta p_G$$

Uppskattningsvis antas då i standard R = 100 Pa/m, samt f_{comp} = 0.3. Δp_{FH} = 25 kPa är tillägg om golvvärme finns. Δp_G är tryckfallet över generatorn (pannan, värmepumpen). Om inte denna är känd antas värden ur Tabell 8.

Tabell 8: Uppskattning av tryckfall genom generatorn (CEN SS-EN 15316-2-3, 2007).

Typ av generator		Δp_G (kPa)
Vatten i generator > 0.3 l/kW		1
Vatten i generator ≤ 0.3 l/kW	< 35 kW	$20 \cdot \dot{V}_{des}^2$
Vatten i generator ≤ 0.3 l/kW	≥ 35 kW	80

$\dot{V}_{des}$ är volymflödet i värmesystemet vid DUT, i enheten m³/h.

Systemets rörlängd kan uppskattas enligt

$$L = 2 \cdot \left(L_L + \frac{L_W}{2} + N \cdot h + l_c\right)$$

där L_L är zonens (byggandens) längd, L_W är zonens bredd, N är antal uppvärmda plan, samt där l_c är 10 m för två-rörsystem eller L_L+L_W för enrörssystem.

Det är nu möjligt att beräkna den ”hydrauliska” effekten (i Watt) som måste tillföras enligt

$$P_{hydr,des} = \frac{1000}{3600} \cdot \Delta p_{VS} \cdot \dot{V}_{des}$$

där trycket anges i kPa och flödet i m³/h. Den årliga hydrauliska energin som behövs beräknas enligt

$$W_{VS,hydr} = \frac{P_{hydr,des}}{1000} \cdot \frac{\sum_1^{8760} \frac{\dot{Q}_{VS}}{\dot{Q}_{VS,DUT}}}{8760} \cdot t_{op}$$

där t_{op} är antal timmar per år som kräver uppvärmning, vilket är byggnadsspecifikt. Drivenergin (kWh/år) till pumpen som behövs beräknas från

$$W_{VS} = W_{VS,hydr} \cdot \left(1.25 + \sqrt{\frac{200}{P_{hydr,des}}}\right) \cdot 1.5 \cdot b \cdot \left(C_{P1} + C_{P2} \cdot \left(\frac{\sum_1^{8760} \frac{\dot{Q}_{VS}}{\dot{Q}_{VS,DUT}}}{8760}\right)^{-1}\right)$$

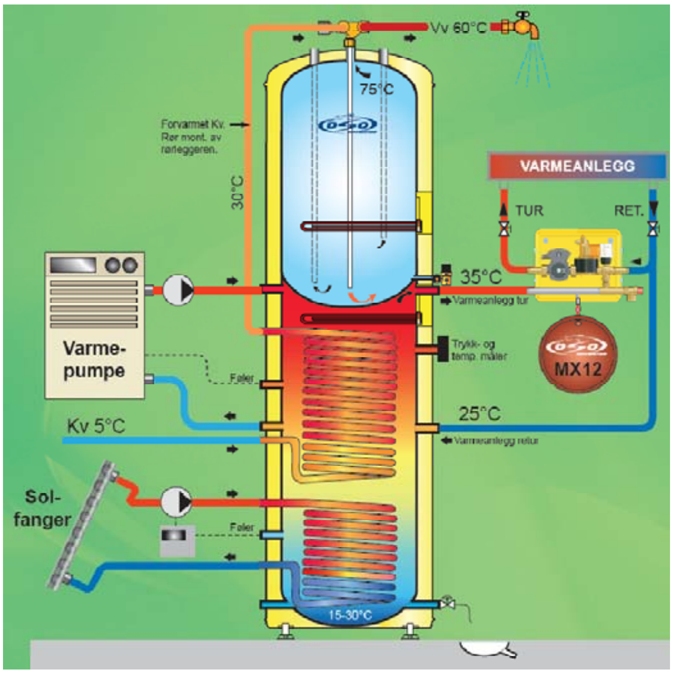
där b är 1 för nya och 2 för existerande byggnader. Övriga variabler väljs enligt Tabell 9.

Tabell 9: Konstanter för beräkning av elenergi till cirkulationspump.

Typ av pumpkontroll	C_{P1}	C_{P2}
Ingen kontroll	0.25	0.75
Varvtalsstyrning, Δp= konst	0.75	0.25
Varvtalsstyrning, Δp= variabelt	0.90	0.10

Ackumulatortank och styrsystem

I många enkla system med värmepumpar finns kanske enbart en tappvarmvattentank, och ingen tank på radiatorsystemet. I kombinationssystem behöver systemen samordnas och t.ex. pannor är inte speciellt känsliga på drifttemperaturen på vattnet in utan ger relativt konstant verkningsgrad, medans värmepumpar och solfångare ger en noterbar försämring ju högre temperaturen är. Det är viktigt att värmekällorna samordnas och kan avge sin värmeenergi till systemet på bästa sätt. Det är nästan alltid därför nödvändigt att ansluta en ackumulatortank mellan värmeproducerande komponenterna och värmesystemet, se Figur 15.



Figur 15: Inkoppling av ackumulatortank mellan värmeproducerande enheter och värmsystemet (OSO, 2010).

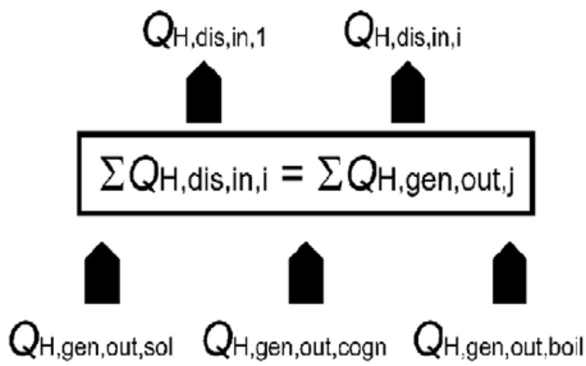
För att till fullo utnyttja en bra tank i beräkningsmetoden skall den vara testad enligt (CEN SS-EN 12977-3, 2008) eller (prCEN/TS 12977-4) beroende på tanktyp.

Om tankens prestanda inte verifierats enligt standardiserade metoder antas det att tanken sett ur värmesynpunkt är helt omblandad, dvs. inga temperaturgradienter finns i tanken. Detta är som tidigare nämnts negativt sett både ur solfångarprestanda som ur värmepumpsprestanda. Det kan finnas anledning att se över om en standardiserad beräkningsalgoritm behövs för ackumulatortankar för uppvärmningssystem i bygganderr[9].

För enkla system med enbart en värmegenererator (med eller utan tillsatsvärme) kan ackumulatorn finnas inne i själva generatorn (pannor) eller inte finnas alls. I dessa fall används radiatortemperaturerna enligt avsnitt ”Värmesystemet”.

Ackumulatortankens syfte är alltså att tillåta överproduktion av värmeenergi i subsystemen som sedan lagras i tanken. I kombinationssystem med flera värmegeneratorer är troligen ackumulator-tanken starkt förknippat med värmesystemets styrsystem eftersom de värmegenerande komponenter kombineras sedan till ackumulatortanken, se Figur 16.

[9] Vilket jag förstår pågår inom standardiseringsarbetet.



Figur 16: Sammanföring av olika generatorkomponenter (CEN SS-EN 15316-4-1, 2008).

Fördelningen av energiproduktionen mellan de olika sub-generatorsystemen sker enligt leverantörens styrschema. Om inget sådant finns tillgängligt antas det att alla delgeneratorer (sol, värmepump, panna) alla går på samma dellastförhållande, baserat på dess kapacitet vid den givna driftspunkten (CEN SS-EN 15316-4-1, 2008). Dellastförhållandet är alltså då

$$\beta = \frac{\dot{Q}_{VS,IN}}{\sum_i \dot{Q}_{gen,i}}$$

Om styrsystemet prioriterar ordningen på värmegenerande komponent (t.ex. först sol, sedan värmepump, sedan pannor, osv…), är dellastförhållande på sista inkopplade generatorn

$$\beta_i = \frac{\dot{Q}_{VS,IN} - \sum_{i-1} \dot{Q}_{gen,i,på}}{\dot{Q}_{gen,i}}$$

Det är via smart systemlösning kombinerat med smart kontroll av systemen som olika tillverkares uppvärmningssystem skiljer sig åt. Mer information kan fås i SS-EN 15232:2007 (CEN SS-EN 15232, 2007). Förluster i systemen p.g.a. styrningen beskrivs i detalj, i huvudsak hänvisar SS-EN 15232:2007 tillbaka till respektive del av standarden SS-EN 15316. Dessa förluster har alltså redan behandlats eller behandlas i efterföljande avsnitt.

Termiska förluster i ackumulatortank finns beskrivet i SS-EN 15316-4-3 (CEN SS-EN 15316-4-3, 2007), som behandlar solfångarsystem. Det föreslås att i standarden beskriven metodik tillämpas något modifierat på den gemensamma ackumulatortanken. Termiska förluster från tanken är då

$$\dot{Q}_{ack,loss} = UA_{ack} \cdot \left(T_{tank} - T_{omg}\right)$$

där U_{ack} fås från leverantör, alternativt

$$UA_{ack} = 0.16 \cdot \sqrt{1000 \cdot V_{sol}} \qquad \text{(W/K)}$$

om ingen data finns. V_{sol} avser volym i tanken (m³) som används för lagring av energi från solfångaren, vilket inte behöver vara hela tankens fysiska volym. Tankens temperatur antas vara medelvärdet av radiatortemperaturerna, enligt avsnitt ”Värmesystemet”. Omgivande temperatur (T_{omg}) är innetemperaturen om tanken står inne, utetemperaturen om den står utomhus, samt

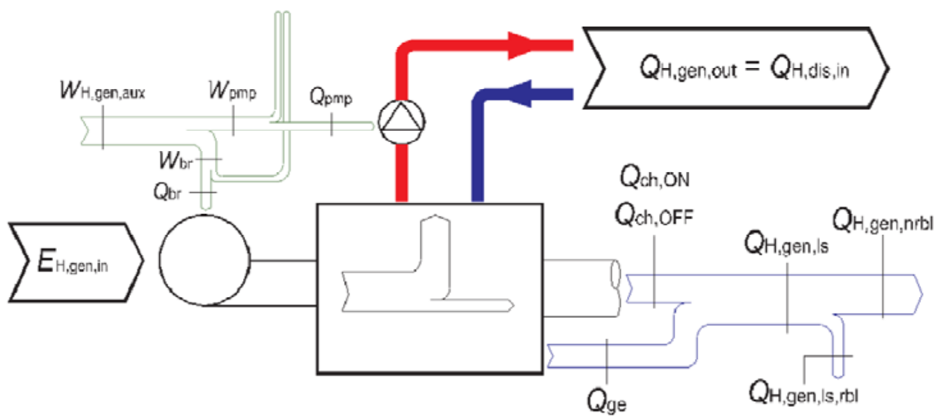
$$T_{omg} = T_{ute} + \frac{T_{inne} - T_{ute}}{2}$$

om tanken står i ett ouppvärmt bi-utrymme.

Andelen av denna energi som kan återvinnas i form av värme till byggnaden avgörs var tanken är placerad och om förlusterna ger upphov till övertemperaturer i detta utrymme. Eventuell inbyggda tankar i generatorn (pannor och värmepumpar) beräknas separat under respektive komponent. Förutsatt att tanken är placerad inne i ett uppvärmt utrymme anger (CEN SS-EN 15316-4-3, 2007) att 100 % av energin kan återvinnas[10]. Tankar placerad ute återvinner ingen av denna energi medans om tanken är placerad i ett ouppvärmt bi-utrymme kan 50 % återvinnas.

Värmepannor

Det finns en mängd med pannor, och även om kombinationen panna/värmepump är vanlig kan andra kombinationer förekomma. Gällande standard för beräkning av prestanda av förbrännings-baserade pannor är SS-EN 15316-4-1:2008 (CEN SS-EN 15316-4-1, 2008). Även beräkningsexempel för antal pannor finns inkludera i informativa annex. Proceduren tar hänsyn till diverse förluster som uppstår med denna typ av värmekomponent, se Figur 17. Pannors prestanda skall vara uppmätt enligt gällande standards[11].



Figur 17: Pannmodell enligt SS-EN 15316-4-1 (CEN SS-EN 15316-4-1, 2008).

Metoden för uppskatta pannors prestandan bygger på avsnitt 5.3 i (CEN SS-EN 15316-4-1, 2008). Metoden principiellt följer struktur enligt

a) Data hämtas för
    a. Verkningsgrad vid 100% last
    b. Verkningsgrad vid dellast (30%)[12]
    c. Förluster vid "tomgång" (0%)
b) Verkningsgrader och prestanda korrigeras för aktuell driftpunkt
c) Förluster vid fullast och dellast beräknas
d) Tillsatsenergier (t.ex. pumpar) uppskattas
e) Återvunna förluster beräknas

[10] En mer rimlig andel som kan återvinnas kan förslagsvis vara lika stor som aktuellt värmeeffektbehov jämfört med värmeeffektbehov vid DUT.
[11] EN 297, EN 303-5, EN 304, EN 656, EN 15034, EN 15035 och/eller EN 15456.
[12] Ska enligt standard motsvara minsta effekt med brännaren igång.

Om systemet innehåller flera värmegenererande system (komponenter) fördelas deras last enligt avsnitt ”Ackumulatortank och styrsystem”.

Pannans dellast beräknas enligt

$$\beta = \frac{\dot{Q}_{behov}}{\dot{Q}_{100\%}}$$

där effektbehovet korrigerats för avgiven värme från andra högre prioriterade värmegeneratorer (t.ex. solfångare eller värmepump). Pannans avgivna effekt är då

$$\dot{Q}_{\beta} = \beta \cdot \dot{Q}_{100\%}$$

Pannans verkningsgrad kan korrigeras om drifttemperaturen på pannan skiljer sig från testpunkten, enligt

$$\eta_{100\%,T} = \eta_{100\%} + f_{100\%,T} \cdot (T_{100\%} - T)$$

Motsvarande ekvation används för att korrigera för dellast. Korrektionsfaktorn, $f_{100\%,T}$, skall anges i nationellt annex, men om inget sådant finns kan Figur 18 användas.

Generator type	Boiler average water temperature at boiler test conditions for full load $\theta_{gnr,w,test,Pn}$	Correction factor $f_{corr,Pn}$
Standard boiler	70 °C	0,04 %/°C
Low temperature boiler	70 °C	0,04 %/°C
Gas condensing boiler	70 °C	0,20 %/°C
Oil Condensing boiler	70 °C	0,10 %/°C

Generator type	Generator average water temperature at boiler test conditions for intermediate load $\theta_{gnr,w,test,\ Pint}$	Correction factor $f_{corr,Pint}$
Standard boiler	50 °C	0,05 %/°C
Low temperature boiler	40 °C	0,05 %/°C
Gas condensing boiler	30 °C (*)	0,20 %/°C
Oil Condensing boiler	30 °C (*)	0,10 %/°C
(*) Return temperature		

Figur 18: Panntemperaturkorrektionsfaktor enligt (CEN SS-EN 15316-4-1, 2008).

För tomgång korrigeras värmeförlusten för panntemperatur enligt

$$\dot{Q}_{0\%,loss,T} = \dot{Q}_{0\%,loss} \cdot \left(\frac{T - T_{rum}}{\Delta T_{\text{test}}}\right)^{1.25}$$

Typiska värden finns i redovisade i (CEN SS-EN 15316-4-1, 2008). För den faktiska driftpunkten beräknas pannans verkningsgrad (och därmed dess termiska förluster) genom linjär interpolering[13].

Hjälpenergi för pannan (om sådan finns) interpoleras på motsvarande sätt från testdata, baserat på pannans dellast. Även hjälpenergi som åtgår under perioder då pannan inte används skall beaktas.

[13] $y = \frac{x - x_1}{x_2 - x_1} \cdot (y_2 - y_1)$

Den energi som kan återvinnas som nyttig värme beräknas enligt

$$\dot{Q}_{rec} = \dot{Q}_{0\%,loss,T} \cdot (1 - b_{brm}) \cdot f_{env} \cdot \beta$$

b_{brm} och f_{env} beror på placering och brännartyp. I avsaknad av nationellt annex kan data enligt Figur 19 användas.

Burner type	$f_{gnr,env}$
Atmospheric burner	0,50
Fan assisted burner	0,75

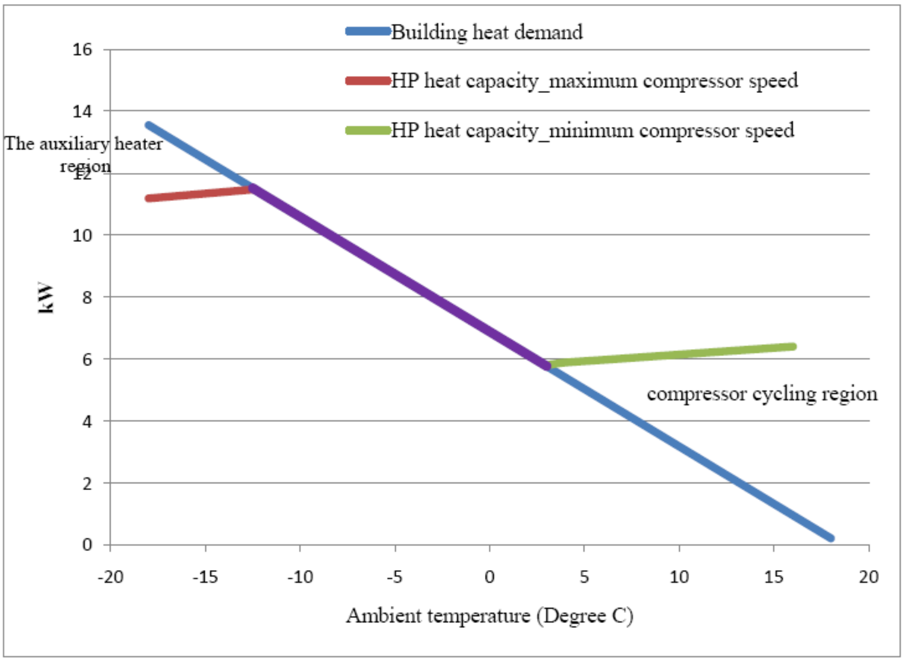
Generator location	Temperature reduction factor b_{brm} -	Installation room temperature $\theta_{i,brm}$ °C
Outdoors	1	θ_{ext}
In the boiler room	0,3	13
Under roof	0,2	5
Inside heated space	0,0	20

Figur 19: Återvinningsfaktorer för placering och typ av brännare.

Värmepumpar

Värmepumpar är vanligt förekommande i svenska bostäder och är alltid ett kombisystem på så sätt att det finns en elektrisk (oftast) värmare som spetseffekt. Ofta ingår dessa i samma mekaniska enhet. Aktuell standard för beräkning av värmepumpkomponenten i ett energisystem i byggnader är SS-EN 15316-4-2:2008 (CEN SS-EN 15316-4-2, 2008). Det är starkt fokus på att beräkna värmepumpens prestanda enligt den s.k. bin-metoden[14], vilket är delvis konstigt eftersom i de andra delarna av samma standardserie (SS-EN 15316) används tidsindelning snarare än bin. Även Annex 28 inom IEA använder bin-metoden (Wemhöner, o.a., 2006). Det är inte svårare rent beräkningsmässigt att beräkna 8760 gånger än 50 ggr, tar bara längre tid. PRESTIGE (Forsén, 2004) använder sig av timsteg, och tar enbart sekunder att räkna, bortsett från ytjordsmodulen som tar längre tid. Dessa metoder är inte avsedda för handräkning. Används kalkylark är tidssteg inte nämnvärt mer komplicerat än bin, snarare tvärtom. Vidare blir bin-metoden helt oanvändbar då kombinationssystem med flera värmegenererande komponenter med tillhörande kontrollsystem, eftersom det inte är självklart vilket värmegenererande system som körs vid given utetemperatur. Vidare är byggnadens energibehov unikt kopplat till utetemperaturen utan även till byggnadens historik och användning. Figur 20 illustrerar enkel ”linjär” byggnad och värmepumpens värmeavgivning för en kapacitetsreglerad värmepump (från projekt P5).

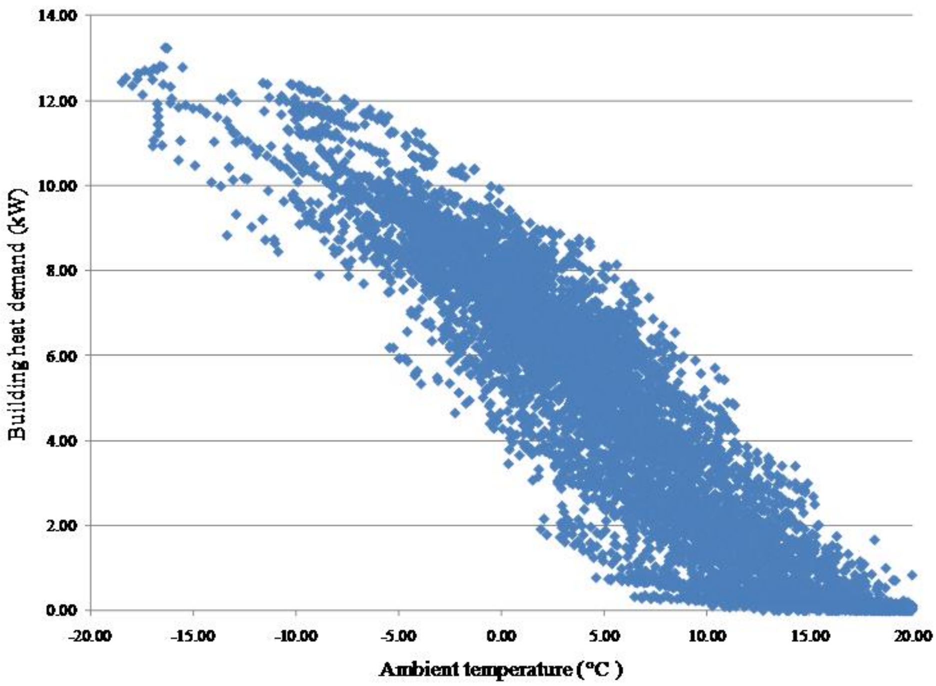
[14] Liknande varaktighetskonceptet. Visst förvalt utetemperaturintervall (bin) (t.ex. 1°C) sorteras efter antal timmar som denna temperatur inträffar, varpå värmepumpens prestanda beräknas, kompressorns effekt beräknas och dess energianvändning fås genom att multiplicera med antal timmar som aktuellt intervall (bin) råder.



Figur 20: Koppling mellan ”enkel” linjär byggnads effektbehov och utetemperatur (från P5).

Eftersom byggandens effektbehov i första hand skall beräknas med dynamiska detaljerade simuleringsmodell (se avsnitt ”Byggnadens effektbehov”) och då kan byggnadens effektbehov se ut enligt Figur 21.

Det föreslås således att även värmepumpar beräknas med samma tidssteg som övriga delar i systemet, dvs. en timme.



Figur 21: Byggnadens effektbehov från simulering i TRNSYS (från projekt P5).

Från värmesystemet eller ackumulatorn erhålls dels flödet och dels temperaturen in på varma sidan av värmepumpen. Denna tillsammans med kalla sidans ingående temperatur används för att bestämma värmepumpens prestanda vid aktuell driftpunkt. Värmepumpens prestanda skall vara uppmätt enligt gällande standards, dvs. hela SS-EN 14511-1 till 4 för icke kapacitetsreglerade värmepumpar (CEN SS-EN 14511, 2007), SS-EN 255-3 för tappvattenvärmning (CEN SS-EN 255-3, 1997), samt SS-EN 14825 om och när den träder i kraft för kapacitetsreglerade värmepumpar (CEN TC 113 WG 7, 2009).

Eftersom värmepumpen enbart mäts upp med fåtal punkter behöver aktuell driftpunkts prestanda interpoleras utifrån de mätta punkterna.

Tabell 10: Mätpunkter definierad för vatten/vatten-värmepump (CEN SS-EN 14511, 2007).

		Outdoor heat exchanger		Indoor heat exchanger	
		Inlet temperature °C	Outlet temperature °C	Inlet temperature °C	Outlet temperature °C
Standard rating conditions	Water	10	7 [a]	40	45
	Brine	0	-3 [a]	40	45
	Water (for floor heating or similar application)	10	7 [a]	30	35
	Brine (for floor heating or similar application)	0	-3 [a]	30	35
Application rating conditions	Water	15	[b]	[b]	45
	Brine	5	[b]	[b]	45
	Brine (for floor heating or similar application)	5	[b]	[b]	35
	Brine	- 5	[b]	[b]	45
	Brine	0	[b]	[b]	55
	Water	10	[b]	[b]	55

[a] For units designed for heating and cooling mode, the flow rate obtained during the test at standard rating conditions in cooling mode (see Table 8) is used.
[b] The test is performed at the flow rate obtained during the test at the corresponding standard rating conditions.

I SS-EN 15316-4-2 beskrivs i det informativa Annex C att interpoleringen kan ske under antagande om konstant exergetisk verkningsgrad, dvs. konstant Carnot-verkningsgrad[15]. Denna interpoleringsmetod föreslås framför den linjära interpolering av avgiven värme och kompressoreffekt som används av PRESTIGE (Forsén, 2004).

I Tabell 11 visar katalogdata för IVT Greenline G35 (IVT Industrier AB, 2006). Metoden för interpolering illustreras för denna maskin. Tabelldata gäller för 0 °C in i förångaren. Beräkning av COP_{2C} och COP2 och sedan carnot-verkningsgraden ger Tabell 12. Det är tydligt att just denna maskin har relativt konstant Carnot-verkningsgrad inom det angivna temperaturintervallet.

Tabell 11: Data för IVT Greenline G35 (IVT Industrier AB, 2006).

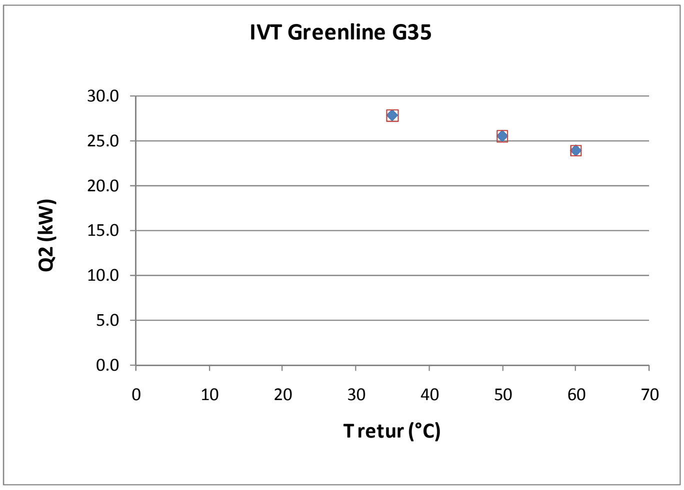
T1	Q1	E
35	36.2	8.4
50	36.5	11.0
60	37.1	13.2

[15] Detta är inte helt sant, i standarden framgår det att det inte är Carnot-verkningsgraden, utan att värmefaktorerna skall användas. Vi är dock traditionellt mer vana vid att Carnot-verkningsgraden för en värmepumpande maskin använder sig av köldfaktorerna.

Tabell 12: Beräknade värden för IVT Greenline G35.

T1	Q2	COP1	COP2	COP1c	COP2c	Carnot eff.
35	27.8	4.3	3.3	8.8	7.8	0.42
50	25.5	3.3	2.3	6.5	5.5	0.42
60	23.9	2.8	1.8	5.6	4.6	0.40

Genom att linjärinterpolera i Carnot-verkningsgraden när T_1 varierar fås ett robust värde på värmepumpens effektivitet jämte den teoretiskt möjliga övre gräns. För att komma vidare behöver antingen $\dot{Q}_1$, E eller $\dot{Q}_2$ bestämmas på något sätt. Här visar det sig att $\dot{Q}_2$ beter sig "snällt", se Figur 22.



Figur 22: $\dot{Q}_2$ s.f.a. inkommande temperatur på värmepumpens varma sida.

Det är alltså möjligt att även interpolera så att kyleffekten kan uppskattas för det aktuella driftfallet. När både kyleffekt och Carnot-verkningsgrad är framtagna är det enkelt att beräkna värmepumpens avgivna värmeeffekt vid aktuell driftpunkt.

Metoden beskriven är lite mer robust än att direkt interpolera med värmeavgivande effekt och kompressoreffekt är författarens erfarenhet. Metoden är även robust nog för moderata extrapoleringar.

Värmepumpens prestanda beror på både temperaturen på kondensorsidan (varma sidan, värmesänkan) och på förångarsidan (kalla sidan, värmekällan). Standard (CEN SS-EN 15316-4-2, 2008) anger att alla tillgängliga testpunkter skall användas. Åtminstone skall standardtestpunkt och applikationstestpunkter användas, men även andra om sådana finns. Figur 23 illustrerar exempel på indatablad[16] till beräkningsmetodiken. Notera att testpunkterna inte överresstämmer med gällande standard utan är de som används i PRESTIGE. Det är standardpunkter enligt SS-EN 14511 som skall användas.

[16] Struktur taget från indata i PRESTIGE (Forsén, 2004) med tillägget för tappvarmvatten.

Tillverkare	SVEP		Max framledning	55	°C
Typ av värmepump	Uteluft/Vatten		Max returledning	48	°C
Modell	Bäst		Lägsta driftpunkt	-10	°C
Köldmedium	R666		Nom KB-pump		W
Fyllnadsmängd	0.1	kg	**Tappvarmvatten**	55	°C
			COP vv	1.67	-
			Carnotverkningsgrad	0.29	

Värmeeffekt (kW)	Framledningstemperatur (ut ur kondensorn) °C					
Utetemperatur °C	**35**	**50**	**Max**			
-7	2.47	2.05	1.96			
2	3.47	3.09	2.975			
7	4	3.63	3.485			

Tillförd effekt (kW)	Framledningstemperatur (ut ur kondensorn)					
Utetemperatur °C	**35**	**50**	**Max**			
-7	0.96	1.05	1.09			
2	1.03	1.15	1.23			
7	1.06	1.208	1.303			

Figur 23: Testdata enligt SS-EN 14511 och SS-EN 255 3.

Interpolering (och extrapolering) skall ske linjärt mellan de två närmaste punkterna. Vid extrapolering antas att carnotverkningsgraden är samma som vid närmaste punkten, medans kyleffekten extrapoleras enligt de två närmaste punkterna.

Interpoleringen skall ske först på köldbärarsidan, för de olika tabellerade värmebärarsidans temperaturer. Den erhållna listan med prestandadata för den eftersökta driftpunkten interpoleras sedan på värmebärarsidan, varpå en ”bi-linear” interpolation på en två-dimensionell tabell har genomförts.

Kyleffekt (kW)		Framledningstemperatur (ut ur kondensorn)					
Utetemperatur °C		**35**	**50**	**Max= 55**			
	-7	1.51	1	0.87			
	2	2.44	1.94	1.745			
	7	2.94	2.422	2.182			

Figur 24: Prestandadata för exempelvärmepump[17].

I Figur 24 visas typiska prestandadata för en värmepump. Interpolering sker först på köldbärarsidan. För en driftpunkt på 0°C/45°C är första interpoleringen (-7°C):

$$\dot{Q}_{W35} = \frac{\dot{Q}_{B2/W35} - \dot{Q}_{B-7/W35}}{2°C - (-7°C)} \cdot (0°C - (-7°C)) + \dot{Q}_{B-7/W35}$$

[17] Notera att punkterna inte överrensstämmer med SS-EN 14511.

och andra interpoleringen

$$\dot{Q}_{W50} = \frac{\dot{Q}_{B2/W50} - \dot{Q}_{B-7/W50}}{2°C-(-7°C)} \cdot (0°C - (-7°C)) + \dot{Q}_{B-7/W50}$$

Tredje interpoleringen sker nu på värmebärarsidan, med värden från interpolering 1 & 2 enligt

$$\dot{Q} = \frac{\dot{Q}_{W50} - \dot{Q}_{W35}}{50°C-35°C} \cdot (50°C - 35°C) + \dot{Q}_{W35}$$

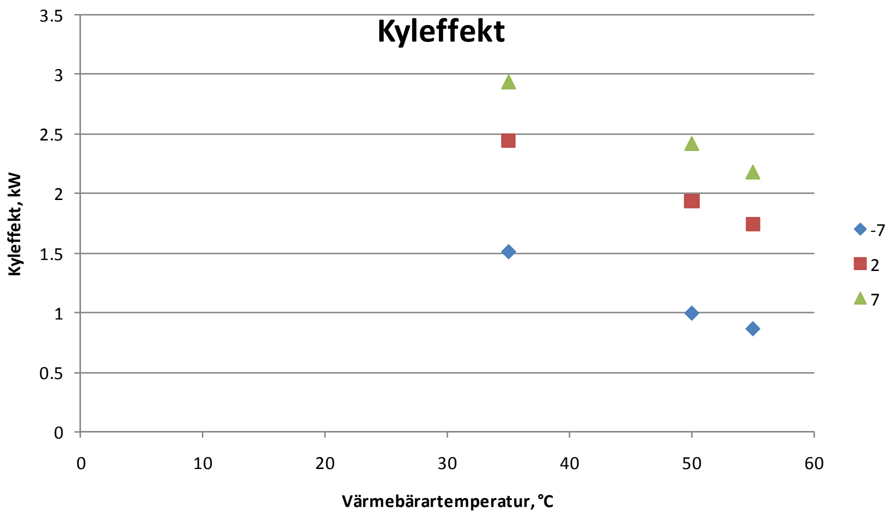
I detta fall blir då kyleffekten då 1.90 kW. Metoden används i (CEN SS-EN 15316-4-2, 2008) men även PRESTIGE (Forsén, 2004) använder samma matematiska modell.

Carnot-verkningsgrad		Framledningstemperatur (ut ur kondensorn)					
		0	**35**	**50**	**Max= 55**	**55.010**	**110.02**
Utetemperatur °C	**-50.005**	0	0	0	0	0	0
	-10.001	0	0	0	0	0	0
	-10	0.25	0.25	0.20	0.19	0	0
	-7	0.25	0.25	0.20	0.19	0	0
	2	0.28	0.28	0.29	0.27	0	0
	7	0.28	0.28	0.31	0.29	0	0
	100	0.28	0.28	0.31	0.29	0	0

Kyleffekt (kW)		Framledningstemperatur (ut ur kondensorn)					
		0	**35**	**50**	**Max= 55**	**55.010**	**110.02**
Utetemperatur °C	**-50.005**	0.00	0.00	0.00	0.00	0.00	0.00
	-10.001	0.00	0.00	0.00	0.00	0.00	0.00
	-10	2.29	1.21	0.70	0.59	0.00	0.00
	-7	2.64	1.51	1	0.87	0.00	0.00
	2	3.65	2.44	1.94	1.745	0.00	0.00
	7	4.25	2.94	2.422	2.182	0.00	0.00
	100	14.87	12.46	11.91	10.95	0.00	0.00

Figur 25: Modifierad prestanda effekt för användning vid extrapolering och implementering av värmepumpens arbetsområde.

För extrapolering och implementering av värmepumpens arbetsområde kan prestanda tabellerna modifieras (egentligen behöver enbart tabell för Carnot-verkningsgrad samt kyleffekt modifieras) enligt Figur 25. Det framgår i figuren att värmepumpen inte arbetar nedanför -10°C på kalla sidan (uteluft i denna uteluft/vatten-värmepump) eller med värmebärartemperaturer över +55°C. Inom arbetsområdet men utanför testpunkterna har kyleffekten extrapolerats för varje respektive värmebärartemperatur genom linjär regression av kyleffekt. Eftersom enbart tre punkter var tillgängliga användes en rät linje, vilket data antyder gäller i detta fall, se Figur 26. Viss försiktighet bör råda vid denna extrapolering, framförallt om data antyder att funktionen inte följer en rät linje.



Figur 26: Kyleffekt från testdata för olika utetemperaturer.

Interpolering i tabellerna för carnotverkningsgrad (η_C) samt kyleffekt ($\dot{Q}_2$) (Figur 25) tillsammans med beräkning av den teoretiska köldfaktorn (Carnotska köldfaktorn) bestämmer värmepumpens avgivna värmeeffekt genom relationerna:

$$COP_{2C} = \frac{T_{KB}+273.15}{T_{VB}-T_{KB}}$$

$$COP_2 = COP_{2C} \cdot \eta_C$$

$$\dot{E} = COP_2 \cdot \dot{Q}_2$$

$$\dot{Q}_1 = \dot{Q}_2 + \dot{E}$$

där $\dot{E}$ är tillförd kompressoreffekten utan eventuell köldbärarpumpeffekt, vilket läggs till senare i till värmepumpens effektanvändning. Eftersom prestanda för värmepumpen inte är känt än så är inte T_{VB} eller känt, utan iteration får genomföras för att bestämma utgående värmebärartemperatur. Dessutom kan flödet genom värmepumpen inte vara samma som under testerna. Om så är fallet skall prestandan för värmepumpen justeras (CEN SS-EN 15316-4-2, 2008). Flödet genom kondensorn kan vara samma som för radiatorsystemet, vilket kan vara samma för alla driftpunkter eller justeras genom antingen termostater eller annan kapacitetsstyrning. Det kan också vara en separat krets med sitt egna flöde, vilket torde vara det sannolika för system med flera värmekällor som avger sin värme till en ackumulatortank. Vilket flöde som används i varje driftpunkt kan alltså inte sägas generellt utan är en del av systemuppbyggnaden och styralgoritmen. Det viktiga är att hänsyn tas till om flödet genom kondensorn inte är samma som för testpunkterna enligt SS-EN 14511-2. Detta görs enligt (CEN SS-EN 15316-4-2, 2008)

$$COP_{\Delta T} = COP_{SS-EN\ 14511} \cdot \left[1 - \frac{\frac{\Delta T_{SS-EN\ 14511} - \Delta T_{VB}}{2}}{\left\{T_{VB} - \frac{\Delta T_{SS-EN\ 14511}}{2} + \Delta T_{sk} - (T_{KB} - \Delta T_{sc})\right\}}\right]$$

där

$$\Delta T_{sk} = \begin{cases} 4 & \text{för vattenkylda kondensorer} \\ 15 & \text{för luftkylda kondensorer} \end{cases}$$

och $$\Delta T_{sc} = \begin{cases} 4 & \text{för vattenkylda förångare} \\ 15 & \text{för luftkylda förångare} \end{cases}$$

Varmvattenproduktion

Värmepumpen skall vara testad för varmvatten enligt gällande svenska standard, för närvarande SS-EN 255-3. Proceduren börjar med testdata enligt standard. Figur 23 illustrerar nödvändig data. Från standarden angiven $COP_{2C,vv}$ beräknas värmepumpens Carnot-verkningsgrad för testpunkten, se Figur 27 för aktuella temperaturnivåer. Det antas att varmvattenbehovet tillgodoses helt av värmepumpen om värmepumpen befinner sig inom sitt driftsområde. Som tidigare nämndes används dygnsmedelvärde på temperaturen ute för bestämning av kyleffekten i efterföljande steg, medans Carnot-verkningsgraden antas vara konstant (CEN SS-EN 15316-4-2, 2008). För att beräkna värmepumpens $COP_{2C,vv}$ behövs tillsammans med Carnot-verkningsgraden även den Carnotska köldfaktorn ($COP_{2C,vv}$). Denna beräknas utifrån inkommande köldbärartemperatur (luft för luftvärmda förångare) tillsammans med den enligt energimyndighetens observerade varmvattentemperatur (57°C för lägenheter, 52°C för villor). Den i kompressorn använda energi som åtgår för varmvattenproduktion per dygn beräknas enligt

$$E = COP_{2C,VV} \cdot Q_{VV}$$

om värmepumpen befinner sig inom sitt driftsområde. Om värmepumpen inte kan gå på grund av omgivande temperaturnivåer (t.ex. för kallt ute för en uteluft/vattenvärmepump) används nästa värmekälla (t.ex. tillsats).

Type of heat source	Heat source temperature °C	Range of ambient temperature of heat pump °C	Ambient temperature of storage tank °C
Outside air	7 (6)	from 15 to 25 from 0 to 7 *	20
Indoor ambient air	15 (12)	15 **	15
Exhaust air	20 (12)	from 15 to 25 from 0 to 7 *	20
Water	10	from 15 to 25 from 0 to 7 *	20
Brine	0	from 15 to 25 from 0 to 7 *	20

Figur 27: Temperaturnivåer för testning av olika värmepumpstyper i varmvattenproduktion.

Om inte testdata för värmepumpen finns enligt SS-EN 255-3 kan denna uppskattas från värmepumpens prestanda enligt SS-EN 14511 enligt

$$COP_{2,vv} = \frac{COP_{2,EN14511} + 1}{2}$$

d.v.s. ett medelvärde mellan prestanda i värmedrift och direktverkande el[18].

Energikälla för värmepumpen

Fördelen med värmepumpar är att stor andel av energin som avges som värme ”fås gratis” from omgivningen. Som tidigare nämnts är värmepumpens effektivitet starkt beroende på vilka temperaturer som den arbetar mellan. Därför är även temperaturnivån på värmekällan viktig att

[18] Det ska noteras att det inte finns stöd för denna uppskattning i standard.

bestämma korrekt. Ingen standard har identifierats som anger hur temperaturnivån explicit skall bestämmas. Det står i SS-EN 15316-4-2 att:

- För uteluftvärmepump skall uteluftens temperatur användas
- För frånluftsvärmepump utan värmeåtervinning skall innetemperaturen användas.
- För frånluftsvärmepump med värmeåtervinning skall återvinningsvärmeväxlaren och värmepumpens prestanda mätas tillsammans eller så skall temperaturen beräknas efter återvinningen genom värmeväxlarens temperaturverkningsgrad.
- För mark-, berg-, eller sjövärmepumpar hänvisas till nationellt annex. Om inget sådant finns kan **Error! Reference source not found.** användas för markvärmepump eller 0°C för grundvattenvärmepump. (CEN SS-EN 15316-4-2, 2008).

Det finns något som skulle kunna kallas nationellt annex, dvs. det arbete som utfördes under första effsys-programmet inom ramen för PRESTIGE (Forsén, 2004). Där beskrivs modell för både markvärme och bergvärme. Eventuellt kanske en mer noggrann modell kan ersätta bergvärmemodellen i PRESTIGE, då arbete fortgår (Javed, 2010) att utveckla en mer detaljerad och noggrann, men till dess användes modeller enligt PRESTIGE (Forsén, 2004).

Solfångarsystem

Solenergi som används i en byggnad anses precis som en värmepumps energiupptag from omgivningen som ”gratis”. Solfångarsystem beräknas enligt standard SS-EN 15316-4-3:2007 (CEN SS-EN 15316-4-3, 2007). Solfångarsystem kräver som indata solstrålning, så de klimatfiler som används i t.ex. PRESTIGE behöver kompletteras med soldata. Typiska samhöriga data för utetemperatur och solinstrålning måste bestämmas.

Enligt standard skall solfångarsystem beräknas per månad, två olika metoder för detta finns att tillgå. Detta är inte tillräckligt små tidssteg då kombisystem skall utvärderas där prioritetsordningen på systemen har stor inverkan. Istället föreslås att även solfångarsystem beräknas per timme.

En solfångares verkningsgrad kan skrivas som (Solar Energy Laboratory, 2006) (Duffie, o.a., 1991)

$$\eta = \eta_0 - a_1 \cdot \frac{\Delta T}{I_T} - a_2 \cdot \frac{\Delta T^2}{I_T}$$

där η_0, a_1 och a_2 är konstanter som bestäms experimentellt enligt standard SS-EN 12975-2. Solfångarens effekt fås alltså sedan som

$$\dot{Q}_{sol} = A \cdot I_T \cdot \eta$$

Solfångaren antas alltid leverera positiv effekt, d.v.s.

$$\dot{Q}_{sol,ut} = \max\left(0, \dot{Q}_{sol}\right)$$

Om det i solfångarloopen även sitter en värmeväxlare mellan tank och solfångare skall dess negativa inverkan på solfångarens effektivitet tas hänsyn till (Duffie, o.a., 1991). Även förlusterna i anslutande rör samt solfångarens IAM som påverkar solfångarens prestanda skall inkluderas (Solar Energy Laboratory, 2006) (Duffie, o.a., 1991).

CO_2-belastning av värmesystemet

Avsikten med projektet var också att värmesystemets miljöbelastning kunde bedömas, vilket i huvudsak inbegriper mängden CO_2 som systemet orsakar. Boverket (Boverket, 2010) har gett rekommendationer i sina anvisningar till energiexperter som utför energideklarationer hur dessa kan uppskattas. Det är därför lämpligt att även värmepumpsbaserade system använder samma bedömningsgrund.

För bränslen (fasta, flytande, gasformiga) som används i värmesystemet hänvisar Boverket till Naturvårdsverket (Naturvårdsverket, 2010) som anger omräkningsfaktorer för olika bränslen och dess miljöpåverkan. För fjärrvärme hänvisar Boverket till fjärrvärmedistributören eller energibolaget (Boverket, 2010). För el som är kanske mest relevant för värmepumpsbaserade uppvärmningssystem hänvisar Boverket till att en individuell bedömning av CO_2-belastningen genom att beakta byggnadsägarens elavtal och elleverantörens uppgifter kring utsläpp för det aktuella avtalet. I praktiken innebär detta väl att väljs ett elavtal som innebär vindel, vattenkraft eller kärnkraft påverkar värmepumpen inte miljön via ökade utsläpp.

Detta har sedan länge varit ett omdiskuterat område hur el skall miljöbedömas, länge har det ansetts att eftersom värmepumpen använder el ”på marginalen” skall också ”marginal-el” användas för miljöbedömningen, vilket för värmepumpar varit ofördelaktigt. Boverkets synvinkel synes vara mer pragmatisk.

Slutsatser

Projektet har syftat till att identifiera de behov för vidareutveckling och homogenisering av beräkningsalgoritmer för att bestämma värmepumpsbaserade uppvärmningssystems effektivitet per årsbasis i småhus och flerfamiljshus. Enkät har genomförts vilket visar att alla program inte kan hantera mer komplexa uppvärmningssystem som t.ex. solfångare tillsammans med värmepump. Skillnaden på årsvärmefaktorn mellan de fiktiva fastigheterna är inte speciellt besvärande, med beaktande att en leverantör räknade avfrostning dubbelt. Dock pekar det på vikten av riktig och rak kommunikation kring den information som utbytes mellan kund och säljare.

Som andra del i projektet har metodik för beräkning av byggnaders energiprestanda tagits fram med beaktande på de internationella och svenska standarder som finns. Strukturen hänvisar i stor utsträckning till standarder där så tillämpligt och i vissa fall hänvisas till det arbete som genomfördes av värmepumpbranschen i samband med effsys1 och programvaran PRESTIGE.

Till sist beskrivs de nu gällande rekommendationer från Boverket gällande miljöbedömning av byggnaders energianvändning, där i huvudsak kontrakterad energis belastning enligt uppgift av leverantör rekommenderas användas. Detta är en viss skillnad från tidigare då framförallt el räknades som marginal-el, vilket nu längre inte är fallet.

Referenser

Adalberth Karin och Wahlström Åsa Energibesiktning av byggnader - Flerbostadshus och lokaler [Bok]. - Stockholm : SIS Förlag AB, 2009.

ATON Teknikkonsult AB Energideklarering av bostadsbyggnader - Metoder för besiktning och beräkning, version 2 [Rapport]. - Stockholm : ATON Teknikkonsult AB, 2007.

Boverket BFS 2007:4 BED 1: Boverkets föreskrifter och allmänna råd om energideklaration för byggnader [Rapport]. - Karlskrona : Anders Larsson, 2007. - BFS 2007:4 BED 1.

Boverket Dimensionerande vinterutetemperatur, DVUT [Online] // Dimensionerande vinterutetemperatur, DVUT. - Boverket, den 16 November 2009. - 2010. - http://www.boverket.se/Global/Bygga_o_forvalta/Dokument/Bygg-%20och%20konstruktionsregler/BBR_avsnitt_9/dvut_2009%201_4_dagar.pdf.

Boverket Indata för energiberäkningar i kontor och småhus - En sammanställning av brukarrelaterad indata för elanvändning, personvärme och tappvarmvatten [Rapport]. - Karlskrona : Boverket, 2007. - ISBN: 978-91-85751-65-5.

Boverket Regelsamling för energideklaration med kommentarer [Rapport]. - Karlskrona : Boverket, 2010. - ISBN: 978-91-86342-59-3.

Boverket Regelsamling för energideklaration med kommentarer [Rapport]. - Karlskrona : [u.n.], 2010. - ISBN: 978-91-86342-59-3.

CEN SS-EN 12831 SS-EN 12831: Värmesystem i byggnader - Metod för beräkning av dimensionerande värmebehov [Bok]. - Stockholm : SIS Förlag AB, 2003. - SS-EN 12831:2003.

CEN SS-EN 14511 SS-EN 14511, Luftkonditioneringsaggregat, vätskekylare och värmepumpar, med elmotordrivna kompressorer, för rumsuppvärmning, Del 1, 2, 3, 4 [Rapport]. - Stockholm : SIS Förlag AB, 2007. - SS-EN 14511:1-4.

CEN SS-EN 14511-1--4 Luftkonditioneringsaggregat, vätskekylare och värmepumpar,med elmotordrivna kompressorer, för rumsuppvärmning och -kylning – [Bok]. - Stockholm : SIS Förlag AB, 2007. - SS-EN 14511-1--4.

CEN SS-EN 15232 SS-EN 15232:2007, Byggnaders energiprestanda - Inverkan av byggnadsautomation, styrning och teknisk byggnadsdrift [Rapport]. - Stockholm : SIS Förlag AB, 2007. - SS-EN 15232:2007.

CEN SS-EN 15242 Luftbehandling – Beräkningsmetoder för bestämning av luftflöden i byggnader inklusive infiltration [Rapport]. - Stockholm : SIS Förlag AB, 2007. - SS-EN 15242:2007.

CEN SS-EN 15316-2-1 SS-EN 15316-2-1 Värmesystem i byggnader - Metod för beräkning av energibehov och systemeffektivitet, Del 2.1 Värmeemmisionssystem för rumsuppvärmning [Rapport]. - Stockholm : SIS Förlag AB, 2007. - SS-EN 15316-2-1.

CEN SS-EN 15316-2-3 SS-EN 15316-2-3 Värmesystem i byggnader - Metod för beräkning av energibehov och systemeffektivitet, Del 2.3 Värmedistributionssystem för rumsuppvärmning [Rapport]. - Stockholm : SIS Förlag AB, 2007. - SS-EN 15316-2-3:2007.

CEN SS-EN 15316-3-3 SS-EN 15316-3-3 Värmesystem i byggnader - Metod för beräkning av energibehov och systemeffektivitet, Del 3.1 Krav på tappvarmvattensystem [Rapport]. - Brussels : CEN, 2007. - SS-EN 15316-3-1:2007.

CEN SS-EN 15316-3-3 SS-EN 15316-3-3 Värmesystem i byggnader - Metod för beräkning av energibehov och systemeffektivitet, Del 3.2 Tappvarmvattendistribution i bostadshus [Rapport]. - Brussels : CEN, 2007. - SS-EN 15316-3-2:2007.

CEN SS-EN 15316-3-3 SS-EN 15316-3-3 Värmesystem i byggnader - Metod för beräkning av energibehov och systemeffektivitet, Del 3.3 Varmvattenberedning i bostadshus [Rapport]. - Brussels : CEN, 2007. - SS-EN 15316-3-3:2007.

CEN SS-EN 15316-4-1 SS-EN 15316-4-1:2008 Värmesystem i byggnader - Metod för beräkning av energibehov och systemeffektivitet. Del 4-1: Värmegenerande system, förbränningsbaserade system (pannor) [Bok]. - Stocholm : SIS Förlag AB, 2008. - SS-EN 15316-4-1:2008.

CEN SS-EN 15316-4-2 SS-EN 15316-4-2:2008 Värmesystem i byggnader - Metod för beräkning av energibehov och systemeffektivitet. Del 4-2: Värmegenerande system, värmepumpsbaserade system [Rapport]. - Stockholm : SIS Förlag AB, 2008. - SS-EN 15316-4-2:2008.

CEN SS-EN 15316-4-3 SS-EN 15316-4-3:2007 Värmesystem i byggnader - Metod för beräkning av energibehov och systemeffektivitet. Del 4-3: Värmegenerande system, termiska solvärmesystem [Rapport]. - Stockholm : SIS Förlag AB, 2007. - SS-EN 15316-4-3:2007.

CEN SS-EN 15316-X-X Värmesystem i byggnader – Metod för beräkning av energibehov och systemeffektivitet [Bok]. - Stockholm : SIS Förlag AB, 2008. - SS-EN 15316-X-X.

CEN SS-EN 255-3 SS-EN 255-3, Luftkonditioneringsaggregat, vätskekylare och värmepumpar med eldriven kompressor - Värmedrift - Del 3: Provning och krav för märkning av aggregat för tappvattenvärmning [Rapport]. - Stockholm : SIS Förlag AB, 1997. - SS-EN 255-3.

CEN SS-EN ISO 13790 SS-EN ISO 13790:2008, Byggnaders energiprestanda – Beräkning av energianvändning för uppvärmning och kylning (ISO 13790:2008) [Bok]. - Stockholm : SIS Förlag AB, 2008. - SS-EN ISO 13790:2008.

CEN TC 113 WG 7 Air conditioners, liquid chilling packages and heat pumps, with electrically compressors, for space heating and cooling- Testing and rating at part load conditions and calculation of seasonal performance [Bok]. - Brussel : CEN TC 113 WG 7, 2009.

DesignBuilder Software Ltd DesignBuilder - Building design, simulation and visualisation - Building Simulation... Made Easy [Online] // DesignBuilder - Building design, simulation and visualisation - Building Simulation... Made Easy. - 2010. - 2010. - http://www.designbuilder.co.uk/.

Energimyndigheten Mätning av kall- och varmvattenanvändning i 44 hushåll [Rapport]. - Eskilstuna : Statens energimyndighet, 2009. - ER 2009:26.

Equa www.equa.se [Online]. - 2010. - den 22 06 2010. - www.equa.se.

EVI Heat Solvärmepumpen -en genialisk lösning [Online] // www.eviheat.se. - 2010. - 2010. - http://www.eviheat.se/pages.aspx?r_id=44255.

Forsén Martin Prestige. - [u.o.] : KTH - Energiteknik, 2004.

Forsén Martin Simulering av värmepumpssystem [Rapport]. - Stockholm : effsys, 2004.

Integrated Environment Solutions IES Virtual Environment [Online]. - 2010. - 2010. - http://www.iesve.com/UK-ROI.

IVT Industrier AB IVT Greenline G, modell 35-45 kW. - Tranås : [u.n.], den 05 Januari 2006.

Javed Saqib Design of ground source heat pump systems - Thermal modelling and evaluation of boreholes [Rapport]. - Göteborg : Chalmers Tekniska Universitet, 2010. - ISSN 1652-6007, Technical Report D2010:02.

Levin Per Brukarindata för energiberäkningar i bostäder [Rapport]. - Stockholm : SVEBY-programmet, 2009. - 2009-04-14.

Naturvårdsverket Beräkna utsläpp av växthusgaser [Online] // www.naturvardsverket.se. - 2010. - 2010. - http://www.naturvardsverket.se/sv/Klimat-i-forandring/Minska-utslappen/Verktygslada-for-kommuner-och-foretag/Berakna-utslapp-av-vaxthusgaser/.

OSO OSO Hotwater Norge [Online] // Optima EP2 300 & 400. - OSO Hotwater, 2010. - 2010. - www.oso.no.

Solar Energy Laboratory TRNSYS 16 - a TRaNsient SYstem Simulation program - Volume 5, Mathematical Reference [Bok]. - [u.o.] : Solar Energy Laboratory, University of Wisconsin-Madison, 2006.

Wemhöner Carsten och Afjei Thomas IEA Heat pump programme Annex 28, Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating [Rapport]. - Borås : IEA Heat Pump programme, 2006. - HPP-AN28-1; ISBN 91-85533-30-0.

Appendix 1 – Enkät 1

I denna enkät efterfrågades huruvida programmen hade möjlighet att beräkna eller presentera specifik information eller specifik systemlösning. Enkäten var indelad på följande delar

- Huset
- Värmesystemet
- Värmegenererande komponenter
- Klimat
- Resultat

Nedan kommer varje fråga för sig att presenteras, först själva enkätfrågan, sedan svaren.

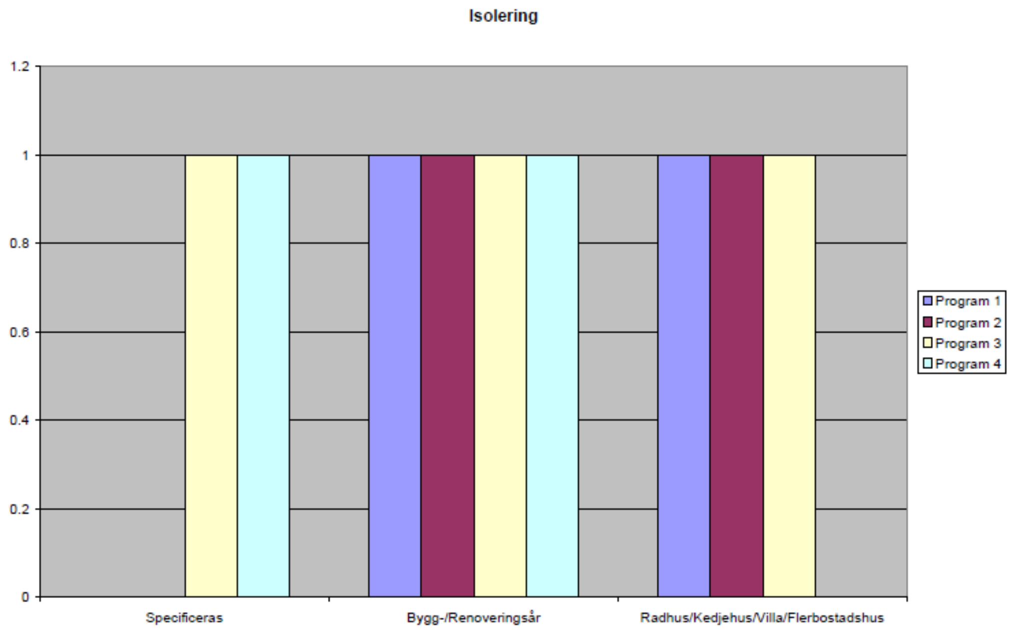
Huset

Huset: Vilket typ av information kräver modellen/programmet

Isolering:

- [ ] Specificeras ($W/m^2 \cdot K$ eller motsvarande)
- [ ] Bygg-/Renoveringsår
- [ ] Radhus/Kedjehus/Villa/Flebostadshus

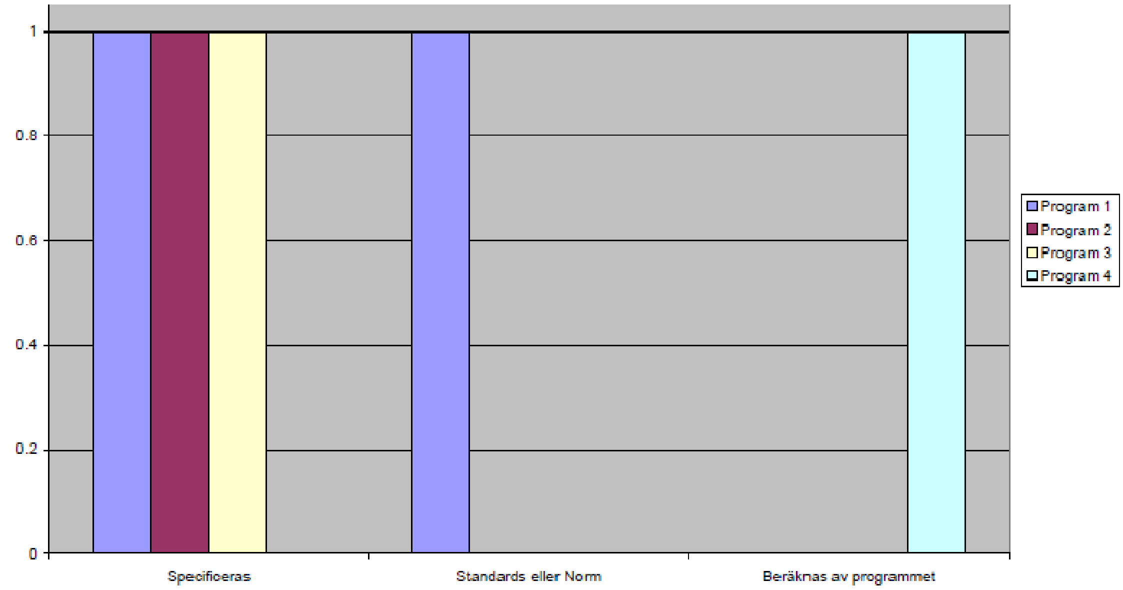
Annat:



Hur hanteras byggnadens energianvändning:

- [ ] Specificeras (t.ex. 215 kWh/m²·år)
- [ ] Standards eller norm (t.ex. BBR eller Passivshus) Ange norm(-er):
- [ ] Beräknas av modellen/programmet

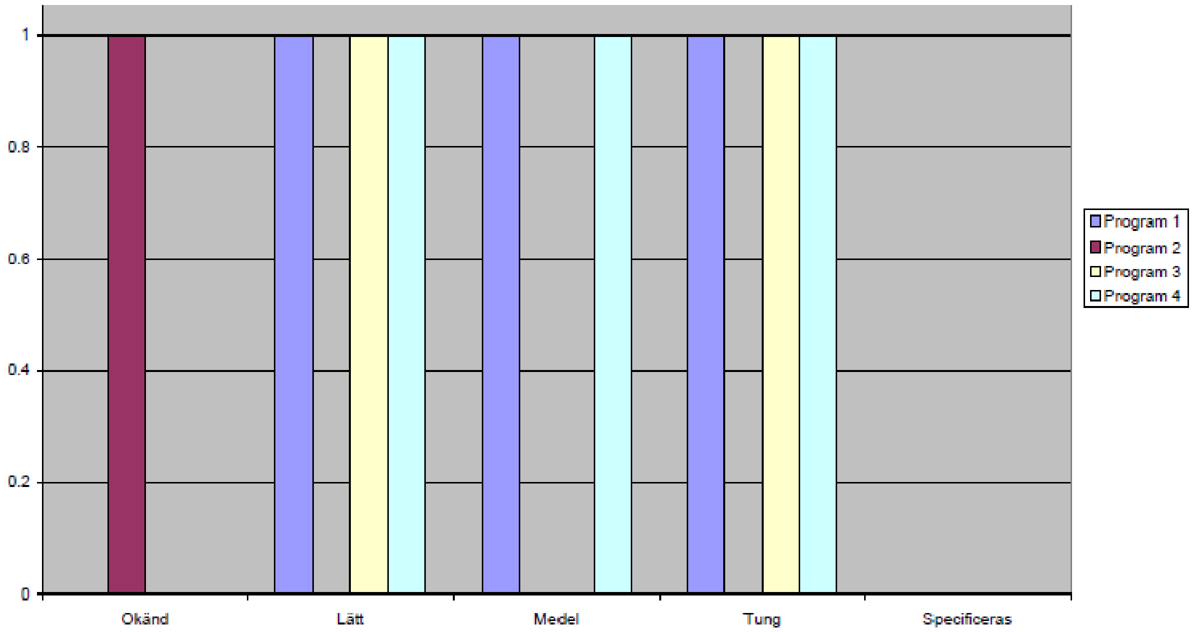
Annat:



Byggnadens karaktär/tidskonstant:

- [ ] Okänd
- [ ] Lätt
- [ ] Medel
- [ ] Tung
- [ ] Specificerad (t.ex. 138 timmar)

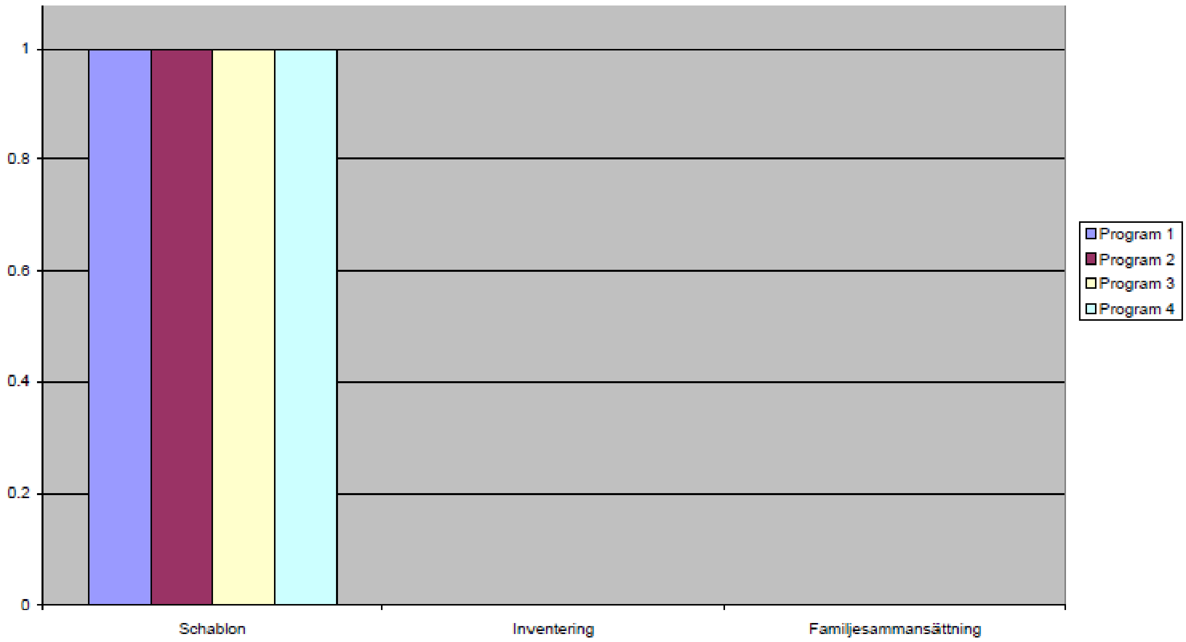
Om okänd, hur hanteras effektdimensioneringen?



Internlaster - Boendebeteende:

- [ ] Schablon (t.ex. 3 °C tillskott)
- [ ] Inventering av komponenter (antal och användning)
- [ ] Familjesammansättning (t.ex. två vuxna och tre barn varav två tonåringar och en av dessa är pojke).

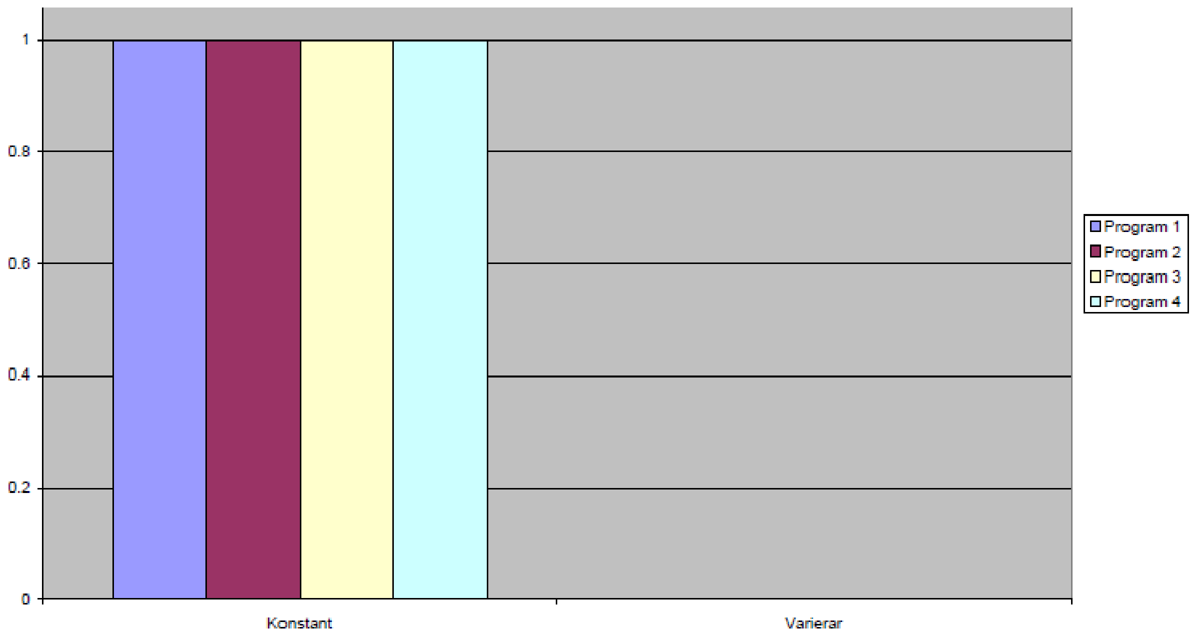
Annat:



Innetemperatur

- [ ] Temperatur sätts konstant, modellen antar således "oändlig" effekttillgänglighet (VP inkl. spets eller kompliment)
- [ ] Temperaturen tillåts variera, beroende på effektbehov och effekttillgänglighet

Annat:



Ventilationssystem

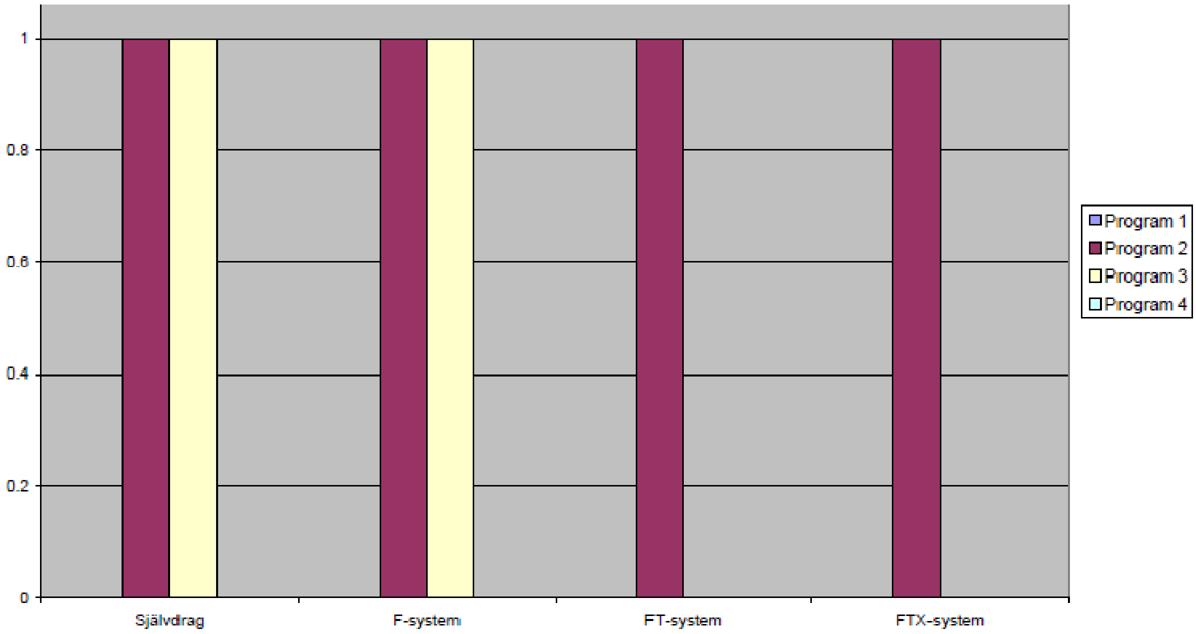
☐ Självdrag

☐ F-system (Mekanisk frånluft + Infiltration)

☐ FT-system (Mekanisk frånluft + Mekanisk Tilluft + Infiltration)

☐ FTX-system (Mekanisk frånluft + Mekanisk Tilluft + Infiltration + Återvinning)

Annat:

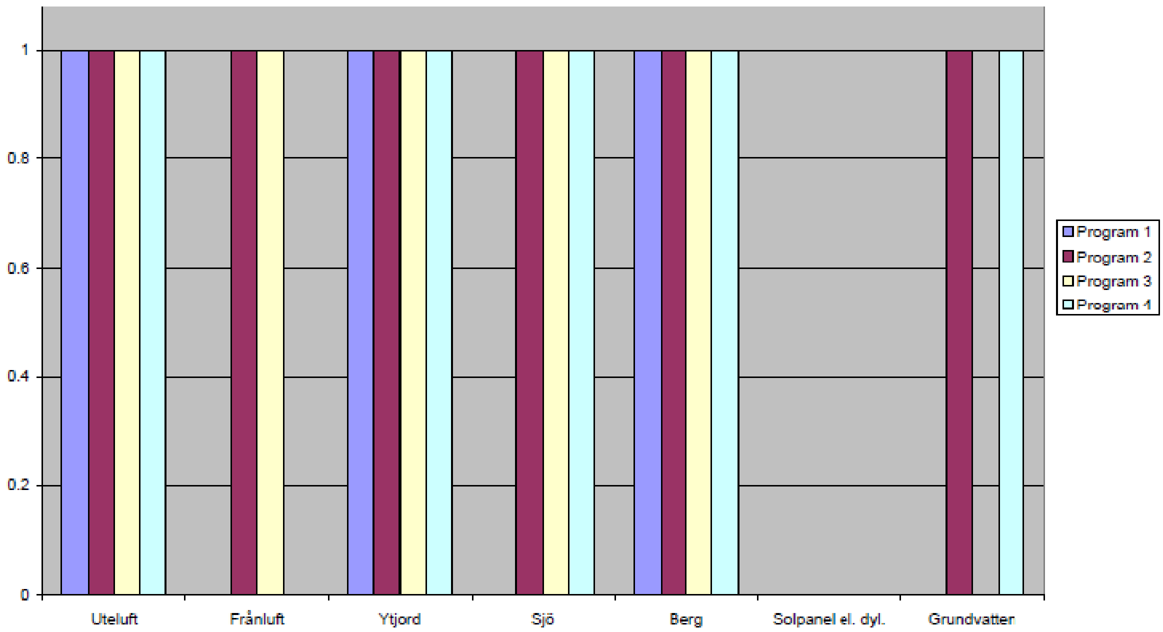


Värmegenererande komponenter

Värmekälla

- [ ] Uteluft
- [ ] Frånluft
- [ ] Ytjord
- [ ] Sjö
- [ ] Berg
- [ ] Solpanel el. dyl.
- [ ] Grundvatten

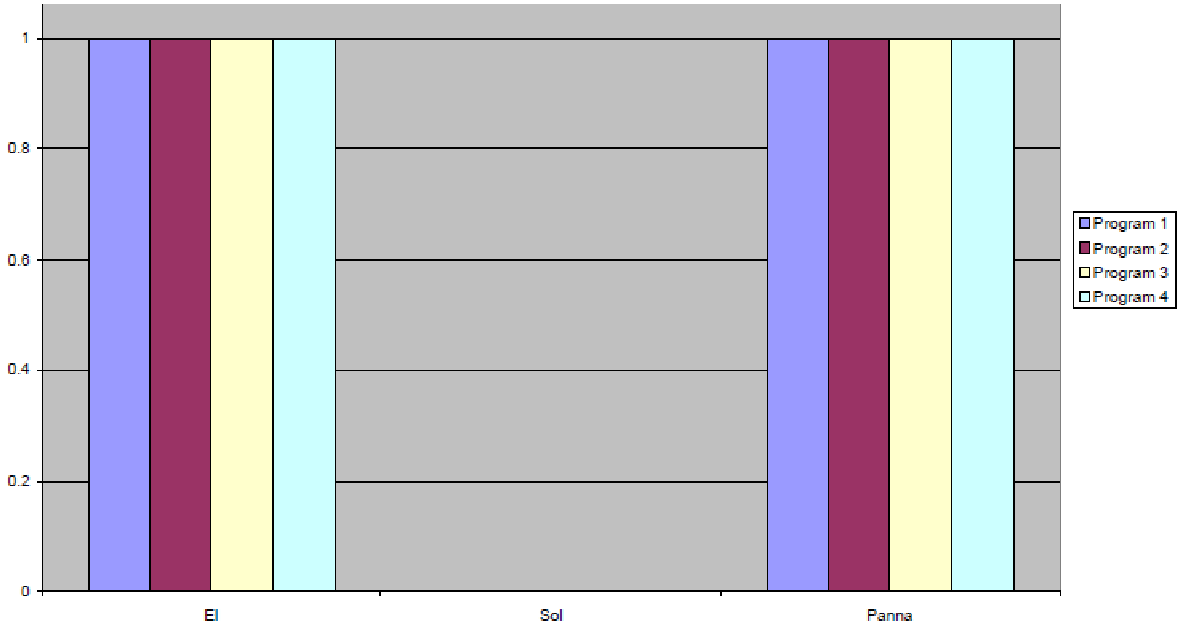
Annat:



Kompletterande energikälla (även spets)

- [ ] El
- [ ] Sol
- [ ] Panna (Olja, Pellet, osv.)

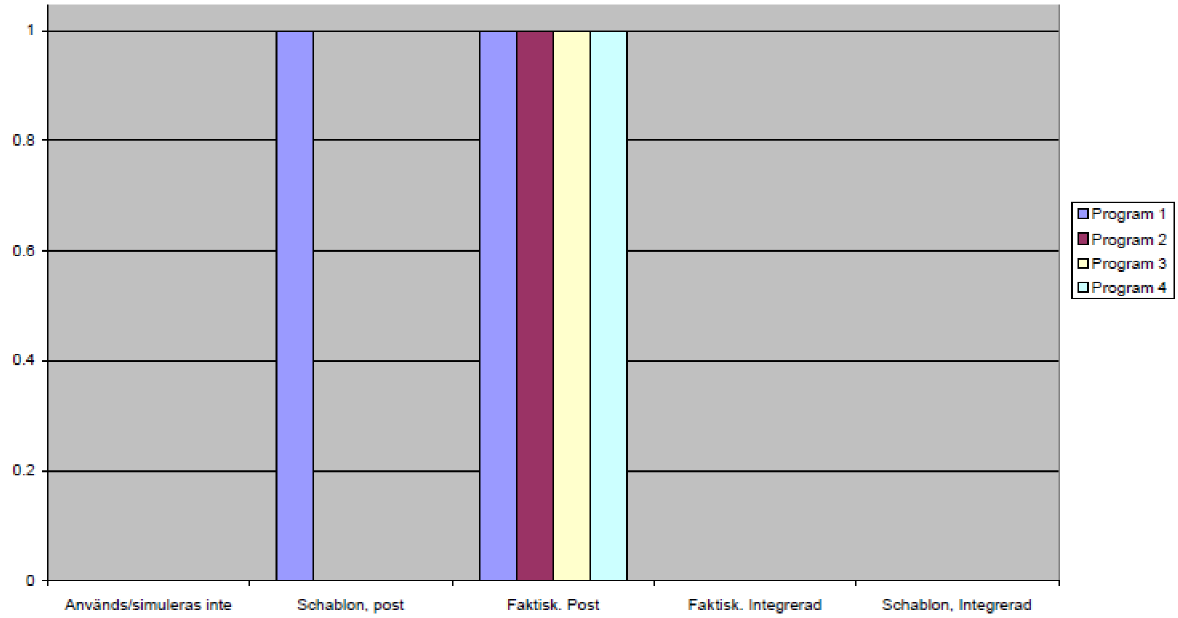
Annat:



Tappvattenvärmning

- [ ] Används/simuleras inte
- [ ] Schablon, läggs till efter simulering av året som kWh enbart
- [ ] Faktisk förbrukning, läggs till efter simulering av året som kWh enbart
- [ ] Fördelas enligt kundens beteende över tiden, VP går i TVV-mode
- [ ] Fördelas enligt schablon (t.ex. IEA/ECBCS Annex42) , VP går i TVV-mode

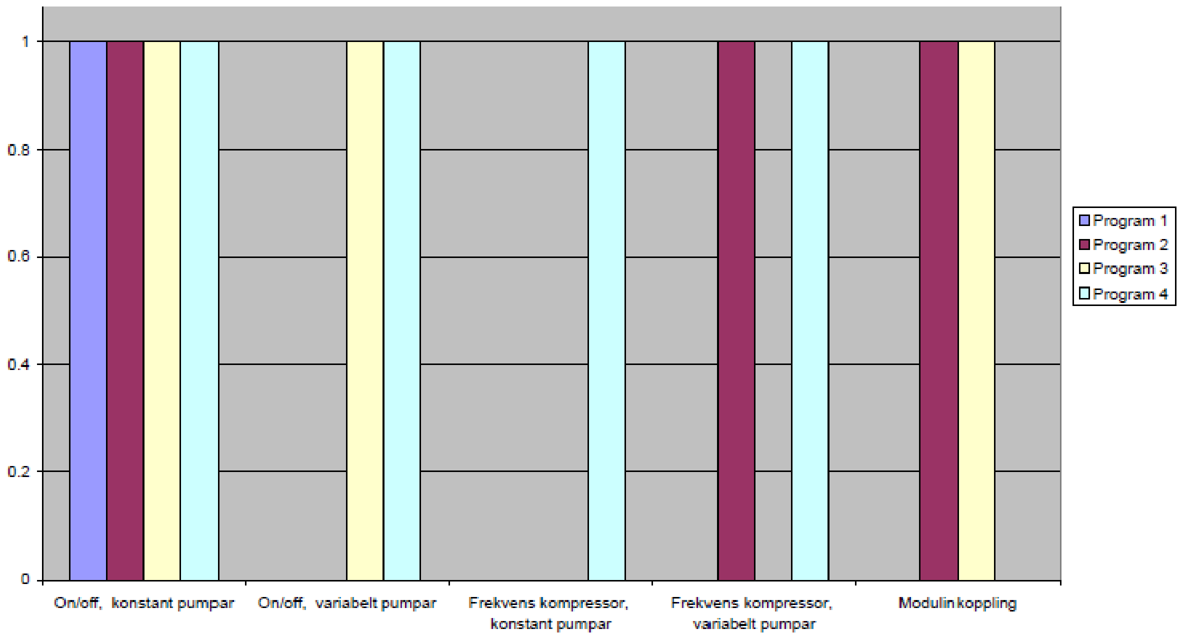
Annat:



Kapacitetsreglering av värmepumpenheten

- [ ] On/off kompressor, konstant varvtal pumpar
- [ ] On/off kompressor, variabelt varvtal pumpar
- [ ] Variabelt varvtal kompressor, konstant varvtal pumpar
- [ ] Variabelt varvtal kompressor, variabelt varvtal pumpar
- [ ] Modulkopplade värmepumpsenheter (stegvis inkoppling)

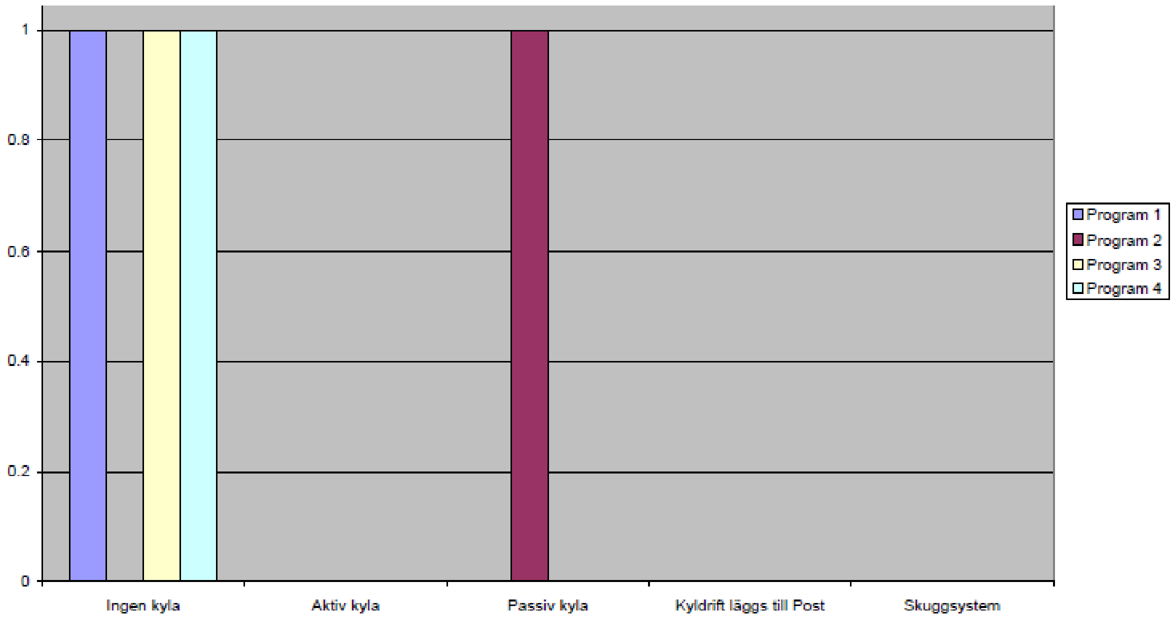
Annat:



Kyldrift

- ☐ Värmepumpen kan inte leverera kyla
- ☐ Värmepumpen kan leverera kyla
  - ☐ Aktiv kyla
  - ☐ Frikyla
- ☐ Kyldriften läggs till efter årssimulering av systemet
- ☐ System för minska solinstrålning tas hänsyn till (t.ex. markiser)

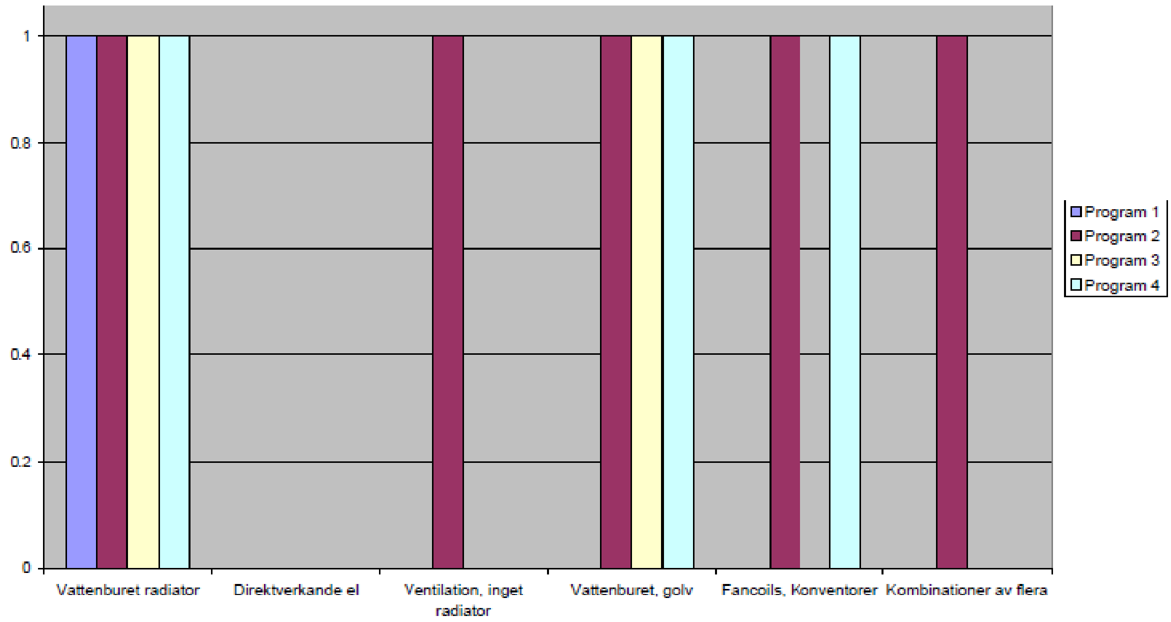
Annat:



Värmesystemet

Radiatorsystem

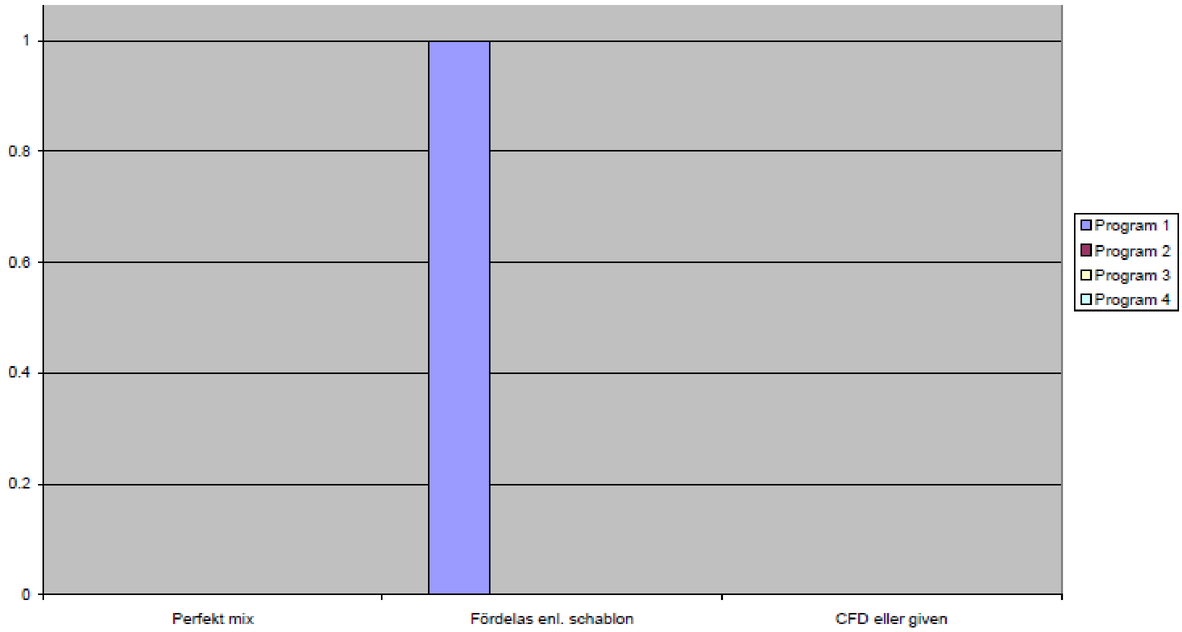
- ☐ Vattenburet radiatorsystem
- ☐ Direktverkande el
- ☐ Ventilationsuppvärmt (inget radiatorsystem)
- ☐ Vattenburet golvvärme
- ☐ Fancoils/Konvektorer
- ☐ Kombinationer av ovan kan också användas?



Luft/Luft Värmepump, hur hanteras husets planlösning

- ◯ Antar perfekt mixing av luften
- ◯ Fördelar energin enligt schablon
- ◯ Fördelar energin enligt CFD eller annan förutbestämd fördelning

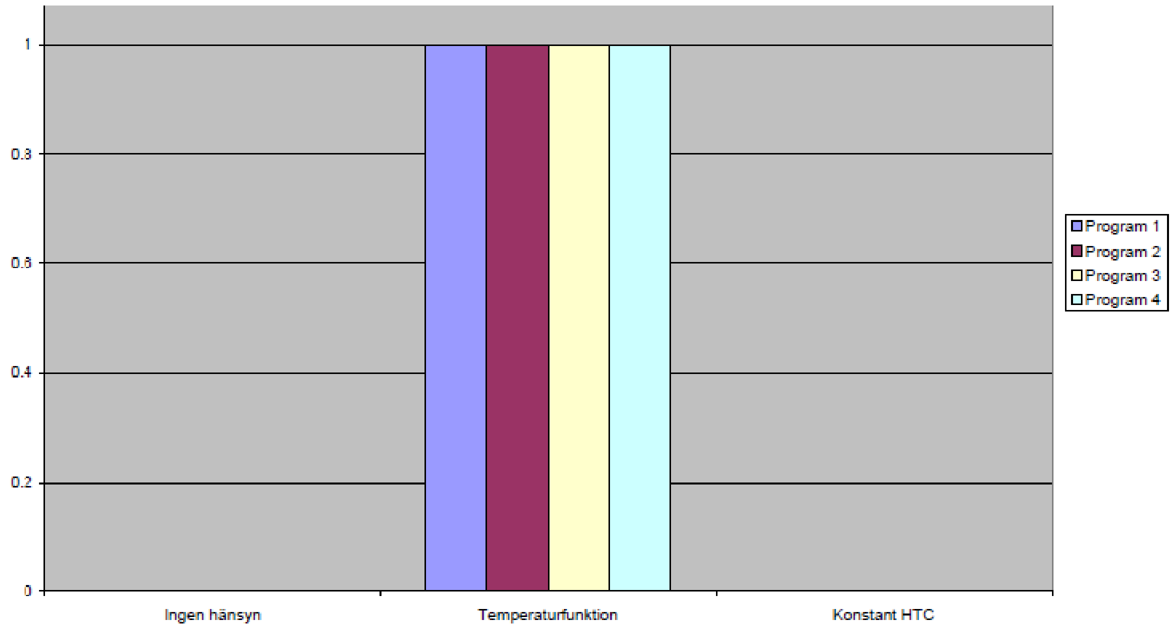
Annat:



Storlek på radiatorsystem

- ◯ Tar inte hänsyn till detta, antar oändligt stora
- ◯ Tar hänsyn till detta, antar effekt - temperaturfunktion utifrån utlagd temperaturnivå vid DUT (t.ex. 55/45)
- ◯ Antar konstant värmeövergångstal

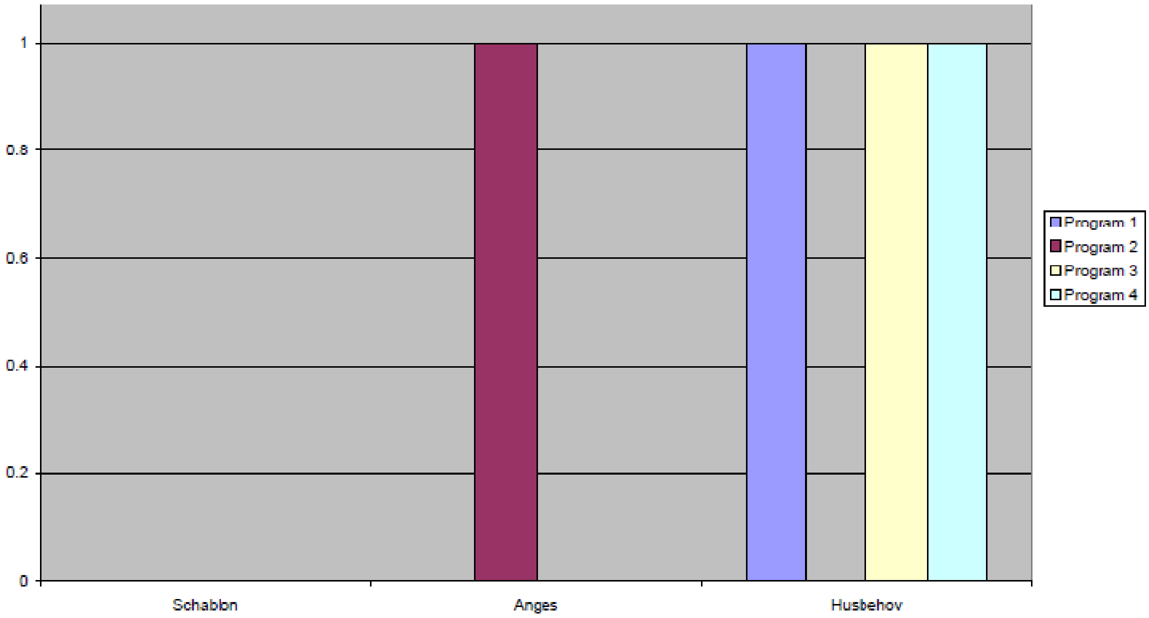
Annat:



Hur bestäms temperaturkurvan för värmesystem, dvs framledningstemperaturen som funktion av utetemperatur

- Schablon
- Anges
- Beräknas i simulering enligt husets/radiator-behov

Annat:



Klimat

Klimatdata

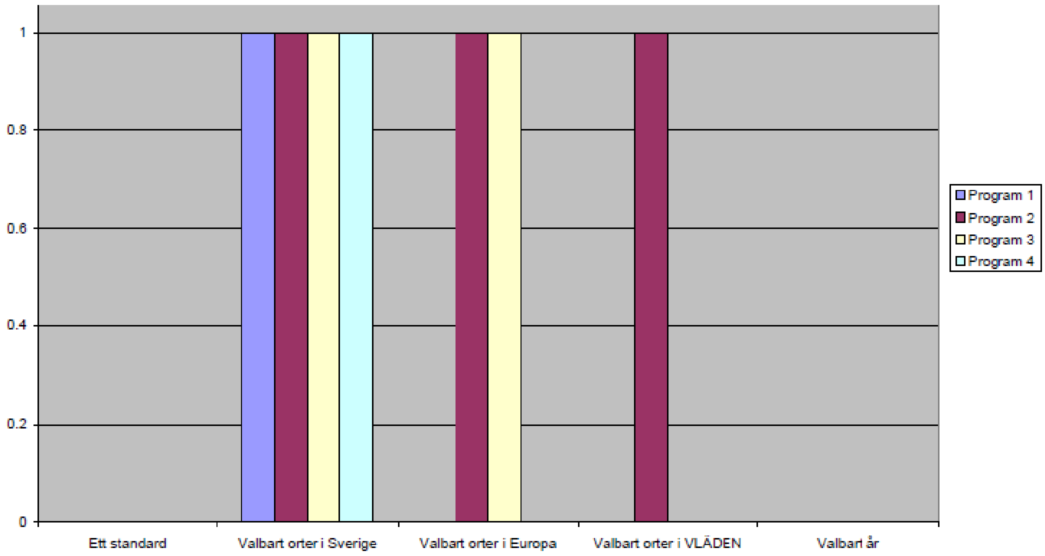
- ☐ Ett standardklimat
- ☐ Standardklimat för flera olika **svenska** städer, valbart
- ☐ Standardklimat för flera olika **Europeiska** städer, valbart
- ☐ Standardklimat för flera olika **internationella** städer, valbart
- ☐ Valbart klimatår

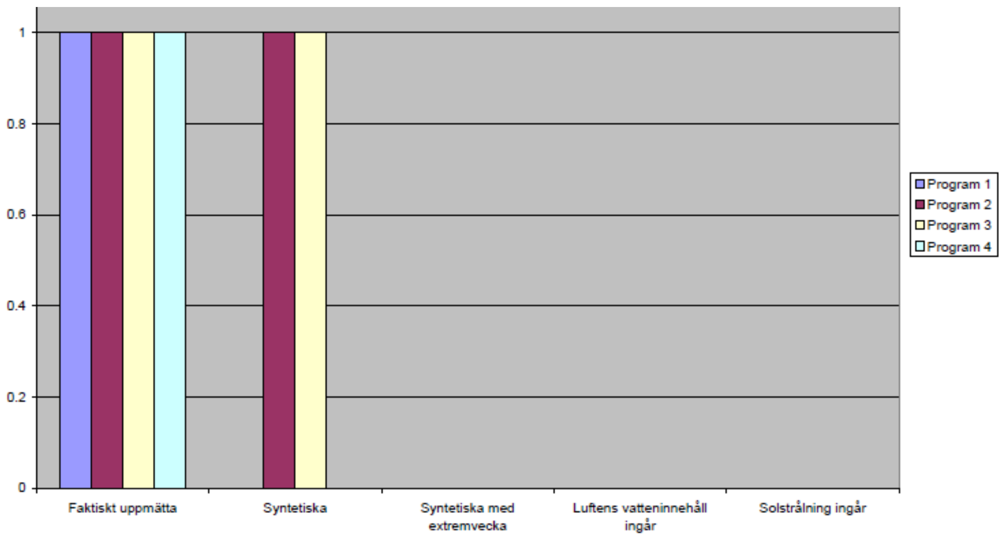
Klimatdatagrund

- ☐ Faktiskt uppmätta data
- ☐ Syntentiska data
- ☐ Syntentiska data med extremvecka

- ☐ Luftens vatteninnehåll ingår i klimatdata (våt temperatur/daggpunkt/relativ fuktighet/vatteninnehåll)
- ☐ Solinstrålning ingår i klimatdata

Annat:





Resultathantering

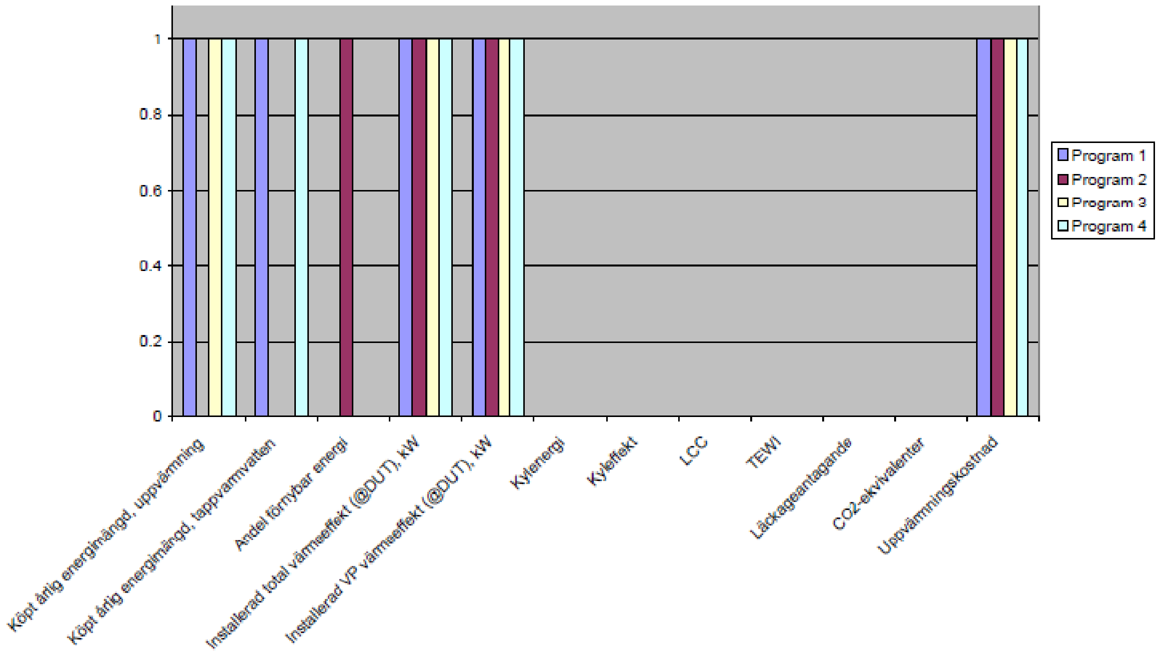
Resultat:

Simuleringsresultat

- [ ] Köpt årlig energimängd för uppvärmning (el, pellets, olja, osv.)
- [ ] Köpt årlig energimängd för tappvarmvatten (el, pellets, olja, osv.)
- [ ] Andel förnybar energi (Sol, luft, mark, pellet, osv)
- [ ] Installerad (totalt) värmeeffekt (vid DUT eller annat tillstånd) (kW)
- [ ] Installerad värmeeffekt från värmepumpen (vid DUT eller annat tillstånd)
- [ ] Kylenergibehov (kWh)
- [ ] Kyleffektbehov (kW)
- [ ] LCC (Life Cycle Cost)
- [ ] TEWI (Klimatbelastning) / LCA (Life Cycle Analysis)

Läckageantagande

- [ ] CO2 ekvivalenter (Klimatbelastning) (årsbasis eller livscykel)
- [ ] Uppvärmningskostnad (per år) i kronor



Appendix 2 – Enkät 2

Fyra olika hus har simulerats av olika tillverkarna i sina egna beräkningsprogram. Nedan presenteras resultaten

Hus 1:

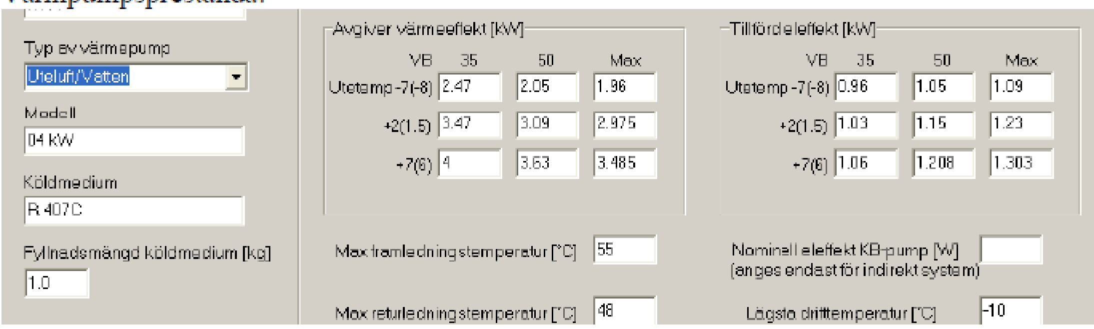
Indata:

124 m², Fristående 1.5 plan, ingen källare, ej inredd vind.
Stockholm, Bromma.
Vattenburen värme (45/35).
Innetemperatur 22°C.

Uteluft VP med eltillsats (95% verkningsgrad).

13 500 kWh för uppvärmning (värmeenergi).
4 500 kWh för tappvarmvatten.

Värmpumpsprestanda:



Utdata:

Tillförd energi (el) till kompressor (inkl. varmvattenproduktion):____________________kWh

Tillförd energi (el) till tillsats:____kWh

Hus 2 :

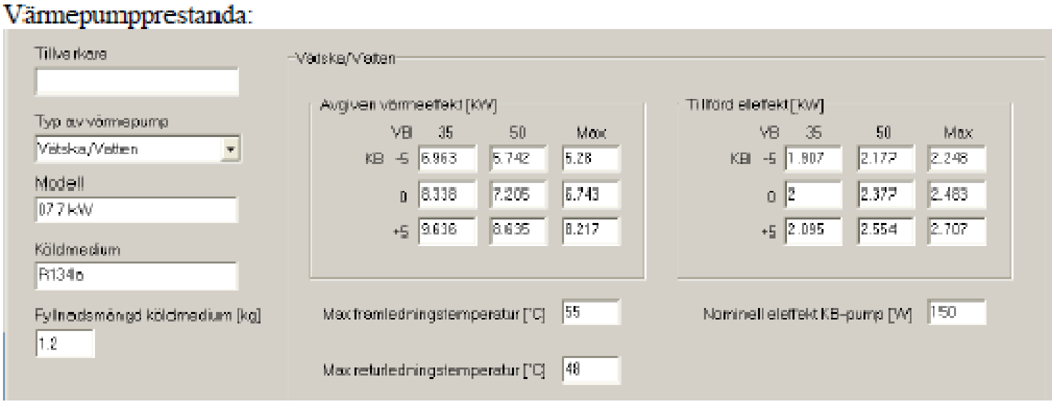
Indata:

176 m², Fristående, byggår 1966, Tung byggnad (150timmar tidskonstant), 2 plan, med inredd källare, ej inredd vind.
Luleå
Vattenburen värme (55/45)
Innetemperatur 22°C

Bergvärme med eltillsats (95% verkningsgrad). Berg Granit med 3.4 W/mK. Medeltemperatur på Brine -3°C

31 500 kWh värmebehov (värmeenergi).
4 500 kWh varmvattenbehov

Värmepumpprestanda:



Utdata:

Tillförd energi (el) till kompressor (inkl. varmvattenproduktion):____________________kWh

Tillförd energi (el) till tillsats:____kWh

Borrhålsdjup:____m

Hus 3:

Indata:
153 m², Radhus, byggår 1970, Medeltung byggnad (95 timmar tidskonstant), 2 plan, ingen källare, ingen vind.
Nybro
Direktverkande elradiatorer
Innetemperatur 22°C

Luft/Luft värmepump, elradiatorer tillsats (100% verkningsgrad)

19 890 kWh värmebehov (värmeenergi).
3 500 kWh varmvattenbehov

Värmepumpprestanda:

Uteluft (°C)	**Värme (kW)**	**Kompressor (kW)**
7	5	3.2
2	3.4	2.6
-7	3	2.4
-15	2.8	2.3

Utdata:

Tillförd energi (el) till kompressor (inkl. varmvattenproduktion):____________________kWh

Tillförd energi (el) till tillsats:____kWh

Hus 4:

Indata:

180 m², Fristående, byggår 2009, Lätt byggnad (24 timmar tidskonstant), 2 plan, ingen källare, ingen vind.

Borås

Vattenburen golvvärme (35/28)

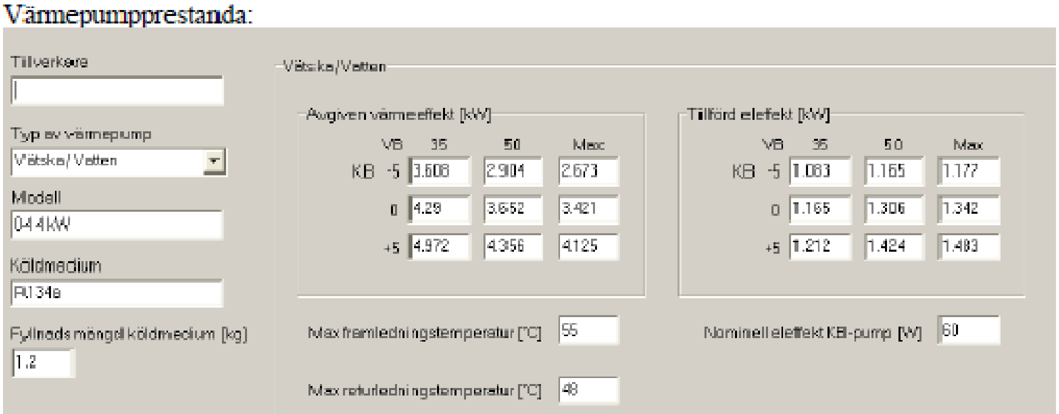
Innetemperatur 22°C

Mark/Vatten värmepump, eltillsats (95% verkningsgrad)

19 800 kWh värmebehov (värmeenergi).

4 500 kWh varmvattenbehov

Värmepumpprestanda:

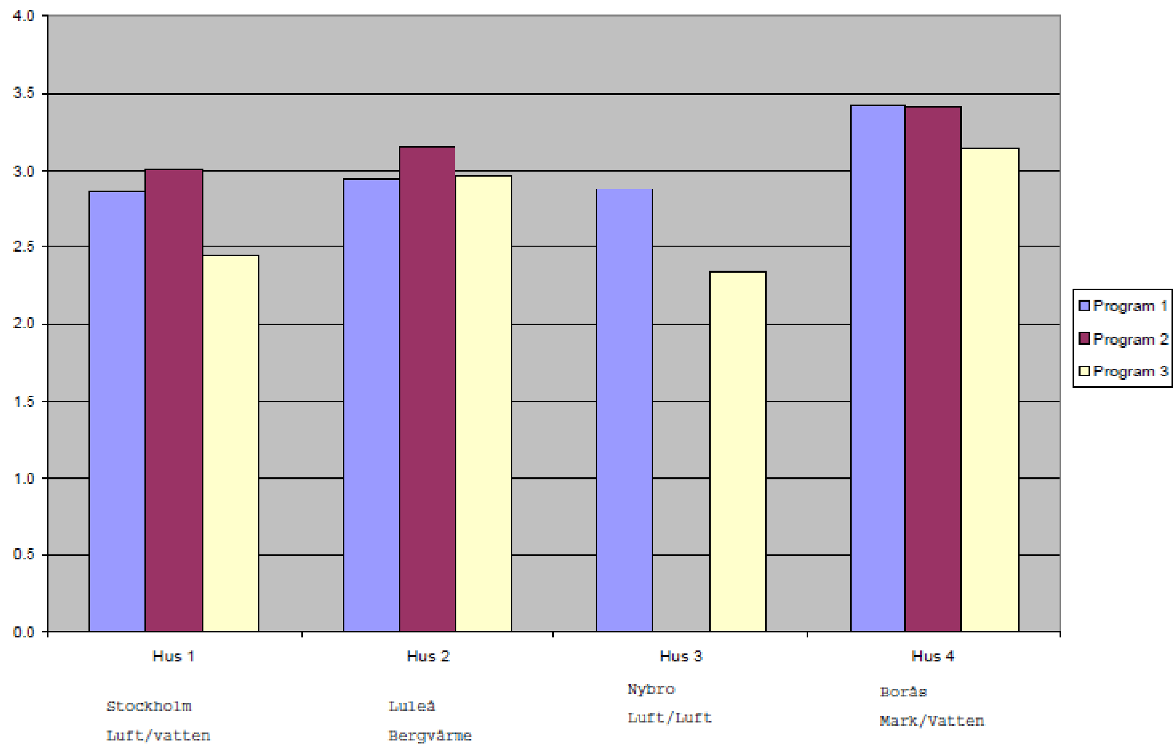


Utdata:

Tillförd energi (el) till kompressor (inkl. varmvattenproduktion):____________________kWh

Tillförd energi (el) till tillsats:____kWh

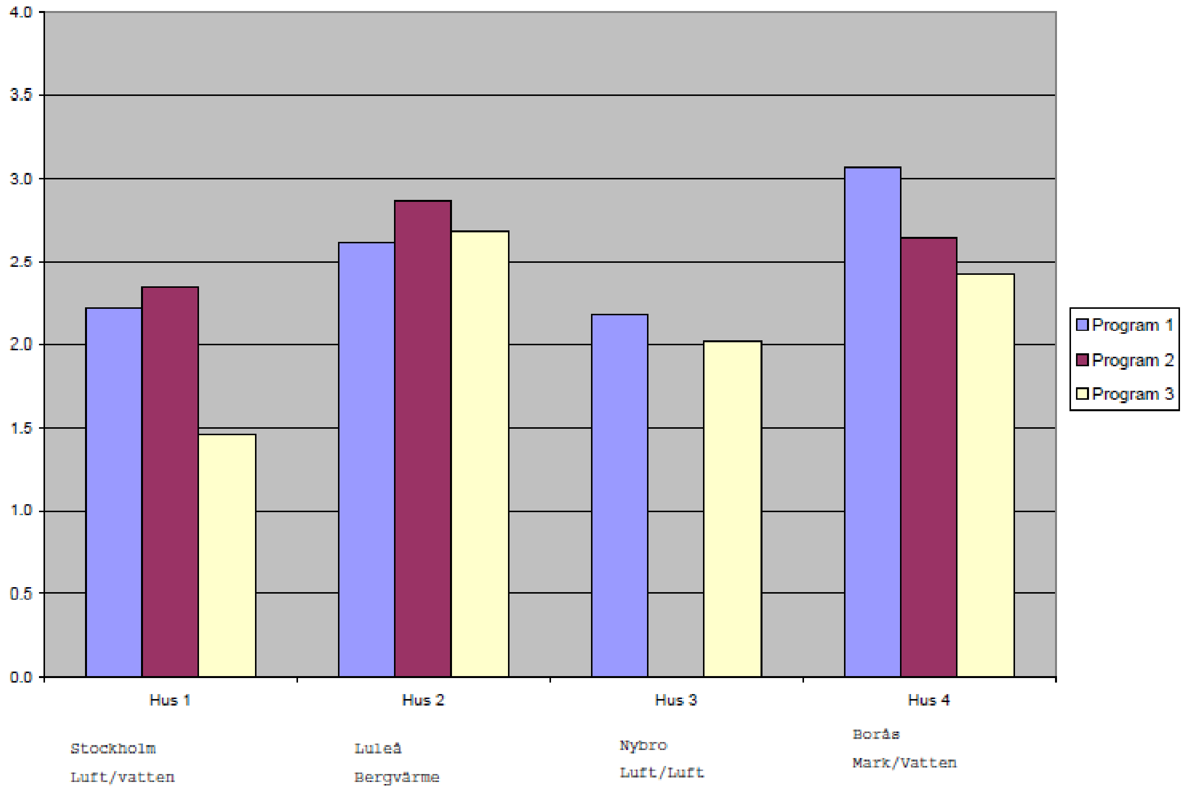
Slinglängd:____m



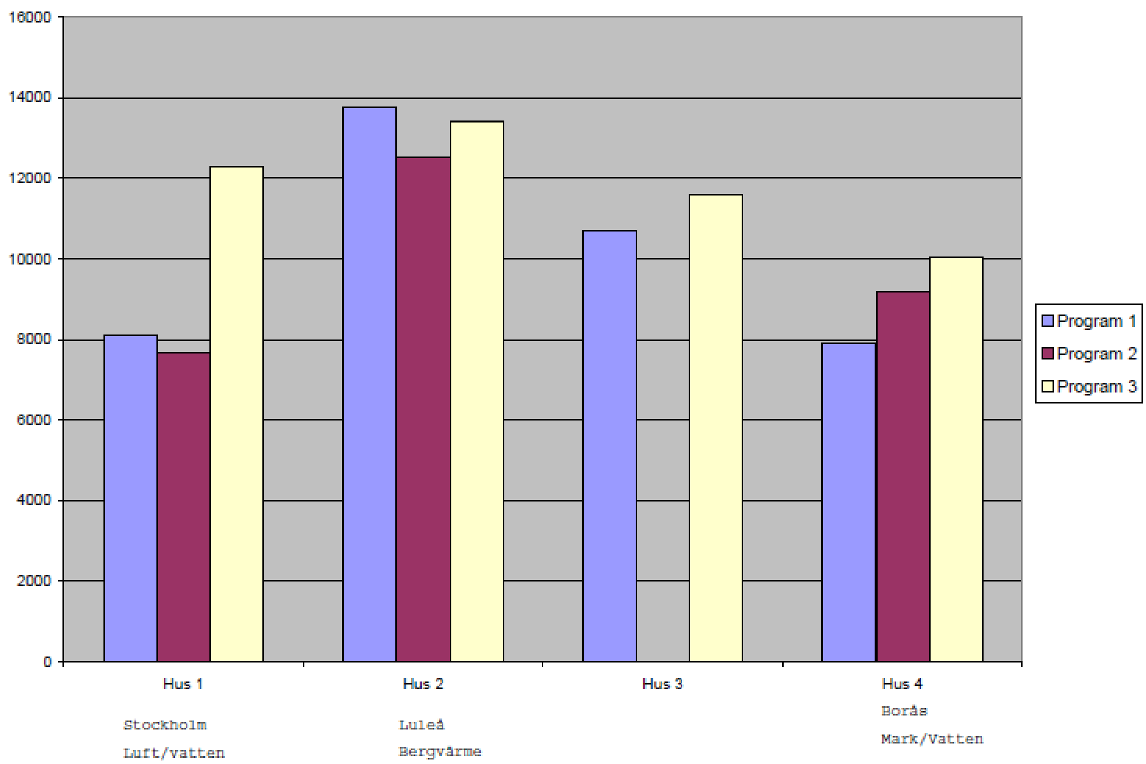
Figur 28: SPF för värmepumpsenheten.

Figur 28 visar värmepumpsenhetens årsvärmefaktor. Det är tydligt att viss variation mellan de olika programmen råder. Det skall noteras att Program 3 tog hänsyn till avfrostning två gånger, då prestandafilen redan innehåller den informationen. Avvikelsen är alltså inte så stor som det synes i Figur 28. Det är ingen egentlig systematik kring skillnaderna förutom att just Program 3 ligger i underkant på alla fyra husen. Avvikelsen är i storleksordningen mindre än 10 %.

Figur 29 visar hur årsvärmefaktorn för de olika husen varierar mellan de olika programmen. Eftersom Program 3 räknade med avfrostning två gånger är skillnaden nu ännu större. Noterbart är den stora skillnaden för hus 4, med en ytjordsvärmepump. I övrigt är det inga signifikant skillnader. Motsvarande information syns i Figur 30, men i form av köpt energi.



Figur 29: SPF för värmesystemet



Figur 30: Köpt energi.

Appendix 3 - Studentarbeten

På efterföljande sidor bifogas två stycken studentarbeten som utförts vid KTH – Energiteknik.

KTH Industrial Engineering and Management

Comparison between three different heat pump sizing comptuter program concerning SPF and usability

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MJ2409 Applied Energy Technology PROJECT course

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TABLE OF CONTENTS

1 OBJECTIVES

When installing a heat pump heating system, it is often of interest to know the amount of energy saving that will be achieved. The SPF (seasonal performance factor) is an index that gives information about the behaviour of a particular heat pump installed into a particular building in terms of energy consumption during a full season, say a year. Such an index can be used to compare different available solutions both for energetic and economic purposes.

Heat pump manufacturers basically follow the same frame for the SPF calculation procedure; however there are some differences in their approach, which are the target points of our assessment. Each manufacture has developed its own software tool in order to get the SPF value of a system based on the heat pump, the building and the weather conditions prevailing where the system is located.

In this project, an effort has been made to systematically compare the tools used by some heat pump manufacturers for calculation of SPF.

The outcome of this project would be a comparison between three software packages in terms of the features that characterize each program. Based on qualitative comparison of different SPF calculation tools, conclusions will be presented at the end in order to propose needs for further development.

2 SCOPE

2.1 AVAILABLE TOOLS

The programs we got access to are VITOCALC, PRESTIGE and VPW2100. In order to get these software packages we have directly contacted several manufacturers and institutions and we have included into our assessment the ones we got allowance to use. We did not get access to tools developed by other manufactures mainly because of the distribution remains internal among the technicians of the manufacturers. All the software packages we use are covered by Copyright..

2.2 APPROACH

A quantitative approach would be achievable under different conditions of homogeneity.

A first difficulty in starting a quantitative comparison is the fact that the applications have already built-in characteristics for the heat pumps available for choosing. Another limitation we have to deal with is that the software source codes are not available and the time the group members can spend discussing with the developers of the software program or specialists is limited. This fact often results in difficulties to understand the assumptions behind the calculations performed by the programs. Thus, even if the same heat pump, in term of performance characteristic, would be available to choose, the results of the energy calculations given by different programs would be different.

A quantitative assessment of the software tools that would lead to investigate the goodness of the assumptions behind the calculations and their accuracy is therefore beyond the scope of this project.

2.3 OUTCOMES

Our assessment results in a comparison of the features of the computer programs and their options in term of input and output data, level of sophistication, ease and clearness of use.

Suggestions for further improvement of the analyzed tools will also be given.

When it is possible and necessary, in order to achieve these goals an effort is made in order to figure out what are the assumptions and the calculations the programs are based on.

3 INTRODUCTION

3.1 DEFINITIONS

3.1.1 GENERAL INTRODUCTION

Heat Pumps can be divided into different categories based on their [2]:

- Heat source
- End use (heat sink)
- Cycle
- Energy drive
- System configuration
- Operation mode (functionality)

Heat source

Heat pumps can use air, water, or ground as their heat source.

Ground source and water source heat pumps have a higher coefficient of performance (COP) due to the fact that ground temperatures (below 2.5 meters) are relatively constant and high all year round, while outside air temperatures can vary a lot.

End use

Heat pumps can deliver different services such as:

- Space heating (SH)
- Domestic hot water (DHW)
- Air conditioning (AC) - Space cooling (SC)

Cycle

Heat pumps can use either a vapor compression cycle (mechanically driven), or an absorption cycle (thermally driven).

Energy drive

Heat pumps can use electricity, fuel (natural gas, propane, diesel oil), or high-temperature waste heat as their driving energy.

System configuration

In a monoenergetic heat pump system, HP almost covers the total annual heating requirement (usually 90-95 %), and the rest is supplied by a back-up heater (Figure 1Figure 1)

Heat pumps can also be used in a bivalent configuration in which two separate heat sources are combined to provide the total annual heating requirement (Figure 2)

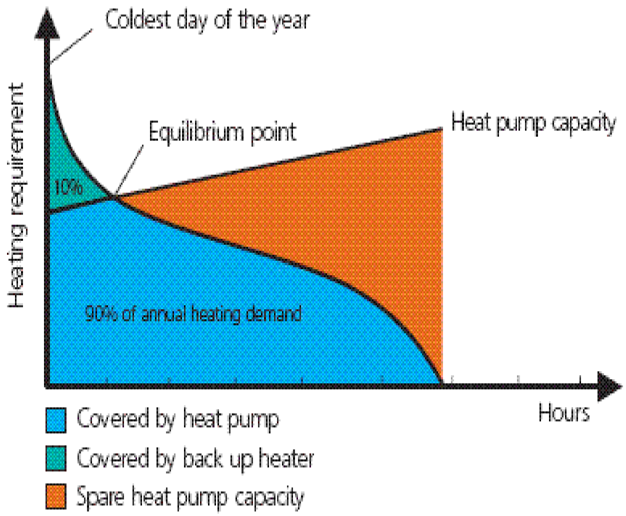


FIGURE 1 MONOENERGETIC SYSTEM

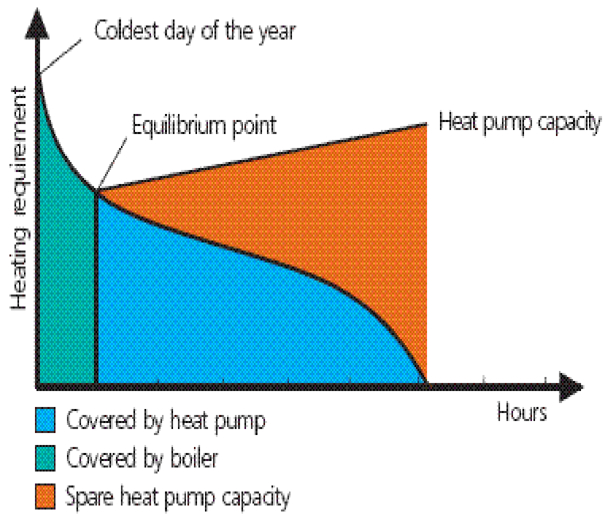


FIGURE 2 BIVALENT SYSTEM

Source:[www.gsmlimited.com/energy/flexibility.html]

As a rule of thumb a heat pump, installed in Sweden, covering 50% of the needed power covers 85% of the required energy. [5]

As it can be seen in the Figure 1and Figure 2, the capacity of heat pump balances the heating demand at equilibrium point.

Bivalent systems can run in a parallel mode in which HP and the supplementary heat source operate in parallel at temperatures lower than equilibrium point, or they can operate in an alternative mode in which either the HP or supplementary heat source operates at a time.

Bivalent alternative systems are common for air source heat pumps where HP operation is stopped at low ambient temperatures (usually lower than -10 to -15°C), and the supplementary heat source will supply all the heating demand during this period. [5]

Operation mode

Heat pumps can have different functionalities, such as:

- Space heating only mode
- Water heating only mode
- Combined Space cooling and Water heating mode
    - Alternate operation (HP is switched between SC and DHW operation)
    - Simultaneous operation of SC and DHW
- Combined Space heating and Water heating mode
    - Alternate operation (HP is switched between SH and DHW operation)
    - Simultaneous operation of SH and DHW

Each mode has its own operating conditions affecting the overall performance of the heat pump system and its related performance indicators.

3.1.2 SEASONAL PERFORMANCE FACTOR

Usually heat pumps are provided by the manufacture with COP values. Depending on the availability of data, those values are given only for particular working conditions.

COP itself does not express the total amount of energy required over a heating season. In fact, COP is an index defined for the HP itself and does not consider the building and heat distribution system in which it works.

The SPF is defined in such a way that permits to compare the energy consumption of different heat pumps installed in a particular building.

SPF is used to compare different heating systems in terms of energy consumption, emissions, and operation costs. Energy or quality labels can, also, be given to a system based on its SPF. Similarly, it shows the quality of a product, which might be considered to allocate subsidies or funding. [3]

The operating performance of an electric heat pump over the season is called the seasonal performance factor (SPF).

It is generally defined as the ratio of the heat supplied by the heat pump during a period to the total energy consumption (usually electricity) of the heat pump and its auxiliary devices during the same period. An exact formulation of the seasonal performance factor is given in(EQ.1):

$$SPF = \frac{\int_{t=t_0}^{t=t_{end}} \dot{Q}_{used} dt}{\int_{t=t_0}^{t=t_{end}} \dot{E}_{input} dt} \qquad \textbf{(EQ.1)}$$

Where,

Q_{used} = all forms of useful energy as output of the system

- E_{input} = sum of all power input to the system

t_0= beginning of the period

t_{end} = end of the period

dt = time step

Depending on the boundary that is going to be chosen, different types of SPF can be defined.

Figure 3 shows a comprehensive system boundary for a residential heat pump with combined operation of space heating and domestic hot water. [3]

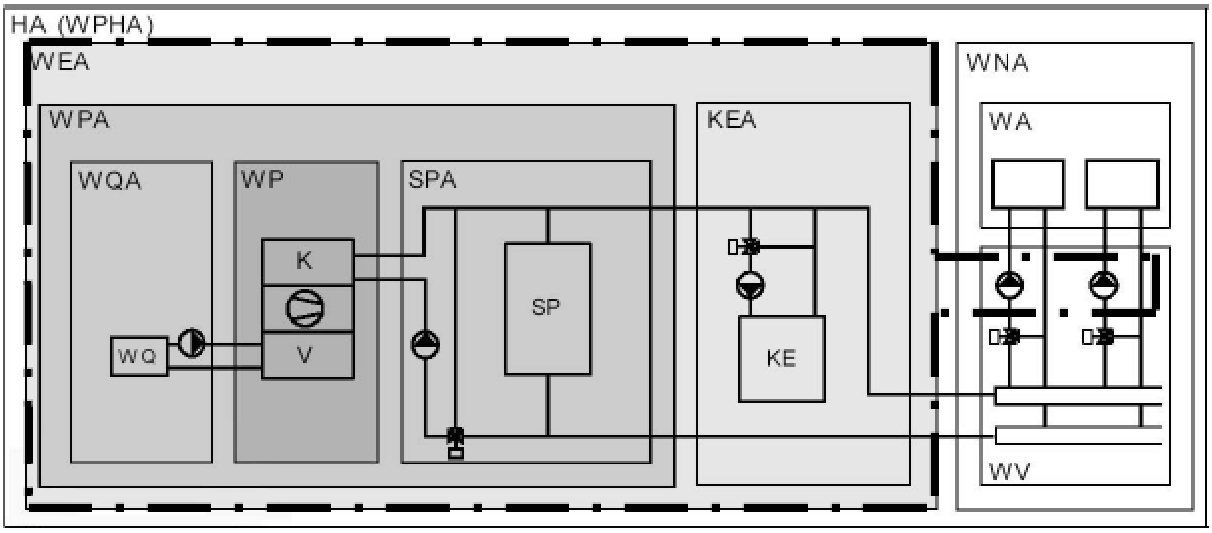


FIGURE 3 SYSTEM BOUNDARY DEFINITION FOR A RESIDENTIAL HEAT PUMP WITH COMBINED OPERATION OF SH AND DHW (ADAPTED FROM FHBB METHOD, SWISS FEDERAL OFFICE OF ENERGY (SFOE), 2003)

Where,

HA (WPHA) = heat pump (heating system)

WEA = heat generation system (heat pump system and back-up)

WPA = heat pump system (including buffer storage)

WQA = heat pump source system

WP = heat pump

SPA = storage system

KEA = back-up system

WNA = heat utilization system (heat distribution and heat emission)

WV = heat distribution system

WA = heat emission system

Seasonal performance factor for a monovalent heat pump (SPF_{hp}), which corresponds to the boundary of "heat pump system" (WPA) including the circulation pump can be written as:

$$SPF_{hp} = \frac{Q_{bl} + Q_{dhw}}{E_{hp} + E_{aux}} \qquad \textbf{(EQ.2)}$$

Where,

SPF_{hp} = seasonal performance factor of the heat pump (without consideration of back-up)

Q_{bl} = energy requirement of the building

Q_{dhw} = domestic hot water energy requirement

E_{hp} = electrical energy input to heat pump

E_{aux} = additional auxiliary energy input (source pumping, sink pumping, etc.)

To calculate the overall seasonal performance factor (SPF_{sys}) the system boundary corresponds to the boundary of "heat production system" (WEA) (dotted line in Figure.3), i.e. the generation part including the source, the storage and the back-up system, while the distribution and emission systems (WNA) are excluded here.

Equation.3 gives the overall seasonal performance factor (SPF_{sys}):

$$SPF_{sys} = \frac{Q_{bl} + Q_{dhw}}{\frac{\lambda_{hp,bl} \cdot Q_{bl}}{SPF_{hp,h}} + \frac{\lambda_{hp,dhw} \cdot Q_{dhw}}{SPF_{hp,dhw}} + \frac{(1-\lambda_{hp,bl}) \cdot Q_{bl}}{\eta_{bu,bl}} + \frac{(1-\lambda_{hp,dhw}) \cdot Q_{dhw}}{\eta_{bu,dhw}}} \qquad \textbf{EQ.3}$$

Where,

SPF_{sys} = overall seasonal performance factor (with consideration of back-up heater)

$SPF_{hp,h}$ = seasonal performance factor for space heating only mode

$SPF_{hp,dhw}$ = seasonal performance factor for water heating only mode

Q_{bl} = energy requirement of the building

Q_{dhw} = domestic hot water energy requirement

$\lambda_{hp, bl}$ = fraction of building energy requirement covered by heat pump

$\lambda_{hp, dhw}$ = fraction of domestic hot water energy requirement covered by heat pump

$\eta_{bu, bl}$ = efficiency of the space heating back-up heater

$\eta_{bu, dhw}$ = efficiency of the domestic hot water heating back-up heater

SPF calculation tools may take into account the variable heating demands, the variable heat source and heat sink temperatures over the year and according to the system boundary include the energy demand for peripheral devices, such as [1 & 2]:

Ground water pump

- Brine circulation pump
- Auxiliary heater (not usually included in SPF calculations unless specified [5])
- Fans
- Defrosting system
- Control devices (on-off compressors or variable speed compressors)

3.2 PROBLEM FORMULATION

The software programs considered in this project do not strictly follow a particular standard.

First of all, the SPF calculation methods found in national standards or guidelines present some limitations. This is mainly because they take into account either only space heating or an alternative operation of space heating and hot water production. ASHRAE 137 is the only existing standard which takes into account the combined operation of SH and DHW but only for air-to-air heat pumps. [3]

Furthermore, a layout strictly based on a standard probably would result in a software program very difficult to use: the input data required by the standards are usually very detailed and they are seldom available to the customers.

However, some of standards for SPF calculation have been investigated in order to figure out what are the main issues involved.

Table.1 shows the characteristics and limitations of some important national standards used for the calculation of SPF. [3]

Standard	Scope	Limitation
VDI 2067-6 (detailed version) [6]	Electrically driven Combustion engine driven Air-to-water Ground-to-water Water-to-water	- Arbitrary operation (test) points - Cyclic operation at part load not considered - No explicit consideration of auxiliary energy, storage, distribution losses - COP values are based on manufacturers' data (not based on standard test results) - Input and output temperature requirements of the hot water are not considered
HTA Lucerne (detailed version) [7]	Electrically driven Air-to-water Ground-to-water	- Changing of source temperatures in the case of ground-coupled heat pumps not considered - Restriction to monovalent systems (no bivalent system is considered) - Cyclic operation at part load not considered
CEN/TC 228/WG 4 [8]	Electrically driven Air-to-water	- Difficult calculations, since every bin has to be calculated - No transparent correction factors for the user (partly empirical or taken from measurements) - No hot water calculation is considered yet (draft version) - Correction with default degradation

		coefficient for consideration of part load - Consideration of auxiliary energy with constant fraction dependent on output capacity
ASHRAE 116 [9]	Electrically driven Air-to-air	- SPF calculation for heating only mode - Testing method is different from the European testing results (different testing standard) - No explicit consideration of auxiliary energy - No COP evaluation (only the evaluation of electrical energy consumption)
DIN 4702-8 [10]	Adaption of performance calculation methods for boilers to heat pumps	- Underlying meteorological data are fixed for a standard site and cannot be changed - Correction factors for auxiliary energy and storage losses are not considered - Not suited for the assessment of system performance (system boundary limited to the generator- SPF)

TABLE 1 CHARACTERISTICS AND LIMITATIONS OF SOME IMPORTANT NATIONAL STANDARDS FOR SPF CALCULATION (ADAPTED FROM FHBB METHOD, SWISS FEDERAL OFFICE OF ENERGY (SFOE), 2003)

As mentioned in the previous section, combined operation has a significant impact on the SPF and the heating capacity of the heat pumps over a season. Therefore, there need to be a widely accepted easy-to-use method developed to estimate the SPF of heat pumps based on currently available standard procedures of calculation and measurements. [3]

What necessitate the development of a common method of calculation are briefly the following considerations [4]:

- HVAC designers need to compare different available systems
- Costumers need to compare the amount of energy and money saved by using different products
- There should be regulations governing the data of the products of manufacturers
- There should be a basis for the consultants and policy makers

Dissimilar results from different methods are because of the following factors:

- Different levels of precision of the existing methods
- Different levels of sophistication of the existing methods
- Variation in outdoor condition

- Variation in the heat source condition
- HP load share (sizing of the HP in relation to the heat demand)
- Different system boundaries
- Different control systems
- Different installations
- Maintenance conditions

As a result of more effective building insulation, losses are diminished providing higher accessible temperatures for DHW applications. Consequently, the ratio of energy requirement of supplying domestic hot water (DHW) to that of space heating (SH) is increasing, emphasizing on the need for standard assessment tools for systems providing combined DHW and space heating. [3]

In combined operation, in which HP produces heat for both DHW and space heating, it cannot be easily indicated that how much electricity is consumed for each purpose. Nevertheless, by defining the combined operation mode as a whole, SPF calculation procedure is simplified for this working condition. [4]

3.3 MAIN CHARACTERISTICS OF THE ASSESSED SOFTWARE TOOLS

The software tools assessed in this project are: PRESTIGE 2.0, VITOCALC 2007 v2 version 5.0, and VPW 2100. The possibility to extend the assessment to other software packages has been considered but the access to them has turned out to be limited.

PRESTIGE, developed by the Swedish Heat Pump Association, SVEP, has been developed by cooperation between different manufacturers and KTH University.

The PRESTIGE's approach has been followed to create VPW 2100, but the manufacturer has developed it in many aspects to meet its requirements and improve the abilities of the program. In particular, VPW 2100 permits to choose among the products of the IVT's catalogue. It has to be noticed that a reduced version of this program, conceived for customers, is available on the IVT's web page.

VITOCALC 2007 has been developed by the VIESSMANN's technicians. The heat pumps available in VITOCALC 2007 are the ones present in VIESSMANN's catalogue.

All these three software programs have been developed to be used by the resellers and technicians of the manufacturing companies. In particular, they can be used for both initial economic and technical estimation. A more reliable assessment can be done by using more detailed input data once the installation of the heat pump has been proved feasible.

4 ANALYSIS

In this chapter the main characteristics of the assessed software packages are presented. All the features of each software tool, except economic parts, have been investigated thoroughly to find out the effect of each parameter on the final result. This investigation have consisted several combinations of different weather data, building characteristics, heat pumps, operation temperatures, and each program's specific parameters, mentioned under the corresponding heading.

4.1 PRESTIGE 2.0

4.1.1 WEATHER DATA

Location of the house can be chosen from a list of cities. The program uses its built-in database to estimate the weather condition.

4.1.2 BUILDING CHARACTERISTICS

It should be specified whether the building is light, medium, or heavy. Area of the heated space, construction year of the building and form of the house should be also given to the software program. There are eight different forms for the house: with/without cellar and with/without attic in one or two stories. In addition, the type of the house should be chosen from the following two groups: 1- villa, detached or linked house 2- terrace-house or semi-detached house.

There is a box which can be checked to allow the program to use the default value of energy use for the building.

4.1.3 TEMPERATURES

Required indoor temperature set point and the supply and return temperatures to and from radiators at design outdoor temperature are to be determined by the user.

4.1.4 ANNUAL ENERGY REQUIREMENT

Energy consumption of the current heating system and efficiency of the current heating device is used to estimate the required energy of the building. Possible options to enter data are electricity, oil, wood, combination of electricity and oil, combination of electricity and wood, and combination of oil and wood. For electricity, the annual consumption in kWh/year should be entered. For wood and oil consumption is to be indicated in cubic meter per year, so the program assumes the heating value and calculates the amount of energy.

Heating required for DHW can be estimated based on the number of people living in the building or entered directly by the user. In the former case, the program takes 2500kWh/year as basic value and adds 500kWh/year for each resident.

Then a window appears called control. It shows the design outdoor temperature chosen by the program, annual energy requirement of the building, annual energy required for DHW, and heating load at DOT, Design Outdoor Temperature. Besides, there appears a bar chart comparing the current annual energy consumption of the building per square meter, calculated by dividing the value entered by the user by the area of the building, and average value.

4.1.5 Choosing heat pump

There are four types of heat pump to choose based on the heat source: rock, soil, outdoor air, and air/air.

The two latter ones do not work in the coldest days of the year normally; an auxiliary heating system is necessary, then.

If rock is the heat source, only two types of granite can be chosen and the average temperature of the rock should be given to the software. The rest of the parameters needed for calculations are assumed by the software.

If soil is the heat source, its type should be specified by good, mediocre, and bad. It is, also, required to enter the mean temperature of the soil.

There is a drop-down menu to choose the manufacturer, but there is only one option, standard, to select. Then the capacity of the heat pump can be chosen from another drop-down menu.

User should choose the auxiliary system, which can be built-in electrical cartridge, electricity, oil, or wood, and its efficiency.

4.1.6 Heat pump calculations

Result of the calculations are categorized as energy saving, energy utilization, and also a summary of given data in a window similar to Figure 4.

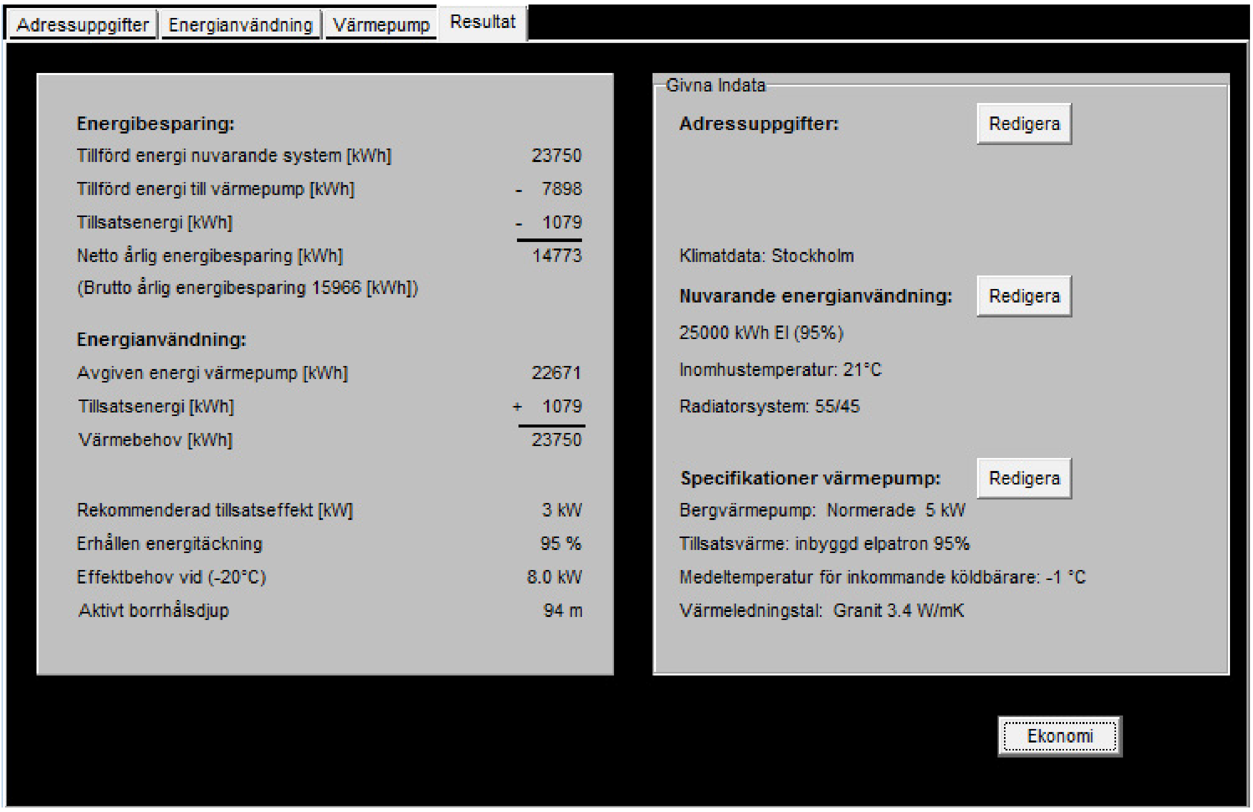


FIGURE 4

4.1.6.1 ENERGY SAVING

Current annual energy requirement of the building is calculated by multiplying the current annual energy consumption, indicated by the user previously, by the efficiency of the current heating system. Then both the energy supplied to the heat pump and the output of the auxiliary system are subtracted from the energy requirement to find the net energy saving in kWh/year.

$$\begin{aligned} &\textit{net energy saving} \\ &\quad = \textit{current annual energy required} - \textit{heat annual pump energy consumption} \\ &\quad - \textit{output of auxiliary unit} \end{aligned}$$ **EQ. 4-a**

$$\begin{aligned} &\textit{net energy saving} \\ &\quad = \textit{current annual energy consumption} \times \eta - \textit{heat pump annual energy consumption} \\ &\quad - \textit{consumption of auxiliary unit} \times \eta \end{aligned}$$ **EQ. 4-b**

Gross annual energy saving is calculated by taking consumptions into account. That is, it is calculated by subtracting both the energy derived from the heat source by the heat pump and the consumption of the auxiliary system from current annual energy consumption of the building.

$$gross\ energy\ saving = \frac{current\ annual\ energy\ required}{\eta} - heat\ pump\ annual\ energy\ consumption - \frac{output\ of\ auxiliary\ system}{\eta}$$

EQ. 5-a

$$gross\ energy\ saving = current\ energy\ consumption - heat\ pump\ annual\ energy\ consumption - consumption\ of\ auxiliary\ system$$

EQ. 5-b

4.1.6.2 ENERGY UTILIZATION

Output of the heat pump, $\int Q_1\,dt = \int Q_2 dt + \int W dt$, and output of the auxiliary system are calculated and it is shown that they meet the energy requirement of the building together.

The recommended heating power of the auxiliary system, for heating loads close to peak load, percentage of required energy covered by the chosen heat pump, and recommended heating power at DOT are also calculated. Furthermore, if the heat source is soil, recommended loop length in m, and if the heat source is rock, the depth of borehole in m is calculated.

Seasonal Performance Factor, SPF, can be easily calculated by dividing annual energy output of the heat pump by its annual energy consumption.

4.1.7 ECONOMIC CALCULATIONS

The software has another feature that investigates economic aspects of using heat pump. However, it is beyond the scope of this study.

4.2 VITOCALC

4.2.1 WEATHER DATA

Annual mean temperature and design outdoor temperature are to be entered by the user. Besides, in the Swedish version, climate values can be roughly found by pointing at a map of Sweden or chosen from a list of cities, and then they can be adjusted for different altitudes.

4.2.2 BUILDING CHARACTERISTICS

Building characteristics is estimated based on the construction year of the building. In Swedish version, there are three types of buildings, “heavy”, “medium” and “light”, affecting the DOT.

4.2.3 TEMPERATURES

Required indoor temperature set point is to be determined by the user.

The supply and return temperatures to and from radiators at design outdoor temperature are presumed to be known, there is also an option to choose between floating condensing and fixed condensing when the supply and return temperatures are fixed by the user. If these two temperatures are not known to the user, they can be estimated based on the current outdoor temperature and measured supply and return temperatures.

4.2.4 ANNUAL ENERGY REQUIREMENT

Required energy for space heating can be estimated in three ways:

4.2.4.1 "ESTIMATE (POWER)"

Present energy source (oil, electricity, gas, wood, or district heating) and the efficiency of the heating device, except in case of district heating, and the power demand in kW is taken as input. Type of present energy source does not change the estimated annual energy demand. It, still, is important in economic considerations. Number of flats and the amount of energy required to supply DHW is also entered as input. Multiplication of number of flats and DHW energy demand gives the total energy demand for supplying DHW. That is, number of flats does not change the space heating energy requirement.

The energy annually required for space heating is calculated based on the weather data, building characteristics, indoor temperature set point, power demand, and number of flats. Then the "net energy use" is calculated by adding the calculated annual energy required for space heating to the DHW energy demand.

4.2.4.2 "PRESENT"

Current energy source (oil, electricity, gas, wood, district heating, or combined boiler) is determined by the user.

In case oil, gas, or wood is currently used, number of consumed cubic meters per year is to be determined in addition to the boiler efficiency. Based on these data and the heating values taken from the built-in database of the program, "net energy use", which includes annual energy required for both space heating and DHW, is calculated in kWh/year. User may change the calculated value of net energy use manually, but the program does not use the manually entered value in further calculations. Displaying the calculated net energy use value in an inactive box could prevent confusion of the user. Then the share of DHW is to be determined by the user in kWh/year.

In case that electricity is the current energy source of the building, the current annual energy consumption of the building is taken as an input and multiplied by the efficiency of the heating device, entered by the user, to find the "net energy use", which includes annual energy required for both space heating and DHW, in kWh/year. Then the share of DHW is to be determined by the user in kWh/year.

Since in some houses there is only one measuring device for electricity consumption, the value read on the meter includes consumption of all of the electrical appliances normally used. It would be convenient to have the possibility to set the number of typical energy-consuming appliances (fridges, washing machines…) for estimating the Space heating and DHW consumption, which may not be known to the user.

If the building presently uses district heating, the current annual energy consumption of the building entered by the user, since no efficiency is applied, is used directly as the "net energy use", which includes annual energy required for both space heating and DHW, in kWh/year. Then the share of DHW is to be determined by the user in kWh/year.

In case more than one (up to three) energy sources are used, their share and efficiencies of the heating devices, if applicable, can be entered under combined boiler option.

4.2.4.3 "Estimate (area)"

The user indicates whether the building is presently heated by using oil, electricity, gas, wood, or district heating. Although it does not change the estimated annual energy demand, it influences economic calculations, which lies beyond the scope of this project. The efficiency of the heating device, except for district heating, and the area of the heated space are to be entered, as well.

In addition, the use of the building should be chosen between the following options: house/housing in combination with work rooms, Caretaking, hotels and restaurants, school, offices, sports, food shop, other stores, and remaining.

Based on the annual mean temperature, building characteristics, currently used fuel, area, and the use of the building, "net energy use", including annual energy required for both space heating and DHW, is calculated in kWh/year. Although the user may want to change the indoor temperature manually here, it does not influence the calculations since indoor temperature for each building use is a fixed value derived from statistical data. The share of DHW is determined by the program in kWh/year according to the building use, weather data, and area. It would be convenient if there was an option to enter the share of DHW manually when the exact value is known.

4.2.5 Choosing heat pump and heat source

There are three types of heat pump to choose: air/water, brine/water, and water/water. For each type there are different product models available to choose from a drop-down menu.

Once a particular model of heat pump is chosen, the program calculates the percentages of power and energy demand covered by the heat pump using the built-in database of the products of the manufacturer.

4.2.5.1 Air/water
The only heat source for air/water heat pump is outdoor air.

4.2.5.2 Brine/water
For brine/water heat pumps rock, surface soil, lake, and exhaust air can be chosen as heat source.

In any case, the average temperature of the incoming brine and should be specified, whereas the power of the circulation pump varies by the model of the heat pump automatically.

4.2.5.2.1 Rock
22 different type of rock can be chosen from a drop-down menu. The default value for the thermal conductivity of the rock can be changed manually. Brine temperature change should be entered by the user. There is a setting menu in which borehole diameter, hose outside diameter, hose inside diameter, CC between hoses, thermal conductivity of hose, thermal conductivity of brine, thermal conductivity of ice, kinematic viscosity of brine, thermal capacity of brine, density of brine, and thermal capacity of water can be specified.

4.2.5.2.2 Surface soil
Under the setting menu five different soil types can be chosen, each of which has more details to be specified such as water content, consistency and quartz content of the soil. Thermal conductivity and latent heat of the soil for each type of soil has its own default value, which can be changed manually. Brine temperature change and under setting menu depth in soil and CC between hoses can be specified.

4.2.5.2.3 Lake
Thermal conductivity of the lake bed, brine temperature change, and lake mean temperature should be entered by the user.

4.2.5.2.4 Exhaust air
Incoming and outgoing air temperature and relative humidity and volume flow rate of air either given directly by the user or calculated according to ACH and volume of the room are used to calculate available heat in the exhaust air and amount of condensed water.

4.2.5.3 Water/water
Water/water heat pumps can only have ground water as the heat source. Incoming ground water temperature and brine temperature change are the parameters should be given to the program, whereas the power of the circulation pump varies by the model of the heat pump automatically.

4.2.6 Heat pump calculations
It should be entered that how many percent of the DHW is supplied by the heat pump. Based on the all information given to the program a diagram showing heating power demand in kW versus hours in a year is generated. In the diagram it can be graphically seen

that how much of the demand is covered by the supplementary heating device, how much the energy saving is, and how much the energy consumption is. Percentage of energy demand coverage and percentage of power demand coverage by the heat pump is calculated, as well. Supplementary heating device can be selected between oil, electricity, gas, wood, or district heating; the efficiency also should be provided for the program. At this point, a printable summary, similar to figure5, of the given data and the results including amount of energy saving, share of energy consumption and supplying of the heat pump and the supplementary system, recommended power of the supplementary device, capacity of the heat pump at DOT, and SPF, annual heating factor, is generated.

4.2.7 ECONOMIC CALCULATIONS

VITOCALC has another feature that investigates economic aspects of using heat pump. However, it is beyond the scope of this study.

VIESSMANN **ENERGYCALCULATION**

4/11/2009

CUSTOMER / SALESMAN	**Project**		**Salesman/Installer** Viessmann Värmeteknik AB	
	Name	**Adress**	**Reference person** PA Törnsäter	**Adress** Gunnebogatan 34
	Postcode/postal adress	**Telefone**	**Postcode/postal adress** 163 53 Spånga	**Telefone** 08 - 4748800 (539)

DATA House / Housing in combination with work rooms	Heated area	250 m2	Annual mean temperature	10 °C
	Building		Design outdoor temperature DOT	-16 °C
			Indoor temperature	20 °C
	Energy demand	**38,875 kWh/year**	Condensing	Floating
	of which, hot water demand	5,000 kWh/year	Flow temperature at DOT	55 °C
	Power demand	**24.5 kW**	Return temperature at DOT	45 °C

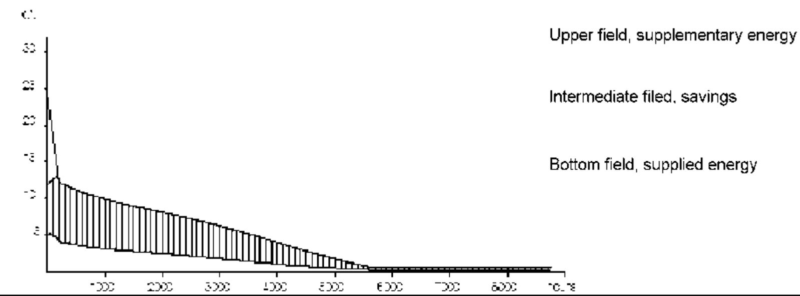
PRODUCT	**1st AWIO 114 350 A**			
			Hot water production from hp	100 %

PREFORMANCE	**Energy demand after installation**	**13,456 kWh/year**	Recommended supplementary power	12.7 kW
	Energy delivered by hp	38,003 kWh/year	Heat pump output at DOT	11.8 kW
	Energy supplied to hp	12,238 kWh/year	Annual heating factor	3.1
	Supplementary energy	1,218 kWh/year	Degree of energy coverage	97 %
	Supplementary energy (Electricity, 95%)	1,282 kWh/year	Dergree of power coverage	48 %
	Savings	**25,500 kWh/year**		

HEAT SOURCE
Outdoor air

ENERGY - DIAGRAM

(Hours of one year with the coldest fist)



VIESSMANN	Viessmann Värmeteknik AB	
	Gunnebogatan 34	Tel 08 - 4748 800
	163 53 Spånga	Fax 08 - 750 60 28

(The calculation is based on details provided and can not be seen as a undertaking)

FIGURE 5

4.3 VPW 2100

4.3.1 WEATHER DATA

There is a drop-down list of 24 European countries plus Australia and Turkey. There is also a list of cities of each country. For Sweden after choosing the city the user should select the location of the house in the city. Weather data used in the program are from METEONORM.

4.3.2 BUILDING CHARACTERISTICS

The house can be detached, terraced, semi-detached, apartment complex, large property, or other kind, as these options appear in a list. Year of construction and the area of the house can also be entered. Type of the house can be selected from another list; it can be one- or two-story building, either with or without furnished cellar, or either with furnished attic or without it. In addition to the aforementioned 8 combinations for house type, it can be 1-story or 2-story hill side house. That is there are 10 options for the house type. Type of the house, year of construction, and area does not influence the calculations; however, if these data are provided, the program, using a template made by SEV, gives the typical consumption for the house, which can be used for checking purposes.

It should be specified whether the calculations are for an existing house or for a new building. Self heating of the house is another parameter should be given to the program; it shows how well-insulated is the building and how much the free internal gains are. As a guideline, the program suggests 6°C for houses built after year 2000 and 3°C for older houses.

4.3.3 TEMPERATURES

The current indoor temperature set point is to be determined by the user.

The supply temperature to radiators and corresponding outdoor temperature should be specified. The maximum supply line temperature corresponds with deign outdoor temperature which can be given to the program by typing DUT in the temperature field.

It is possible to indicate that how much the collected brine from the heat source is warmer than the expected value.

4.3.4 ANNUAL ENERGY REQUIREMENT

Energy demand for hot water is calculated in a same way for both existing and new houses, but method of calculating space heating demand is different for existing houses and new buildings. In either case the user should specify the percentage of power/load coverage by the heat pump.

4.3.4.1 HOT WATER

To estimate the hot water demand, number of the residents in each apartment and the use of DHW selectable between shower, bathtub, or hot tub should be indicated by the user. Default value for DHW energy requirement is the least for showers and the most for hot tubs.

The software program follows the following formula to calculate the energy requirement for DHW [11]:

Hot water requirement = ((2500 + (500*Number of Persons))*Hot water factor)*Number of Apartments

Hot water factor for shower is 1.0, for bathtub is 1.1, and for hot tub is 1.2.

4.3.4.2 New building

Peak heating load is asked in kW. In addition, number of houses and indoor design temperature, based on which the peak heating load is calculated, is asked.

In this method annual energy requirement is calculated based on the peak demand. It can be used for an existing house, the peak heating demand of which is known to the user.

To calculate the heating demand, the effect of wind is not taken into account [1].

4.3.4.3 Existing house

Total current electricity consumption of the house in kWh and the share of household electricity use should be entered. Energy consumption of electrical floor heating and electrical radiators is a part of household electricity use [1]. In case another energy sources currently provide heat along with electricity the program allows the user to choose any combination of oil, wood, gas, or other sources and expects the user to specify the efficiency and consumption of the devices currently using them to produce heat. The user needs to indicate the consumption of oil, wood, or gas in cubic meter per year and the consumption of other sources, such as district heating, in kilowatt-hour per year.

4.3.5 Choosing heat pump and heat source

Regarding the heat source available, one of the main types of the productions of the company, IVT greenline, IVT twin, IVT optima, IVT 490, or IVT premium line X15, should be selected. In each group of products a specific model can be chosen manually; otherwise, the software program does the selection.

4.3.5.1 IVT Greenline

Heat source of IVT greenline heat pumps can be ground soil, rock, or lake as heat source. The user can decide whether the heat pump is to be with or without VBX and with or without FTX. Presence of FTX unit does not change the calculations. There are five different ground conditions, limestone/rock/silt, normal rock/ normal soil, clay, soft rock/dry soil, and hard rock/humid soil, to be chosen. Lake heat also can be chosen. Then the heat source can be selected between geothermal, exhaust air, and groundwater. The temperature of the heat source can be determined automatically by the program or entered manually.

4.3.5.2 IVT twin

Heat sources of IVT twin heat pumps are exhaust air and ground, rock, or lake (two heat sources). There are five different ground conditions, limestone/rock/silt, normal rock/ normal soil, clay, soft rock/dry soil, and hard rock/humid soil, to be chosen. Lake heat can also be chosen. The user should enter the value of exhaust air flow rate in liter per second when the heat source is exhaust air.

4.3.5.3 IVT optima

Heat source of IVT optima heat pumps is outdoor air. The only option is to have a heat pump with FTX or not. Presence of FTX unit does not change the calculations.

4.3.5.4 IVT 490

Heat source of IVT 490 heat pumps is exhaust air. The user should solely enter the exhaust air flow rate.

4.3.5.5 IVT PREMIUM LINE X15

Heat source of IVT premium line X15 heat pumps, which are variable speed, can be ground, rock, or lake. The user can decide whether the heat pump is to be with or without VBX unit and with or without FTX. Presence of FTX unit does not change the calculations. There are five different ground conditions, limestone/rock/silt, normal rock/ normal soil, clay, soft rock/dry soil, and hard rock/humid soil, to be chosen. Lake heat also can be chosen.

4.3.6 HEAT PUMP CALCULATIONS

Finally the program will find the best heat pump fulfilling the percentage of power/load coverage based on the given data if it is not chosen manually. The program then gives a summary of the results of the calculations in a box similar to figure 6.

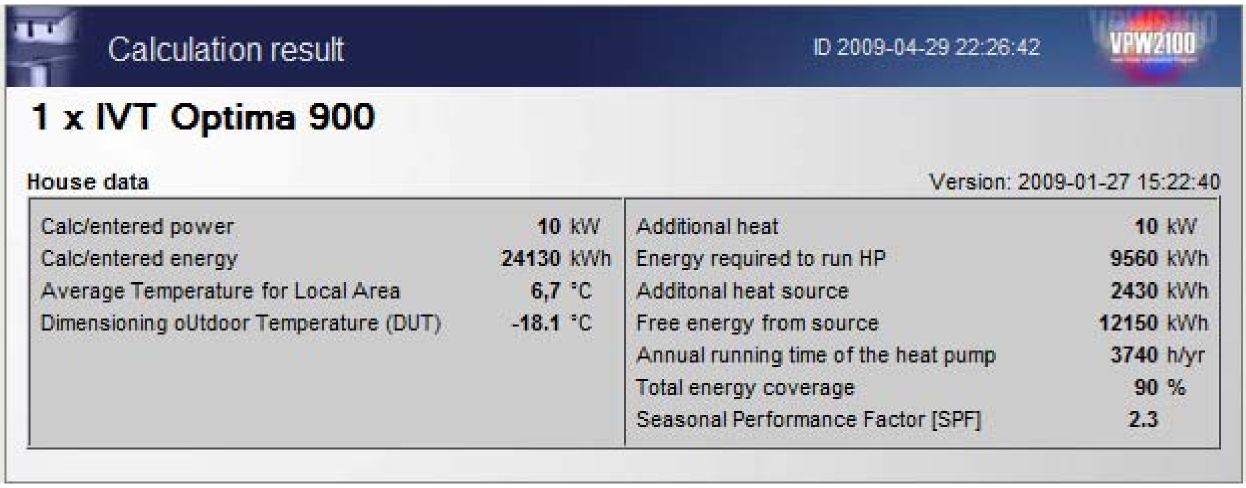


FIGURE 6

If the power demand of the building has not been entered directly, the program calculates it based on total energy consumption and share of household electricity, consumption of other energy sources and the efficiency of the corresponding heating devices, self heating of the house, number of houses, room temperature, number of people in each house, and heat required for DHW, affected by the type of DHW use. When the energy required for DHW has a relatively big share of the total energy demand the power demand is less. For higher current indoor temperatures, when other variables are fixed, the power demand of building will have lower value because when the energy consumption of two buildings are the same but one of them has higher indoor temperature it shows that the warmer house has better insulation and its instantaneous energy loss which should be compensated by the heating system, closely related to power demand, is less.

In areas with exceptionally high DOT, the calculated peak power demand may be different from its real value [1].

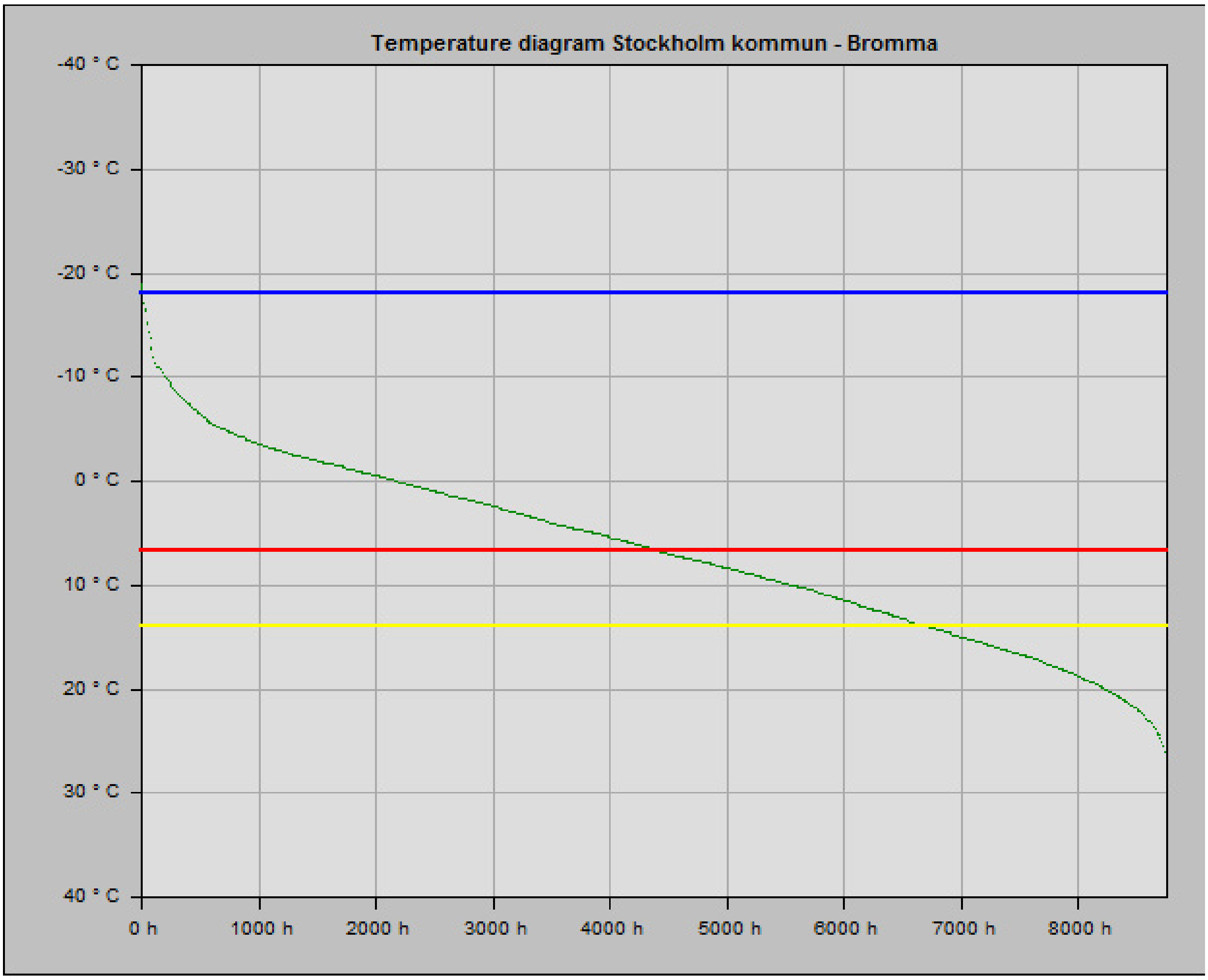
If the energy demand of the building has not been entered directly, the program calculates it based on peak power demand, self heating of the house, number of houses, room temperature, number of people in each house, and heat required for DHW, affected by the type of DHW use. Energy of producing DHW is added to the estimated space heating energy demand of the building. For higher current indoor temperatures of the house, the calculated energy demand is higher because the heating system should work with powers closer to the peak power in more hours to keep the house warmer.

For an existing house, the program calculates the energy requirement of the house using the consumption of electricity, wood, oil, gas, or other sources and the efficiency of the heating devices. The heating value that the program applies for wood is 1200 kWh/m^3, for oil is 9960 kWh/m^3, and for gas is 10.6kWh/m^3 [1].

Average temperature for local area is found forms the hourly temperatures of the location of interest given by METEONORM. The software program has its own procedure, different from Swedish standard SS 02 43 10, to find DOT [1].

According to the specifications of the heat pump the program calculates the energy required to run the heat pump and the amount of energy taken from the source; sum of these two values is the portion of annual energy demand covered by the heat pump. By subtracting the annual heat provided by the heat pump from the total energy requirement, the energy from supplementary heating system is found. Annual energy demand coverage is calculated by dividing the annual heat provided by the heat pump by the total annual energy requirement. Seasonal performance factor is also calculated by dividing the sum of the energy required to run the heat pump and the free energy from the source by the energy required to run the heat pump.

The program creates a diagram, similar to figure 7, showing DOT, average outdoor temperature, the outdoor temperature above which the space heating is not needed, and .



Here is the temperature diagram used by the program to dimension the heat pump and calculate the savings.
The program calculates how the heat pump is operating for every hour in one year (8760).

Stockholm kommun - Bromma

- DUT: -18.1 °C
- Average temperature: 6.7 °C
- Heating need stops at: 14 °C
- Temperature line

FIGURE 7

The amount of free energy from source, the energy from supplementary heating device, and the energy required to run the heat pump are also shown graphically, similar to figure 8, on monthly basis.

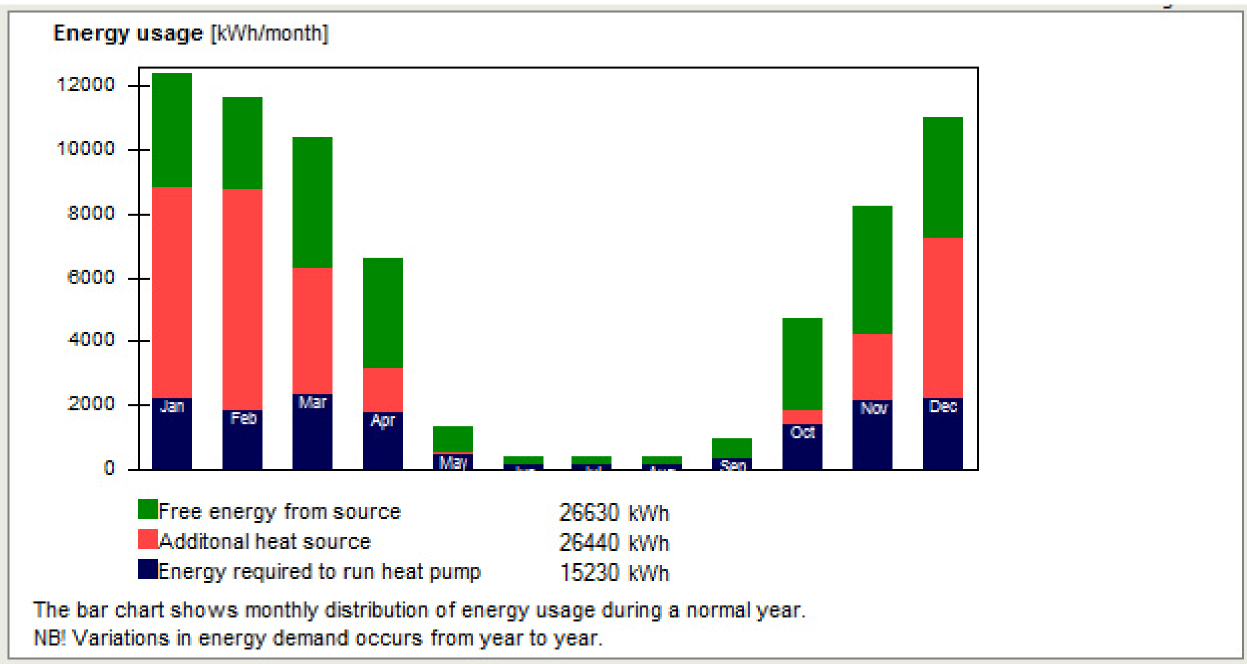
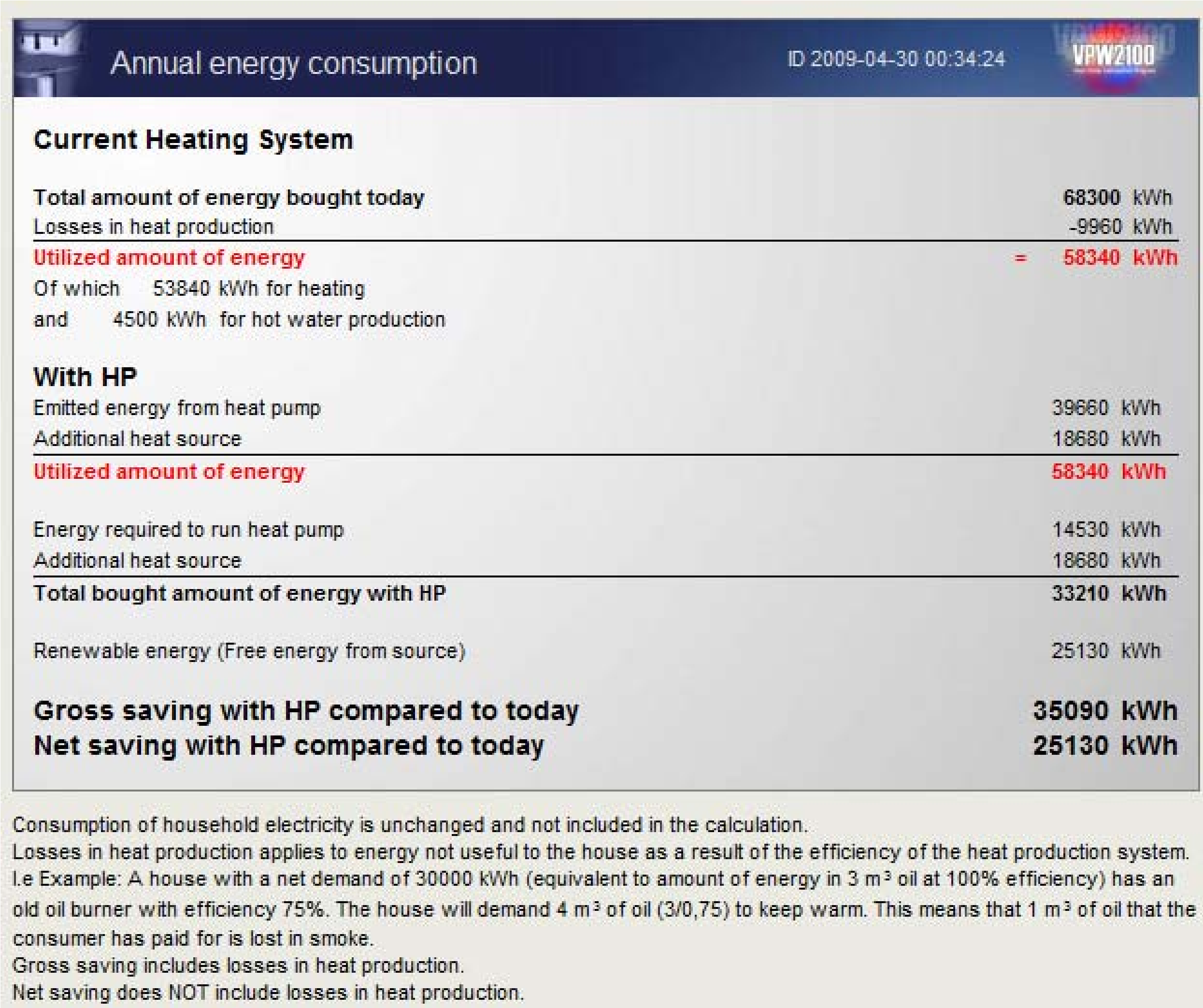


FIGURE 8

For an existing house, calculations made by the program are summarized in another box, similar to figure 9, to calculate net and gross saving.



Annual energy consumption ID 2009-04-30 00:34:24 VPW2100

Current Heating System

Total amount of energy bought today	**68300 kWh**
Losses in heat production	-9960 kWh
Utilized amount of energy	= 58340 kWh
Of which 53840 kWh for heating	
and 4500 kWh for hot water production	

With HP

Emitted energy from heat pump	39660 kWh
Additional heat source	18680 kWh
Utilized amount of energy	58340 kWh
Energy required to run heat pump	14530 kWh
Additional heat source	18680 kWh
Total bought amount of energy with HP	**33210 kWh**
Renewable energy (Free energy from source)	25130 kWh
Gross saving with HP compared to today	**35090 kWh**
Net saving with HP compared to today	**25130 kWh**

Consumption of household electricity is unchanged and not included in the calculation.
Losses in heat production applies to energy not useful to the house as a result of the efficiency of the heat production system.
I.e Example: A house with a net demand of 30000 kWh (equivalent to amount of energy in 3 m³ oil at 100% efficiency) has an old oil burner with efficiency 75%. The house will demand 4 m³ of oil (3/0,75) to keep warm. This means that 1 m³ of oil that the consumer has paid for is lost in smoke.
Gross saving includes losses in heat production.
Net saving does NOT include losses in heat production.

FIGURE 9

Total amount of energy bought today is calculated based on the current consumption of electricity, gas, wood, oil, or other sources. The losses, found based on the efficiency of heating devices, are subtracted from the total amount of energy bought today to find the energy demand of the building. Since the heat pump and the supplementary heating device should be able to provide the annual energy demand, the sum of the emitted energy from heat pump and additional sources is equal to the amount of energy demand. Sum of electricity consumption of heat pump and energy consumption of supplementary heating device gives the total amount of bought energy. If the old heating system is used as the supplementary heating device of the newly installed heat pump the program uses the given efficiency of the old system to calculate the amount of fuel consumption after installation of the heat pump; otherwise, a default value will be used for the supplementary heating device calculations. Then, the gross saving is calculated by subtracting the total bought amount of energy with heat pump from the same value we currently have before installing a heat pump. Net saving is the total annual energy demand minus the total energy bought after installation of heat pump.

4.3.7 Economic calculations

The program has another feature that investigates economic aspects of using heat pump. However, it is beyond the scope of this study.

5 DISCUSSION

5.1 COMPARISON

All the three assessed software packages present similar characteristics in terms of given results and input data. In particular, they provide some values regarding the performances of heat pumps in term of energy consumption, energy saving (compared with the actual heating system), running costs and technical data for the installation.

Table 2 summarizes a comparison between the assessed programs.

	Vitocalc 2008	**Prestige 2.0**	**VPW 2100**
Language	Swedish, English, and German	Swedish	English and Swedish
Space heating annual energy demand	There are three ways to estimate it: based on the area and building use, based on the current electricity or fuel consumption, or based on the heating power demand. Moreover, there are more possible combinations of current fuel consumption to choose.	The only way is to estimate based on the current fuel consumption.	It can be either estimate based on the current electricity and fuel consumption or peak power demand.
DHW annual energy demand	The value should be given to the software tool.	The value should be given to the software tool or the program can estimate a value according to the number of people.	The value can be given to the program manually or the default values for three types of baths can be used.
Weather data	Either both the annual mean temperature and design outdoor	A city in Sweden can be chosen from a list.	A city of 26 countries can be chosen. For Sweden the data are

	temperature should be entered, or, in Swedish version, a city should be chosen from a list or on a map. In the latter case, altitude can also be chosen to adjust the weather data.		more accurate by choosing the location of the house in the city. Weather data used in the program are from METEONORM.
Building characteristics	Estimated based on the year of construction and, only in Swedish version, on the thermal mass of the building (light, medium and heavy.)	Estimated based on year of construction and form and type of the house.	Only self heating of the house shows that how well-insulated the building is.
Building use	In one of the three options to calculate the heating demand building use should be chosen between 9 different uses. It makes the software tool easy to use but reduces the flexibility.	There are 16 combinations, based on which the program assumes the heating demand in kWh/m^2.	Not used in calculations.
Temperatures of the supply and return lines to and from radiators	There is an alternative way to estimate the temperatures if they are not known.	They should be given to the program.	Only temperature of the supply line should be indicated.
Choosing heat pump	Heat pump can be chosen from the products of Viessmann company.	Typical heat pumps with different capacities can be chosen.	It can be done even manually or by the software program.
Heat source	Wider variety of heat sources is available. Although not mandatory, technical details about the heat source can be entered manually to increase the precision of the calculations.	Fixed values are mostly used.	Options are not very detailed.
Calculation results Calculation results have been discussed in detail in chapter 4	The calculated parameters are more or less the same as the other programs although there is a graphical presentation of the results, not used in Prestige.	The calculated parameters are more or less the same as the other programs.	The calculated parameters are more or less the same as the other programs and they are shown in more details graphically.

TABLE 2 COMPARISON TABLE

5.2 SUGGESTIONS

In this chapter some observations and suggestions that have been arisen from the assessment of the software packages are presented.

First of all, it is not possible to compare heat pumps of different manufactures in a very suitable way. In fact, the built-in characteristics of the heat pumps are taken from the catalogue of the manufacture that developed the assessment program. Therefore, a customer that would compare heat pumps of different manufactures should use the different software tools in order to get the performances of these units. However, the assumptions, the calculations and the input data are different for each program. Thus, the results are not comparable in the sense that they are not obtained in the same way.

A limitation common among all the assessed programs seems to be related to the prediction of the percentage of the total energy requirement covered by the heat pump. Although it is not possible to access to the source code, it is still possible to deduce this fact by analyzing the input data. Under particular severe weather conditions, the space heating demand results are high. If, at the same time, there is a high requirement of domestic hot water, the heat pump could not be sufficient and the backup heater could be utilized. These situations are quite frequent especially in wintertime, early morning, when outside air is cold and people tend to shower (e.g. in Sweden). Since a time schedule for domestic hot water consumption is not required as input data by any of the assessed programs, it can be deduced that a large tank is assumed, with enough capacity for storing a large amount of heat. In reality, tanks are usually not large enough, resulting in a need of using backup heating. Accordingly, the calculated percentage of the total energy covered by the heat pump is higher than the real one, meaning higher performance than the actual one is expected.

Notwithstanding reversible heat pumps are becoming popular, none of the assessed tools present the possibility of modelling cooling demand. Thus, it is not possible to estimate the actual electricity consumption of the reversible heat pump over the year but only over the heating season. Furthermore, a customer may think that the bought heat pump has not the expected performance if not well informed by the reseller.

Some bugs of minor importance have been detected especially when entering data. In particular, we experienced that these problems appear if data are changed without refreshing the input data windows .However, it seems that these problem can be easily fixed.

Finally, sometimes the user friendliness could be improved. It has to be noted that instructions and guidelines are available in Swedish language for PRESTIGE 2.0 and VITOCALC

2007 v2 while IVT provides a quite clear and detailed instruction file available on the website of the manufacture.

6 ACKNOWLEDGMENTS

We would like to express our gratitude to Mr. Jan-Erik Nowacki and Mr. Martin Forsen for the time they have spent to help us in this project.

7 REFERENCES

[1] European Heat Pump Statistics, EHPA, Outlook 2008.

[2] IEA Heat Pump Programme's Information Centre (HPC), available at

http://www.heatpumpcentre.org , last modified 30.6.2008, as accessed 9.4.2009

[3] Swiss Federal Office of Energy (SFOE), Seasonal performance calculation for residential heat pumps with combined space heating and hot water production (FHBB Method)– Final report, C. Wemhöner, Prof. Dr. Th. Afjei, October 2003

[4] IEA HPP Annex 28 – standardized testing and seasonal performance calculation for multifunctional heat pump systems, Carsten Wemhoener *, Thomas Afjei, Ralf Dott, Applied Thermal Engineering 28 (2008) 2062–2069

[5] Eric Granryd (2005), ”Refrigerating Engineering” , Department of Energy Technology, Division of Applied Thermodynamics and Refrigeration, Royal Institute of Technology, Stockholm, Sweden.

[6] VDI Richtlinie 2067, Kostenberechnung fur Heizanlagen, Blatt 6, Warmepumpen, Kurzverfahren, VDI-Verlag, Dusseldorf, 1989

[7] Th. Afjei, G. Zwifel, M. Wetter, et al Kostengunstige Niedertemperaturheizung mit Warmepumpe, Technisches Handbuch, darin: Handrechenmethode zur bestimung der Jahresarbeitszahl von Warmepumpenanlagen fur Heiz und Wrmwasserbetrieb SFOE research program heat pumping technologies, cogeneration, refrigeration, 1997 (available at www.waermepumpe.ch/fe/projekte/nth)

[8] C. Feldmann: Methods for calculation of system energy requirement and system efficiency, Working Paper N 259 of CEN/TC 228/WG 4, 2002

[9] ANSI/ASHRAE Standard 116- 1995: Methods of testing for rating seasonal efficiency of unitary air-conditioners and heat pumps, American Society of Heating, Refrigeration, and Air Conditioning Engineers, inc., Atlanta 1995

[10] DIN 4702, Teil 8, Heizkessel, Ermittlung des Normnutzungsgrades und der Normemissionsfaktoren , Beuth Verlag, Marz 1990

[11] Manual VPW2100, <http://www.vpw2100.com/af/vpw2100_manual.asp?lngLangID=5>, as accessed 4/30/2009

Combined solar collectors and heat pump systems for single family dwellings situated in Sweden

Sebastian Lembke 850830
André Szeles 871016

MJ2409 Applied Energy Technology PROJECT course

Department of Energy Technology
Royal Institute of Technology
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Abstract

The objective of this report is to explain and evaluate different kind of heating systems for a single-family house situated in Sweden.
The heating systems that are treated involve solar collectors in combination with a heat pump.
First the techniques of both solar collectors and heat pump systems are described in detail, this is to provide an understanding of which, in order for the reader to understand how the systems can be combined. Combined systems are also described and examples of a few typologies are presented.
A model of one system has been created, in which the performance of a combined system is calculated and analysed. The model provided a number of outputs that explains when and how much the heat pump has to perform in order to complement the solar collectors.
In the analysis of the model-outputs it was found that it is crucial that the system contains a heat storage tank in order for the system to be beneficial. Because the heat provided by the solar collectors has to be available even when the sun is not shining, on cloudy days or during the night a storage tank is needed. If there is no storage tank, the heat pump always has to be running even if the heat provided by the solar collectors is sufficient.
Another interesting subject found during the analysis is that even if the heat energy is sufficient from the solar collectors, the temperatures out of the collector vary a lot. These temperatures are divided into three levels: 1) Over 55 °C and sufficient for both domestic hot water and for space heating, 2) between 30 °C and 55 °C and sufficient only for the space heating and 3) between 5 °C and 30 °C and not sufficient for heating. Where the solar collectors are not sufficient enough the heat pump starts running. In temperature level 3 above, the heat could be used to either rise the evaporation temperature and/or to recharge the borehole.
In conclusion it was found that the best alternative for a combined system, for a house situated in Sweden, is: solar collectors in combination with a heat pump and a ground source heat exchanger as the heat source. Also a storage tank should be installed and the solar collectors should be connected to the evaporator side and the borehole for the heat pump.

Innehållsförteckning

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Nomenclature

α_1	= Asimuthal angle
α_2	= absorptance of the absorber plate
β	= Angle between the tilted surface and the horizontal plane
δ	= declination angle
θ	= Solar zenith angle
ω	= Hour angle
γ	= Variation from south direction
ϕ	= Latitude
τ	= transmission of the cover
A_c	= aperture area of the collector (m^2)
A_{cond}	= Heat transfer area of the condenser
A_{evap}	= Heat transfer area of the evaporator
a	= reflectance coefficient for different ground reflectances
COP_1	= Heating performance
COP_2	= Cooling performance
COP_{1c}	= Carnot heating performance
COP_{2c}	= Carnot cooling performance
$c_{p,cooling}$	= specific heat of the cooling brine
Cp_{htf}	= specific heat of the heat transfer fluid
$\dot{E}_k$	= Compressor work
h_{2k}	= Enthalpy before the compressor, depending on evaporation temp
h_{1k}	= Enthalpy after the compressor, depending on condensing temp
h_{cold}	= enthalpy of the cold water entering at the bottom of the tank
h_{hot}	= enthalpy of the hot water leaving at the top of the tank
Δh_{is}	= is the isentropic enthalpy change over the compressor
Δh_{ref}	= Enthalpy difference of the refrigerant
h_{sci}	= enthalpy of the collector fluid at the entrance of the heat exchanger
h_{sco}	= enthalpy of the collector fluid at the exit of the heat exchanger
I	= irradiance incident on collector aperture (W/m^2)
I_d	= Diffuse solar radiation on a horizontal surface
I_{Ds}	= Direct beam component on the surface

I_{ds} = Diffuse component on the surface

I_g = Instantaneous horizontal global solar radiation

I_h = Intensity of direct solar radiation on the horizontal plane

I_{rs} = reflected component on the surface

$\dot{m}_{cold}$ = mass flow of the cold water entering at the bottom of the tank

$\dot{m}_{cooling}$ = mass flow of the cooling brine

$\dot{m}_{hot}$ = mass flow of the hot water leaving at the top of the tank

$\dot{m}_{htf}$ = mass flow of the heat transfer fluid

$\dot{m}_{ref}$ = refrigerant flow rate

$\dot{m}_{sc}$ = mass flow of the solar collector loop

η_k = isentropic efficiency of the compressor

$\dot{Q}_{cond}$ = Heating capacity

$\dot{Q}_{evap}$ = Cooling capacity

$\dot{Q}_{loss}$ = heat losses from the tank

$\dot{Q}_{\mathrm{t}}$ = energy flow for the tank

T_1 = Condensation temperature of the refrigerant

T_2 = Evaporation temperature of the refrigerant

T_a = temperature of the ambient

$T_{cooling1,2}$ =Temperature of the cooling brine in and out of the condenser

$T_{i,heat-source}$ = Inlet temperature of heat source

T_{in} = the temperature in to the solar collector

$T_{o,heat-source}$ = Outlet temperature of heat source

T_{out} = the temperature out of the solar collector

T_P = temperature of the absorber plate (K)

T_{ref} = Temperature of the refrigerant

T_s = temperature of the storage

U_{cond} = Overall heat transfer coefficient of the condenser

U_{evap} = Overall heat transfer coefficient for the evaporator

U_L = Overall heat loss coefficient (W / (m^2, K))

U_{L1} = component of temperature independent heat loss

U_{L2} = component of temperature dependent heat loss

1. Introduction

This project is a part of an advanced master course at the Institution of Energy Technology at KTH, Stockholm, Sweden.
This report is put together by two students in the master field of Sustainable energy utilization and is treating the subject of a solar collector in combination with a heat pump, for single family houses in Sweden.

The report is describing in detail:

- Function of the solar collector systems and how to calculate the energy output. Also how to design the systems and what parts that is crucial for its functionality.
- Heat pump systems and its function. Also the calculation process for heating and cooling capacity is described.
- Different heat sources for the heat pump are described, also their limitations and advantages against each other.
- Some different combination methods for the solar collector and the heat pump, and what the disadvantages and advantages are for these.

A model of a combined system has also been developed in order to simulate and evaluate the performance. This has further been analyzed in order to present the advantages of such a system and what needs to be done in the future.

The report also describes a number of on market solutions that can be delivered by different companies over the world.

1.1 Background

As energy prices constantly are increasing and extreme weather conditions are relatively common, a higher demand for cheaper solutions to high indoor thermal comfort is seen.
In Sweden, the temperature in the winter time is very low, causing a heating demand during this period. The most common heat pump solution for heating today is a heat pump with ground rock/ground water as the heat source. The compressor in the heat pump is electrically driven, which means that, when the heating demand is at its maximum i.e. during the winter, this will cost a lot of money. Especially as the prices for electrical power is higher during this period. So therefore some alternative cheaper solutions are investigated and evaluated in order to meet the demands.

Some solutions are:

- Passive buildings
- Heat recovery systems
- Better heat pumps and components

This report will treat the aspect of a solar collector in combination with a heat pump.

The main technique is to utilize as much energy from the solar collectors as possible and be able to use the low temperature output.
In these systems there are some optimization difficulties that have to be dealt with, and this report tries to describe these and to provide a solution for a good performing system.

1.2 Objectives

"The objective of the project is to collect, create and evaluate current and future solutions for combined heat pump with assisting solar thermal systems to meet heat requirements of a one family house situated in Sweden.
The most common system set-up found on the market as well as future system will be described in terms of arrangement of components.
The project will from the review of present systems generate a number of conceptual designs of future systems, a classification of systems"

1.3 Limitations

This report will only treat aspects concerning the heat pump technology, solar thermal technology and solutions for combining these. It will not treat the energy load, such as how the performance of these systems is affected by changes in load.

The developed model is simplified due to the complexity of these kinds of systems and should be treated as guidelines.

1.4 Method

To reach the set objectives a series of actions has been made in order to get there.

- Study of literature
  - Collecting information about heat pump and solar collectors at both component and system level.
  - Collecting information about combined heat pump/solar collector system
- Presentation of the study of literature
  - Compilation of the study of literature and presenting it in this report.
- Modeling
  - Development of a simplified model.
- Simulations
  - Make simulations of the model in order to get a foundation for discussion and analysis
- Analysis/Discussion
  - From the study of literature and the simulations make an analysis and discuss the performance of the systems.
- Conclusions
  - Make conclusions based on the analysis/discussions

2. Solar collector systems

There are a lot of ways to collect the energy from the sun and turn it into heat. Basically an absorber is used to transform the suns radiation to heat and transfer this heat from the absorber to a heat carrier.

This section of the report will deal with solar water heating systems that are most suitable for small family dwellings.

The most important components/factors in a solar water heating system:

- Solar irradiance
- Solar collector
- Heat storage
- Heat transfer fluid
- Piping
- Pump
- Control system

2.1 Solar irradiance

Extraterrestrial solar radiation is the amount of radiant solar energy, outside the earth's atmosphere, per unit time and unit area perpendicular to the sun's rays. This value changes during the year (+-3%) (Perez. R et al) due to the changing distance to the sun. At a mean earth sun distance the extraterrestrial solar radiation is 1367 W/m^2, also known as the solar constant. The amount of energy that reaches the surface of the earth is depending on the influence of the atmosphere, clouds position and time. The radiation that reaches the surface of the earth can be divided in two parts; direct radiation that is partially absorbed by the atmosphere but remains unscattered and scattered radiation. During summer the sun has a high orbit over the sky and in the winter it has a low. The suns position in the sky can be determined by the location (latitude), the time of the year and day (see figure 1).

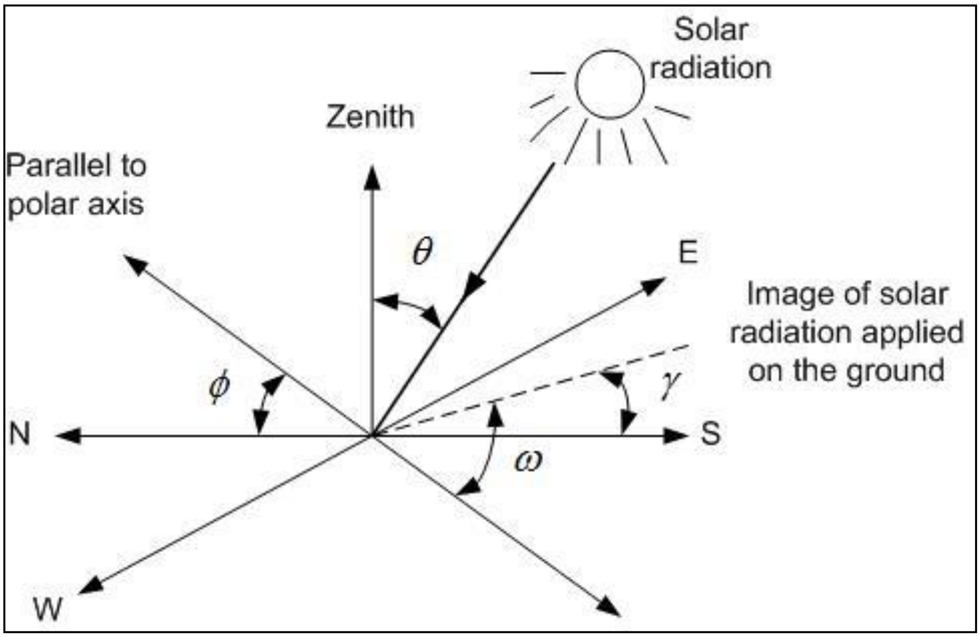

Figure 1: *Solar angles (Williams 1983)*

where θ = Solar zenith angle
ω = Hour angle
γ = Variation from south direction
ϕ = Latitude

The instantaneous global radiation on a tilted surface can be calculated with

$$I_{gs} = I_{Ds} + I_{ds} + I_{rs} \tag{1.1}$$

where I_{Ds} = Direct beam component on the surface
I_{ds} = Diffuse component on the surface
I_{rs} = reflected component on the surface

Solar radiation on a tilt surface is shown in figure 2.

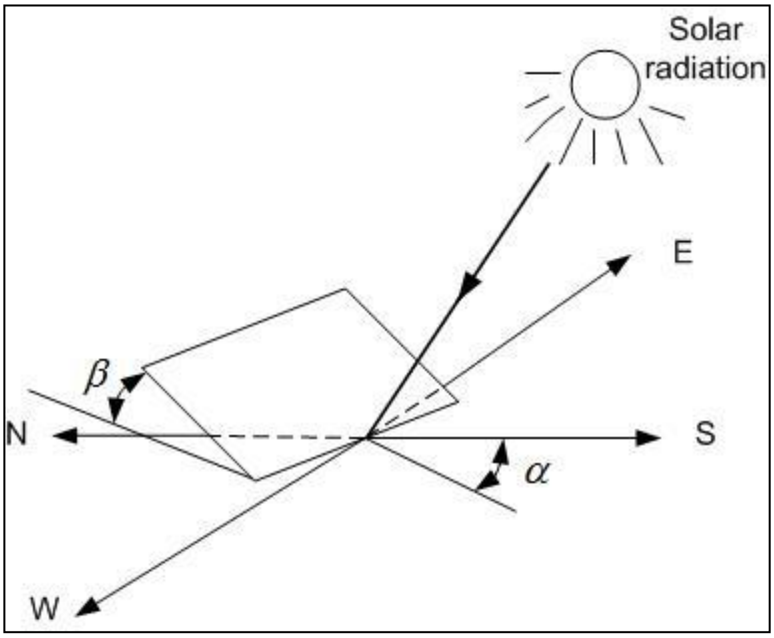

Figure 2: *Solar radiation on a tilted surface (Williams, 1983)*

Where α_1 = Asimuthal angle

β = Angle between the tilted surface and the horizontal plane

If I is the intensity of the direct solar radiation component, the intensity of a tilted surface is

$$I_{Ds} = I \cdot \cos(\psi) \tag{1.2}$$

where ψ can be calculated from

$$\begin{aligned}\cos\psi &= \sin\beta\left[\cos\delta(\sin\phi\cos\alpha\cos\omega+\sin\alpha\sin\omega)-\sin\delta\cos\phi\cos\alpha\right] \\ &+\cos\beta(\cos\delta\cos\phi\cos\omega+\sin\delta\sin\phi)\end{aligned} \tag{1.3}$$

where δ = declination angle.

Measurements of solar radiation are usually given on a horizontal plane. The relation between the horizontal solar radiation and the radiation on the tilted surface are shown in figure 3.

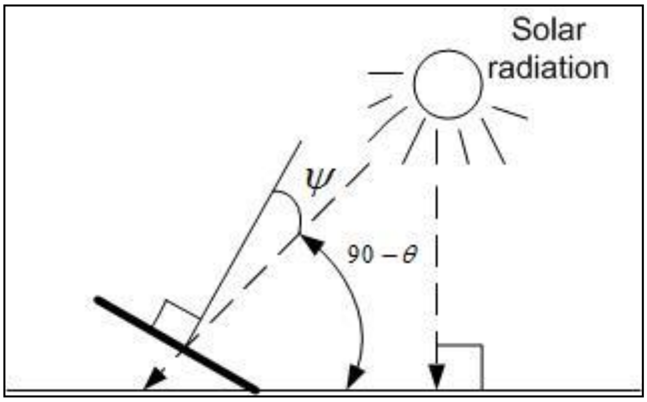

Figure 3: *Relation between radiation on horizontal and tilted surface*

The direct component on the tilted surface can be calculated with

$$\mathrm{I}_{Ds} = I_h \frac{\cos\psi}{\sin(90-\theta)} \qquad (1.4)$$

where I_h = Intensity of direct solar radiation on the horizontal plane

The diffuse component can be calculated with

$$I_{ds} = \left[\frac{(1+\cos\beta}{2}\right]\cdot I_d \qquad (1.5)$$

where I_d = Diffuse solar radiation on a horizontal surface

The reflective component can be calculated with

$$I_{rs} = a\left[\frac{(1-\cos\beta)}{2}\right]\cdot I_g \qquad (1.6)$$

where a = reflectance coefficient for different ground reflectance
I_g = Instantaneous horizontal global solar radiation

(Equations from Williams, 1983)

By optimizing the location of the solar collector the suns direct radiation can be utilized more effectively. According to (Kjellsson, 2004) is the optimal tilt for a solar collector in southern Sweden:

- 30-45° tilt during summer with a maximum of +-15° variance from south direction.
- 60-65° tilta duning Winter.
- 75-80° tilta for december

This implies that depending on the application of the collector system, and when the energy is needed, different tilts can be used.

2.2 Solar collectors

2.2.1 Flat plate solar collector

A flat plate solar collector absorb the energy from the sun light both through direct and diffuse parts of the sun radiation and transfer this heat to a heat transfer fluid that runs through the collector. The absorber plates should have a high thermal conductivity and have good corrosion resistance. Usually some kind of metal or thermoplastics is used. To improve the absorption the plate has a coating with high absorbance and low remittance, e.g. black metallic oxide painter on a metal surface. This kind of coating is called a selective coating. (G.L Morrison, 2001)

A cover glass is used to prevent heat losses from the collector. This cover is usually toughened low iron glass, due to its high transmittance, its relatively good insulation properties and its high resistance to thermal cycles. Anti-reflecting coatings are used to increase the transmittance of the glass. To improve insulation a second layer of glass can be added to the casing and is mostly used in high temperature applications.

Material properties of the back insulation of the casing are important to consider in order to avoid out-gazing when the casing are exposed to elevated temperatures. It is important that the insulation is kept dry or it will lose some of its insulation properties, this can be avoided by sealing and desiccants.

2.2.2 Evacuated solar collector

Consists of an absorption surface inside a vacuum which reduces the heat transfer because convection in eliminated. Some different designs for evacuated collectors are: metal fin attached to a U-tube that removes the heat from the absorber, it can also be attached to a once through pipe, although this design requires bellows to compensate for the expansion of the metal tubes and the cold glass. When using a heat pipe system bellows are not required. The evacuated collectors are more efficient than flat plate collectors but because material stresses due to pressure, the manufacturing of evacuated collectors are more expensive.

2.2.3 Unglazed solar collector

This kind of collectors have no cover glass over the absorber, because of this there is no transmission losses due to a cover, but the significant increase of heat loss to the surroundings makes this kind of solar collector very inefficient at higher temperatures. It can efficiently be used in low temperature applications such as swimming pool heating.

2.2.4 Output and efficiency of flat plate solar collector

For steady state conditions the useful heat output for a flat plate solar collector is given by

$$Q_{out} = A_c[\tau\alpha I - U_L(T_P - T_a)] \tag{1.7}$$

where τ = transmission of the cover

α_2 = absorptance of the absorber plate

A_c = aperture area of the collector (m^2)

I = irradiance incident on collector aperture (W/m^2)

U_L = Overall heat loss coefficient (W / (m^2, K))

T_P = temperature of the absorber plate (K)

T_a = temperature of the ambient (K)

During operation of the collector system it is difficult to determine the temperature of the absorber. Due to this the temperature of the absorber plate is usually replaced with the average fluid temperature of the collector;

$$T_m = \frac{T_{out} + T_{in}}{2}. \tag{1.8}$$

The temperature dependent form of U_L is

$$U_L = U_{L1} + U_{L2}(T_m - T_a) \tag{1.9}$$

where U_{L1} = component of temperature independent heat loss

U_{L2} = component of temperature dependent heat loss

Equation (1.1) then becomes

$$Q_{out} = A_c[\tau\alpha I - (U_{L1} + U_{L2}(T_m - T_a))(T_m - T_a)] \tag{1.10}$$

The thermal efficiency of the collector can be expressed as

$$\eta = \frac{Q_{out}}{A_c I} \tag{1.11}$$

or

$$\eta = \tau\alpha - \frac{[U_{L1}(T_m - T_a) + U_{L2}(T_m - T_a)^2]}{I} \tag{1.12}$$

Equations from (Morrison, 2001)

U_L depends on heat losses from the top through the cover and from heat losses through the insulation at the bottom and the sides of the collector. The main heat loss from a flat plate solar collector is by convection and radiation through the cover.

The heat removal factor, F_R, is defined as the actual useful energy collected divided by the useful energy collected if the entire collector absorber surface were at the temperature of the fluid entering the collector. Introducing this gives:

$$Q_{out} = A_C F_R[\tau\alpha I - U_L(T_i - T_a)] \tag{1.13}$$

Equation (1.12) shows that unglazed solar collectors are very efficient (τ =1) when the mean temperature of the fluid in the collectors is close to the surrounding temperature. When this temperature difference grows bigger the efficiency rapidly decreases. This implies, as discussed in section 1.1.3, that unglazed solar collectors are most efficient used in low temperature applications.

2.3 Heat storage/heat exchange

To be able to use the heat given by the collectors even when the weather conditions are unfavorable a heat storage tank is needed. A heat exchanger is needed to transfer the heat from a closed collector system to the storage. This can be done by an internal heat exchanger within the storage tank (e.g. coil) which has the inlet at a high level and the outlet at a lower level. External heat exchanger can also be used and because of natural circulation no additional pump is needed. Another option can be a mantle heat exchanger. To minimize heat losses from the tank, it is desirable that thermal bridges such as piping are situated at the bottom of the tank where the temperature is low.

When using the same tank for solar heated water and auxiliary heating problems could arise with interactions between the solar heated part and the auxiliary part. Mixing and conduction through the fluid and the tank walls could result in preheating of the water in the bottom of the tank which affects the temperature of the inflow and outflow of the solar collectors. Two tanks can be used to eliminate this problem or auxiliary boosting can take place outside the tank.

It is important to maximize stratification within the tank. One reason for this is to get lower temperature at the bottom of the tank where the collector water outlet is, this implies lower mean temperature in the collector and according to equation (1.12) this will lead to increased efficiency. Another reason to strive for good stratification is the exergy loss due to mixing, if high temperatures can be utilized in the collectors but thru heat exchanging and mixing with the colder water a lower water temperature can be extracted from the tank. This can be reduced by properly design of the in- and outflows of the tank, optimization of the flow rate in the loops as well as choosing the most adaptable collector/tank heat exchanger.

The load can have an impact on efficiency due to storage of heat. If the peak load is in the evening then there is not as much storage during night, hence, reduction of heat loss.

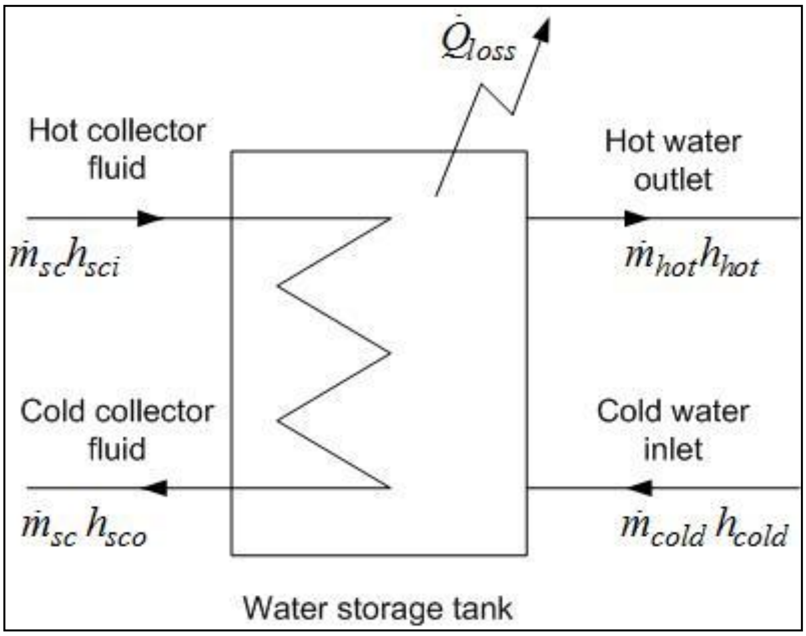

Figure 4: *Energy flow in a water storage tank*

The energy flow for the tank in figure 4 without auxiliary heating can be expressed as

$$\dot{Q}_{\mathrm{t}} = \dot{m}_{sc}(h_{sci} - h_{sco}) + \dot{m}_{cold}h_{cold} - \dot{m}_{hot}h_{hot} + \dot{Q}_{loss} \quad (1.14)$$

where

$\dot{Q}_{\mathrm{t}}$	= energy flow for the tank
$\dot{m}_{sc}$	= mass flow of the solar collector loop
h_{sci}	= enthalpy of the collector fluid at the entrance of the heat exchanger
h_{sco}	= enthalpy of the collector fluid at the exit of the heat exchanger
$\dot{m}_{cold}$	= mass flow of the cold water entering at the bottom of the tank
h_{cold}	= enthalpy of the cold water entering at the bottom of the tank
$\dot{m}_{hot}$	= mass flow of the hot water leaving at the top of the tank
h_{hot}	= enthalpy of the hot water leaving at the top of the tank
$\dot{Q}_{loss}$	= heat losses from the tank

and

$$\dot{Q}_{loss} = \sum UA(T_s - T_a) \tag{1.15}$$

where

T_s = temperature of the storage

T_a = temperature of the ambient

2.4 Heat transfer fluid

When choosing the heat transfer liquid in the solar collector system it's important to consider thermal expansion, viscosity, thermal capacity, freezing- boiling- and flash point. If freezing can occur, an ethylene or propylene glycol-water mixture is often used. Since ethylene is very toxic it is nowadays more common to use propylene due to environmental constrains. To obtain a secure freezing protection recurrent supervision of the properties of the fluid must be carried out and appropriate adjustments be made. If leakage exists in the system and additional water is added it is important to ensure that the fraction of freezing protection is high enough. (Energysavers, 2009)

With higher fraction of anti freezing in the brine the freezing protection gets better, although this is a trade of against higher viscosity, lower thermal conductivity and lower specific heat. (Williams, 1983)

2.5 Pump/piping

Passive or thermosyphon systems uses natural circulation to circulate the water in the system, this implies that the water tank is situated at a higher level than the solar collectors in order for the solar heated water to rise up in the tank. The problem with these systems is the location of the tank and the size of the system. The most common location for solar collectors in small dwellings is the roof and if a passive system is used, the tank must be at a higher level than the collectors. In smaller systems this can be done but in larger systems the problems get bigger due to the weight of the tank. There can also be problems in colder climates if the tank cannot be located inside the house as heat losses will increase because of high temperature differences.

Active systems use a pump to circulate the water through the collectors. The use of a pump makes these systems less efficient than passive ones. There are two kinds of ways to control the speed of the pump, manually or automatic. The automatic ones is usually pressure dependent and most often this kind of pumps are more efficient then the manually regulated. In a test done by Energimyndigheten, 13 different circulations pumps where tested, this test showed that the efficiency of different pumps can differ as much as 18 percent and that the difference between new pumps can be large. (Energimyndigheten 2009a)

In locations where freezing will occur during wintertime a closed loop system is more often used. In a closed loop system the fluid pumped through the collectors transfer the heat to the water storage tank through a heat exchanger. This enables usage of an anti-freezing fluid in the solar collectors. High flow through the collectors results in higher heat transfer to the heat transfer fluid; hence more energy can be extracted from the system. If performance of the total system is considered a low flow system is to prefer. The decrease in output is compensated by lower pumping power in the collector loop, better stratification in the heat storage tank and due to low flow (laminar), the heat transfer is independent of the diameter of the tube and therefore smaller pipes can be used and installation cost is reduced. (Duff, 1996)

It is important for low flow systems that (Duff, 1996):

- The flow in the collector loop should be about 2 to 4 grams/sec-m2.
- The flow into the heat storage should give optimal stratification in the tank.
- Total flow volume through the collector for an average day should be matched to the volume supplied to the load for an average day.
- The collector and load flow rates should be optimally matched.

When using two or more collectors together some different connections can be made. The most common are series or parallel. Series coupled collectors are mostly used in high temperature applications and they are less efficient then parallel coupled due to higher average temperature in the collectors and higher pressure drop. Another drawback is that the series of collector can be taken out of operation due to damage in just one collector. With a parallel system it is important with balanced flows out of each collector.

2.6 Control system

The control system of a solar heating system should consist of sensors, differential thermostats and valves.

For the collector system to be useful there has to be a temperature difference between the outlet of the collectors and the bottom of the tank. The collector loop pump starts when a differential thermostat, connected to sensors at the outlet of the collectors and at the bottom of the tank, reaches a preset value. At this value the pump starts working and a check valve opens (check valves is used to ensure that the no backflow occur when the pump is off). If temperature difference drop while operation the shut down temperature should be lower than the start temperature in order to avoid cycling of the pump.

To determine if the collectors is transferring sufficient heat to the storage tank two other sensors, one at the inlet of the collector/storage heat exchanger and one at the outlet should be mounted. If the temperature difference of these two values becomes too small it is not profitable to run the system anymore and the pump should be shut off.

A tempering valve (mixes hot and cold water) must be used in order to insure that the domestic hot water temperature does not exceed the highest allowed temperature. In some systems a overheat protection is used to avoid stagnation temperatures in the collectors which can decrease the life span of pipes, insulation and brine. Stagnation temperatures can be reached for instance in case of a power outage or if the pump stops due to low temperature difference between the bottom of the tank and the collector. To avoid this heat dumping can be necessary in order to lower the temperature at the bottom of the tank.

In indirect systems an expansion tank is needed to compensate for expansions as the temperature gets higher.

(Williams, 1983)

2.7 System design

To build an optimized solar heating system many factors must be considered.

- To what is the system going to be used?
- E.g. heating and DHW or only DHW
- What is the optimal size for the system considering the investment and the desired degree of coverage?
- E.g. take in different offers
- What kind of auxiliary energy source is going to be used?
- E.g. district heating, wood, electricity, heat pump

When these questions are answered the optimization of the components can begin and depending on the answers different components is used to build the optimal system.

Some main principles when sizing a collector system are according to Energimyndigheten 2009c;

- 5 m^2 solar collector area for DHW for a family dwelling
- 10-15 m^2 solar collector area for heating and DHW

3. Heat pumps

According to the International Energy Agency (IEA), there are two main types of heat pump systems in use today, where one is based on a vapour compression cycle and the other on an absorption cycle. (International Energy Agency 2009)

This report will be focused on the vapour compression cycle type. This system includes four steps: evaporation, compression, condensation and expansion. Evaporation and condensation are two types of heat exchangers where the working fluids of the cycle exchange thermal energy.
The compression is done by a compressor where a (often electrical) motor is the source for the incoming work. (International Energy Agency 2009)

3.1 Heat pump systems

When choosing a suitable heat pump system, there are many factors to be considered, as the incoming heat for the cycle can be obtained in a few different ways. Some of the most commonly used heat sources are; ambient air, ground water, lake/river/sea water, rock, ground and waste water. (International Energy Agency 2009)

It is important that the heat source is stable and at relatively high temperature during the heating season, therefore this is a key factor when deciding for a system. Other important factors are; availability for utilization, thermophysical properties of the heat source and investment costs for the utilization. (International Energy Agency 2009)

Considering above mentioned factors it is also important that the heat source will meet the requirements of the building, such as heat load etc. therefore it would be a good idea to calculate and evaluate the performance of the building before deciding for a heat pump system. (Caneta Research inc. 1995)

3.2 Heat exchangers and heat sources

After deciding which heat pump system to be used the next step is to choose for, and size the heat exchanger that is collecting the heat.

3.2.1 Ground heat exchanger systems

Depending on a few different factors, there are two types of ground heat exchangers, horizontal bore field and vertical field, that can be implemented, both in different variants. Factors to take under consideration are:

- Land area that can be exploited, a horizontal field require a larger area.
- Height of the building (especially when deciding for a high rise multi flat building), the pressure head cannot get exceeded. This is crucial when considering a vertical bore field.

- Location of the horizontal heat exchanger field, different surface materials such as asphalt or grass etc. can give different working temperature for the heat pump because of the annual temperature variations and the absorptivity of the surface.
- Installation cost is the key factor for deciding heat exchanger type, but only if both of the alternatives performances are comparable.

(Caneta Research inc. 1995)

Ground source vertical heat exchanger

There are a few alternatives of vertical heat exchangers available. The basic technique is to let a u-tube circulate a fluid in a borehole in order to heat exchange with ground heat. Low temperature water (with 30% anti freeze liquid, ethanol) is heated in the borehole to a required temperature, to heat exchange again in the evaporator of the heat pump.
(Caneta Research inc. 1995)
As mentioned above, there are different types of vertical heat exchangers, some examples are:

- One borehole per flow circuit and one u-tube per borehole, this is a common technique for regular family residents in Sweden. (Svenska Värmepumpsföreningen 2010a)
- One borehole per circuit and a pair of u-tubes per borehole.
- Multi boreholes per circuit and one u-tube per borehole.

The economic perspective of these techniques is that with multiple u-tubes in each borehole the total price of the pipes can get high, but this cost can on the other hand abolish the cost for drilling. Local prices and conditions decide for which technique to be used.
(Caneta Research inc. 1995)
The size of the heat exchanger is decided upon the loads of the building, which is a function of; building type, size, location, internal heating, infiltration, solar gain, envelope conduction and ventilation losses. Other design parameters are mean ground temperature, type of rock (conduction), efficiency of the actual heat pump and size of the pipe.
Advantages with this system:

- *In Sweden, this is the most common type and therefore it is a large market.*
- *A secure system with relatively long life length*.

Disadvantages:

- *High costs for drilling and installation.*

(Svenska Värmepumpsföreningen 2010a)

Ground source horizontal heat exchanger

The technique here is the same as for vertical heat exchangers, except for that the u-tube circuit is placed in a horizontal plane, approximately 90 – 150 cm under the surface, depending on the location and mean temperature. So in this system the sun has a sufficient influence.
There are different alternatives for this type as well, where the installed system could use:

- A two-pipe alternative with either side-by-side or over-under configuration.
- A four-pipe arrangement, also side-by-side or over-under, or even a combination.
- A six-pipe configuration.

The sizing principle for the horizontal heat exchanger is in general the same as for the vertical type. The difference is that it is more sensible to annual temperature variations and also what kind top layer there is.
This kind of system requires a much larger surface area and can be difficult to implement, especially in locations where the buildings are tight situated. Nowadays there are techniques though, where the pipes are in spirals and therefore the surface area can be reduced.
(Caneta Research inc. 1995)

Advantages with this system:

- *Same as for vertical systems.*

Disadvantages:

- *Could make a large impact on the garden.*
- *Not suitable in areas where the buildings are situated tight.*
- *Larger temperature fluctuations than for the vertical case*

(Svenska Värmepumpsföreningen 2010a)

3.2.2 Groundwater heat exchanger system

This type uses the groundwater as a heat source for the heat pump. The groundwater temperature is usually around 4-8 °C annually and is therefore stable as a heat source. There are two ways of collecting this heat:
(Svenska Värmepumpföreningen, 2010b)

Closed loop system

A closed loop groundwater system uses the same technique as for the other ground source system, except for that the heat source is from the groundwater instead of the rock.
The system collects groundwater with a professionally designed well, often combined with pumps, and circulates it in a loop where it heat exchange with the internal loop of the building and then get recharged to the ground, either just on the surface or injected in to the ground. The latter one is more expensive, but often this is a regulatory demand. The heat exchange to the internal loop is done with a regular plate and frame heat exchanger, where a metal sheet is used to transfer heat between two fluids.
There are several alternatives for a closed loop system; one is the regular where the source is connected to a heat pump, and another use a boiler before the heat pump. This could be an alternative if the heat collecting area is limited.
There are some design parameters when deciding for a suitable well, which should be done by a professional hydro geologist. After investigating for heat loads, ground water availability etc. the hydro geologist will decide for number of wells, spacing and size, for both the supply and recharge wells. Other factors that the hydro geologist have to take into account are the temperature change, pressure- drop/rise, PH-levels, gas concentration etc. for the engineer to design the system.
(Caneta Research inc. 1995)

Open loop system

In an open loop system, the groundwater is directly supplied to the heat pumps with no heat exchange in between. Because of this, there are several requirements that have to be taken into account. There should be plenty of ground water available, it should be of high quality, it should be a low-rise building (in order to have low pump power) and the static water level should be high. It is also important to identify if there is any corrosion species in the water to be able to choose pipes, heat exchanger etc.

Usually there are separate supply and discharge lines, where the supply is pressurized with the well pump, and the expansion valve keeps the required pressure head constant.

(Caneta Research inc. 1995)

Advantages with ground water heat pump systems:

- *High and stable supply temperature.*
- *Can be used when high power requirements.*
- *Long life length.*

Disadvantages:

- *For an open loop there could be pollutions in the ground water from discharge water.*
- *The supply water have to be of high quality.*
- *Expensive drilling.*
- *Insecure knowledge of water availability before drilling.*

(Svenska Värmepumpföreningen, 2010b)

3.2.3 Sea/lake water heat exchanger systems

The sea/lake water is a good heat source because of that the water temperature are at approximately 4°C at its minimum, and does therefore provide a stable supply for the heat pump.

The technique is to place a collector-tube at the bottom of the sea/lake where the water temperature is constant during most of the year, deeper the better, also because of the lower risk for damages. This type is common for buildings with high heat consumption. The heat pump in the system is the same as for ground heat exchanger systems, so the sizing and design factors are the same.

Advantages with sea/lake water systems:

- *Cheap to place a collector-tube in the water.*
- *Same heat pump as in ground source heat exchanger, and therefore a lot of alternatives on the market.*
- *Long life length.*

Disadvantages:

- *An even sea/lake bottom is preferred when placing the tubes.*
- *Have to keep the collector on the bottom prevent damages etc. which could be a problematic task.*

(Svenska Värmepumpföreningen, 2010b)

3.3 Heat pump function and configuration

A heat pump works in four steps:

1. Evaporation: In this step, the working fluid is kept colder than the supply temperature, causing a heat flow from the heat source to the working fluid, which will evaporate.
2. Compression: Now when the fluid is in vapour phase, it is easier to compress. The compressor is driven by a motor (often electrical) and is increasing the pressure of the working fluid. This will cause a temperature rise.
3. Condensation: After leaving the compressor, the working fluid is entering the condenser, still in high-pressure vapour phase, where it condensates and distribute heat to the building.
4. Expansions: The last step in the cycle is an expansion where the pressure of the working fluid is decreased. This will cause a temperature decrease as well, that will enter the evaporator.

(International Energy Agency 2009)

3.3.1 Evaporators

The evaporator design is different depending on what applications it should be used for. So there are different design parameters for this. There are different types depending on what heat source the use; gas (air cooler), liquid (liquid cooler) or solid. This report will consider indirect systems, where the evaporator itself is a liquid cooler but the heat is transferred from the source via a brine (liquid) to the evaporator.

The refrigerant that flows in the heat pump is evaporated here, with the heat source, e.g. ground heat, as the driving force. The temperature of the refrigerant is constant during the evaporation, but is often superheated before the compressor. Before the evaporation there is, as mentioned above, an expansion valve which lower the pressure of the refrigerant and thereby also the temperature. This is a crucial step for making the evaporation possible.

There are two possible ways to force the refrigerant to evaporate:

- Dry expansion: Here the refrigerant is forced through the evaporator and then on to the compressor. In this case a superheating has to be made after the evaporation to prevent that liquid reaches the compressor.
- Flooded evaporation: Here the refrigerant is collected in a low-pressure receiver, where the vapour and liquid are being separated.

There are two main types of liquid coolers, both shell and tube evaporators. One where the refrigerant flows inside the tubes and another with the flow outside the tubes. Advantages with inside flow are that there is no risk for oil accumulation inside, as the refrigerant will transport it to the exit, another is that the refrigerant charge will be smaller. An advantage with outside flow is that the brine side will be easier to clean. This could be a good alternative when using sewage water as a heat source.

The performance of an evaporator is dependent on both the mass flow of the refrigerant and the temperature difference from the inlet and the outlet.

The cooling capacity of an evaporator can be calculated with:

$$\dot{Q}_{evap} = UA_{evap} \cdot \vartheta_m \tag{1.16}$$

where U_{evap} = Overall heat transfer coefficient for the evaporator

A_{evap} = Heat transfer area of the evaporator

and

$$\vartheta_m = \frac{\vartheta_i - \vartheta_o}{\ln(\frac{\vartheta_i}{\vartheta_o})} \tag{1.17}$$

where

$$\vartheta_i = T_{i,heat-source} - T_{ref} \tag{1.18}$$

$$\vartheta_o = T_{o,heat-source} - T_{ref} \tag{1.19}$$

where $T_{i,heat-source}$ = Inlet temperature of heat source

$T_{o,heat-source}$ = Outlet temperature of heat source

T_{ref} = Temperature of the refrigerant

Desired mass flow rate of the refrigerant can then be calculated as:

$$\dot{m}_{ref} = \frac{\dot{Q}_{evap}}{\Delta h_{ref}} \tag{1.20}$$

where Δh_{ref} = Enthalpy difference of the refrigerant

$\dot{m}_{ref}$ = refrigerant flow rate

(Department of Energy Technology, KTH, 2009)

3.3.2 Compressors

A compressor is a device that compresses any compressible fluid or gas. It could be of several types, such as reciprocating, rotary, scroll, screw or centrifugal. In a heat pump the compressor should extract the vapour from the evaporator and force it into the condenser. (Department of Energy Technology, KTH, 2009)

There are two different compression methods, one is referred to as intermittent and the other to continuous. The intermittent method is a cyclic compression where a specific amount of gas is taken in to the process, compressed, and discharged before the next cycle starts. In a continuous compression, the fluid flow is constant through the compressor, which means that there are no interruptions.

There are several types of compressors that can be placed under these above-mentioned methods.

Intermittent:

- Reciprocating: There are two types of reciprocating compressors, piston compressor and diaphragm compressor. The piston compressor is one of the most used compressors, and it works with a displacing action where a piston compresses the media inside a cylinder, from a mechanical shaft work, together with the inlet valves. The outlet valves in the discharge line prevent a backflow to the compression cylinder as the next cycle starts. The piston compressor is well suited for high-pressure applications and is one of the most efficient at a pressure ratio over 1.5. The diaphragm compressor has the same layout as the piston compressor, but it works with oil rather than with gas and is suitable for small flow applications.
- Rotary: The rotary compressors have three things in common; (1) the gas is compressed by rotating elements, (2) the compression is intermittent and (3) there are no inlet or outlet valves.
  They use either helical or spiral lobes to compress the gas between them and the rotary chamber. In the spiral case, the compression starts as the spiral moving over the inlet and bring a specific amount of gas. As the spiral screws inward through the rotary chamber, gas is being compressed as the volume is decreasing. The gas is then discharged in a nozzle at the end of the compression. This type is used only for lower pressure applications.
  The helical type (screw) compressor is either dry or flooded, where the flooded use a liquid to prevent the rotors from touching and the dry uses gears for the same reason. The technique is similar but it is more applicable to higher-pressure applications. The compression state is at the discharge port where the backflow is the driving force for the compression.
  (Brown, R.N. 2005)
- Scroll compressor: In a scroll compressor the compression takes place between two spiral shaped chambers. One of these spirals is fixed with the other one orbiting in a fixed pattern (orbiting motion) inside it. As gas entering in the outer periphery the inner spiral will do one orbit and the size of the "gas pocket" will decrease and also compress the gas. Continuing the process, this pocket will move closer to the center, where the discharge is. Since there are two identical scrolls, one continuously orbiting, there will be a new pocket directly after the discharge pocket, and therefore the process is almost continuous and sometimes it is.
  A scroll compressor is suitable where there is a need for low sound and low vibration. It has relatively high volumetric efficiency and it is not very sensitive to pressure ratio compared to a reciprocating compressor.

But, because of the high precision of the manufacturing of the scrolls, the development has been a little bit constrained. (Department of Energy Technology, KTH, 2009)

Continuous:

- Dynamic: These compressors use a different technique than the above-mentioned. As they are continuous, the compressing never gets interrupted and are suitable for large volumes with a limited compression ratio. They can be either radial or axial, with the flow in that direction respectively. The radial alternative is often called centrifugal and uses an impeller to force the flow in radial direction, from the area near the shaft to the discharge, called a diffuser.

The axial alternative uses rotor blades as the energy converter. There are several of these blades, and before and after every rotor are a stationary blade. So the flow is forced through these stationary blades with the rotor blades, and the fluid is compressed.

(Brown, R.N. 2005)

In order to calculate the work needed for the compression there are several parameters to take into account. First one needs to specify the pressure rise/temperature rise in order to determine the enthalpy change over the compressor itself. Then the compressor work can be calculated as:

$$\dot{E}_k = \frac{1}{\eta_k} \cdot \dot{m}_{ref} \cdot \Delta h_{is} \tag{1.21}$$

where $\dot{E}_k$ = Compressor work

Δh_{is} = is the isentropic enthalpy change over the compressor

η_k = isentropic efficiency of the compressor

Now it is possible to introduce the cooling performance of the cycle, which means the cooling capacity divided by the compressor work needed:

$$COP_2 = \frac{\dot{Q}_{evap}}{\dot{E}_k} \tag{1.22}$$

Where COP_2 = Cooling performance

$\dot{Q}_{evap}$ = Cooling capacity

(Department of Energy Technology, KTH, 2009)

3.3.3 Condenser

At this side of the heat pump, the heat will be distributed to the buildings heating system.

The refrigerant entering the condenser is superheated vapour from the compressor and will leave in a subcooled state.

The size of the condenser and its working temperature depend on the compressor type and its size. This has to do with the required COP for the heat pump, which is a relation between the compressor work and the amount of media that should be heated by the condenser. So if the compressor is poor there is a need for a large condenser.

Air-cooled condenser: This type uses air as coolant where the heat exchange is due to convection, either forced or natural. Natural convection types are commonly used in refrigerators in regular family houses (domestic).

Water-cooled condenser: This type use water as coolant and the heat exchangers that are mostly used are brazed plate-type, tube in tube-type or tube in shell-type. The shell in tube condensers are used in larger plants, where the refrigerant condenses on the outside of the tubes and the water flows inside these tubes.

Depending on the desired indoor temperature and the heat transfer area of the radiators different water temperature must be obtained. In older systems with small heat transfer areas supply temperatures about 80 C° and return temperatures of 60 C° is common. For newer systems temperatures around 55 in supply and 45 in return is common. When using floor heating these temperature are even lower. (Seminar, Claesson 2009)

Evaporative condensers: In this type, the refrigerant passes through the condenser inside coils, which are wetted on the outside by water. This water will then condense and forced to evaporate by air flowing over the coils. This evaporation removes heat from the evaporation.

In order to calculate the heating capacity of the condensers the following expression can be used:

$$\dot{Q}_{cond} = UA_{cond} \cdot \vartheta_m \tag{1.23}$$

where $\dot{Q}_{cond}$ = Heating capacity

U_{cond} = Overall heat transfer coefficient of the condenser

A_{cond} = Heat transfer area of the condenser

and

$$\vartheta_m = \frac{\vartheta_i - \vartheta_o}{\ln(\frac{\vartheta_i}{\vartheta_o})} \tag{1.24}$$

where

$$\vartheta_i = T_{ref} - T_{cooler,in} \tag{1.25}$$

$$\vartheta_o = T_{ref} - T_{cooler,out} \tag{1.26}$$

which is the same approach as for the evaporators (see eq. 1.16) except for that the refrigerant temperature in the condenser is higher than the cooling media.

The heating capacity does also have relation to the mass flow rate of the refrigerant, by the following equations:

$$\dot{Q}_{cond} = \dot{m}_{ref} \cdot \Delta h_{ref} \tag{1.27}$$

Now it is possible to calculate the heating performance of the cycle, which is interesting in this project

$$COP_1 = \frac{\dot{Q}_{cond}}{\dot{E}_k} \tag{1.28}$$

where COP_1 = Heating performance

which also can be written as, for an ideal cycle:

$$COP_1 = COP_2 + 1 \tag{1.29}$$

Equation 1.28 is only true if all the heat rejected is useful, in other words:

$$\dot{Q}_1 = \dot{Q}_2 + \dot{E}_k \tag{1.30}$$

where $\dot{Q}_1$ = $\dot{Q}_{cond}$

$\dot{Q}_2$ = $\dot{Q}_{evap}$

(Department of Energy Technology, KTH, 2009)

2.3.4 Expansion device

The expansion phase of the heat pump has the main purpose of keeping the wanted pressure difference between the high and low-pressure side. Another purpose is to regulate the mass flow of the refrigerant in order to have the correct heat exchange in both the evaporator and condenser.

There are a few types of devices to consider:

<u>The hand expansion valve:</u>

This type is a hand-operated valve. The flow rate through the valve is dependent on the pressure differential across the orifice and how large the hand-operated valve opening is. There are some disadvantages for this type, for example if the load changes on the evaporator side, then the valve has to be operated manually, which could be difficult or not preferable at least.

<u>The capillary tube:</u>

The technique for this type is that the capillary tube, which has a small diameter relative to the other tube, is placed between the condenser and the evaporator. When the refrigerant reaches the capillary tube, the pressure drops because of the smaller diameter and the friction factor. This type is often used in domestic refrigerators and smaller heat pumps.

The automatic expansion valve:

This valve is used when the evaporator temperature is wanted to be constant, so therefore also the pressure the valve is wanted to be constant. The mechanism is automatic and therefore adjust the pressure difference by itself.

The thermostatic expansion valve:

These expansion types' main tasks are to maintain a constant superheating before the compressor. The thermostatic expansion valve consists of a metal membrane, a bulb and a spring. The bulb is filled with a refrigerant gas and is connected to the superheated vapour temperature before the compressor, which will give a force working against the pressure on the membrane. In order for the valve to open, the pressure above the membrane must be larger than the pressure on the bulb and the spring below the membrane. So as the load on the evaporator changes, the spring can be regulated in order to keep the same conditions.

The electronic expansion valve:

This type is a motor driven expansion valve, which is regulated by signals from a regulator. The regulator send signals as it sense both pressure and temperature differences.

4. Combined solar and heat pump systems

The reason for combining solar collector is to increase the overall efficiency of the system. The advantages with these kind of systems are many and various and depends a lot on heat loads, system design and control optimization. There are a lot of possibilities with different system designs and this chapter brings up and discusses some of them and the problems that can occur. Some on market solutions will be discussed as well.

4.1 Energy utilization

There are basically two main reasons to combine solar collectors with heat pumps

1. Low temperatures from the solar collectors can be utilized
2. Lower usage of heat pump

To utilize the energy in the solar collectors when the temperature is insufficient for direct heating or DHW this energy can be used to boost the temperature in the evaporator or condensor side of the heat pump. This way the efficiency of the heat pump goes up. Another way to use this low temperature energy could be to recharge the heat source of the heat pump. This requires that the heat source is able to absorb and store the energy that it's given so it can be used later. (Kjellsson 2009, 2004)

4.2 Heat transfer liquid

The heat transfer liquid in the heat pump loop has to be able to handle temperature down to about -10 °C (Kjellsson, 2004) without freezing. It should have high specific heat and thermal conductivity. Environmental aspects are also important due to the risk of leakage in case of damage to the pipe. As said in section 2.4 propylene glycol- water solutions are common in solar collector systems but due to environmental issues this solution is not to prefer in ground loops. Ethanol solutions are the most common solution in ground loops (see section 3.2.2) but it is not suitable for solar collectors due to explosion risk at stagnation temperatures. When combining the two systems there is a need for a solution that are applicable both to the solar collectors and the ground loop in order to avoid investments in a heat exchanger between the two systems. A rapeseed oil derivative can be used and the difference in performance from propylene glycol and ethanol solutions are visualized in table 1.

Heat carrier at 0°C	Freezing point	Heat conduction	Specific heat	Density
	°C	W/m,K	J/kg,K	kg/m3
Monopropyleneglycol 25%	-10	0,475	3930	1033
Monopropyleneglycol 33%	-17	0,45	3725	1042
Ethanol 25%	-15	0,44	4250	960
Rapeseed oil at 0°C	-15	0,436	3320	1106
Rapeseed oil at 40°C	-15	0,488	3510	1090

Table 1: *Properties of different heat transfer fluids (Kjellsson, 2004)*

4.3 Recharging

Recharging a bore hole or a well is when heat is transferred back to the heat source. In this case solar energy is used to heat up the source of the heat pump. The benefit from this is that when there is heat in excess or low temperature heat from the solar collectors it can be used to recharge the well. Ground heat sources of vertical type are most suitable for this purpose due to its relatively stable temperatures. When heat is taken from the well the temperature of the rock is decreasing, in case of a single bore hole this means that the temperature is decreasing a couple of degrees the first 3-4 years (Kjellsson, 2004). If there are multiple bore holes in the same area and the heat conduction of the rock is poor it is possible that they during the summer aren’t able to recharge naturally to the same temperature as the previous year, causing the cooled area to grow and the temperature in the wells to sink. An effect from this can be under dimensioned bore holes. This effect can be avoided by using the solar collectors for recharging of the well. Another application could be to drill shorter holes and compensated with recharging which would save money for drilling. This can also be used if the load has increased since the installation and the system has got under dimensioned. An important aspect when considering charging is if there is water in the well or not. If so the flow of ground water in the well is important and with high flows a lot of the charging may disappear quickly. On the other hand with high flows there is a fast natural recharging of the well instead. If the thermal conductivity of the rock is good a fast natural recharging takes place but the mechanic recharged heat will transfer faster to the rock. With recharged wells can achieve a higher temperature in rocks with lower thermal conductivity because the recharged heat is not transferred away as easy. Recharging is not effective if the temperature from the solar collector water is a lot higher than the surrounding rock temperature because the recharge heat is rapidly disappearing into the surroundings. It is also important to be aware of the pumping power for the recharge. Optimization and control are important to make sure that the pumping power doesn’t exceed the benefit from recharging.

4.4 Increasing the temperature of the evaporator side

When looking in to an ideal vapor compression cycle, a so-called Carnot cycle, it is easier to understand the relations between the evaporation and condensation. By assuming a Carnot cycle the following equations for the coefficient of performance can be stated:

Cooling performance:

$$COP_{2c} = \frac{T_2}{T_1 - T_2} \tag{1.31}$$

and heating performance:

$$COP_{1c} = \frac{T_1}{T_1 - T_2} \tag{1.32}$$

where T_2 = Evaporation temperature of the refrigerant

T_1 = Condensation temperature of the refrigerant

COP_{1c} = Carnot heating performance

COP_{2c} = Carnot cooling performance

When combining these two the following expression is derived:

$$COP_{1c} = COP_{2c} + 1 \tag{1.33}$$

In heat pump systems, a high value for COP_{1c} is desired. This can be achieved by trying to increase the COP_{2c} value. As seen from equation 1.31 above, this can be done by increasing the evaporation temperature, T_2. This will also give another positive effect, where the COP_{1c} itself will increase with T_2.

It is the expansion valve that performs the increasing of T2, where the pressure drop from the condenser to the evaporator can be decreased and the evaporation temperature will then rise. However, this puts a demand on the incoming temperature from the heat source in order to still provide enough heat for the cooling capacity. This can be understood by looking at the following equations:

$$\dot{Q}_{evap} = UA_{evap} \cdot \vartheta_m \tag{1.34}$$

where

$$\vartheta_m = \frac{\vartheta_i - \vartheta_o}{\ln(\frac{\vartheta_i}{\vartheta_o})} \tag{1.35}$$

and

$$\vartheta_i = T_{heat-source,in} - T_{ref} \tag{1.36}$$

$$\vartheta_o = T_{heat-source,out} - T_{ref} \tag{1.37}$$

It is clear now, that if a higher refrigerant temperature in the evaporator is wanted, the temperature for the heat source also have to be increased in order to maintain enough heat for the evaporation. The cooling capacity can also be calculated from:

$$\dot{Q}_{evap} = \dot{m}_{ref} \cdot \Delta h_{ref,evap} \tag{1.38}$$

Where the enthalpy difference is increasing with rising, T_2 (=T_{ref}).
Now if the evaporation temperature is increased, the enthalpy difference over the compressor will decrease, as these are directly depending on the refrigerant temperature.
The following equation can be used to calculate the compressor work:

$$\dot{E}_k = \dot{m}_{ref} \cdot (h_{1k} - h_{2k}) \tag{1.39}$$

where h_{1k} = Enthalpy after the compressor, depending on condensing temp.

h_{2k} = Enthalpy before the compressor, depending on evaporation temp.

If the heating capacity for the solar collectors is exceeded, this heat can be used to increase the incoming heat for the evaporator, by a simple heat exchange between the ground source collector fluid and the solar collector fluid.

4.5 Increasing the temperature of the incoming brine to the condenser

It could also be beneficial to decrease the temperature on the condenser side, in order to provide a higher temperature for the distribution circuit or to decrease the compressor work. From equation 1.32 the following table has been calculated:

Condensing temp. T1 [C]	Evaporating temp. T2 [C]	COP1c
50	0	6.5
60	0	5.55
70	0	4.9
80	0	4.4
90	0	4.0

Table 2: *Coefficient of heating performance for an ideal cycle at different condensing temperatures*

As seen from table above the COP_{1c} value will, as wanted, increase with decreasing condensation temperature.
By looking at the following expression:

$$\dot{Q}_{cond} = \dot{m}_{cooling} \cdot c_{p,cooling} \cdot (T_{cooling,out} - T_{cooling,in}) \tag{1.40}$$

where $T_{cooling1,2}$ =Temperature of the cooling brine in and out of the cond.

$\dot{m}_{cooling}$ = mass flow of the cooling brine

$c_{p,cooling}$ = specific heat of the cooling brine

By adding the following expression:

$$\dot{Q}_{cond} = UA_{cond} \cdot \vartheta_m \tag{1.41}$$

see section 3.3.3 for further explanation.
By combing the two, the following can be derived:

$$T_{cooling,out} = \frac{T_{cooling,in} - T_1 \cdot (1 - e^{\frac{UA_{cond}}{\dot{m}_{cooling} \cdot c_p}})}{e^{\frac{UA_{cond}}{\dot{m}_{cooling} \cdot c_p}}} \tag{1.42}$$

If assuming that every term on the right hand side are constant except for the temperature of the incoming cooling media, $T_{cooling,in}$, it can be seen that by increasing this temperature also the temperature of the outgoing cooling media will increase. With the restriction that exponential term must be larger than the value one.

From eq. 1.41 above, the following expression can also be derived:

$$T_1 = \frac{T_{cooling,in} - T_{cooling,out} \cdot e^{\frac{UA_{cond}}{\dot{m}_{cooling} \cdot c_p}}}{(1 - e^{\frac{UA_{cond}}{\dot{m}_{cooling} \cdot c_p}})} \quad (1.43)$$

By assuming that every term on the right hand side is constant except for the incoming temperature of the cooling media, $T_{cooling,in}$, it can be seen that the condensing temperature, T_1, can be decreased by increasing $T_{cooling,in}$. Also with the restriction that the exponential term is larger than one.

So there are two possibilities, either to increase the temperature for the space heating circuit with no extra compressor work needed, or to have this temperature constant and decrease the compressor work as the condensing temperature will decrease, see section 3.3.3.

So in conclusion, there are advantages with increasing the temperature of the incoming cooling brine for the condenser. This can be done by using excessive heat from the solar collectors and perform a heat exchange with the cooling brine before it enters the condenser.

4.6 System solutions

The solar energy can fundamentally be used in a few different ways depending on the load and the application. The schematics in this section are for showing the general idea on how the combined system can be used and not for showing the best way to connect the components.

4.6.1 The solar energy is only used for recharging the heat source

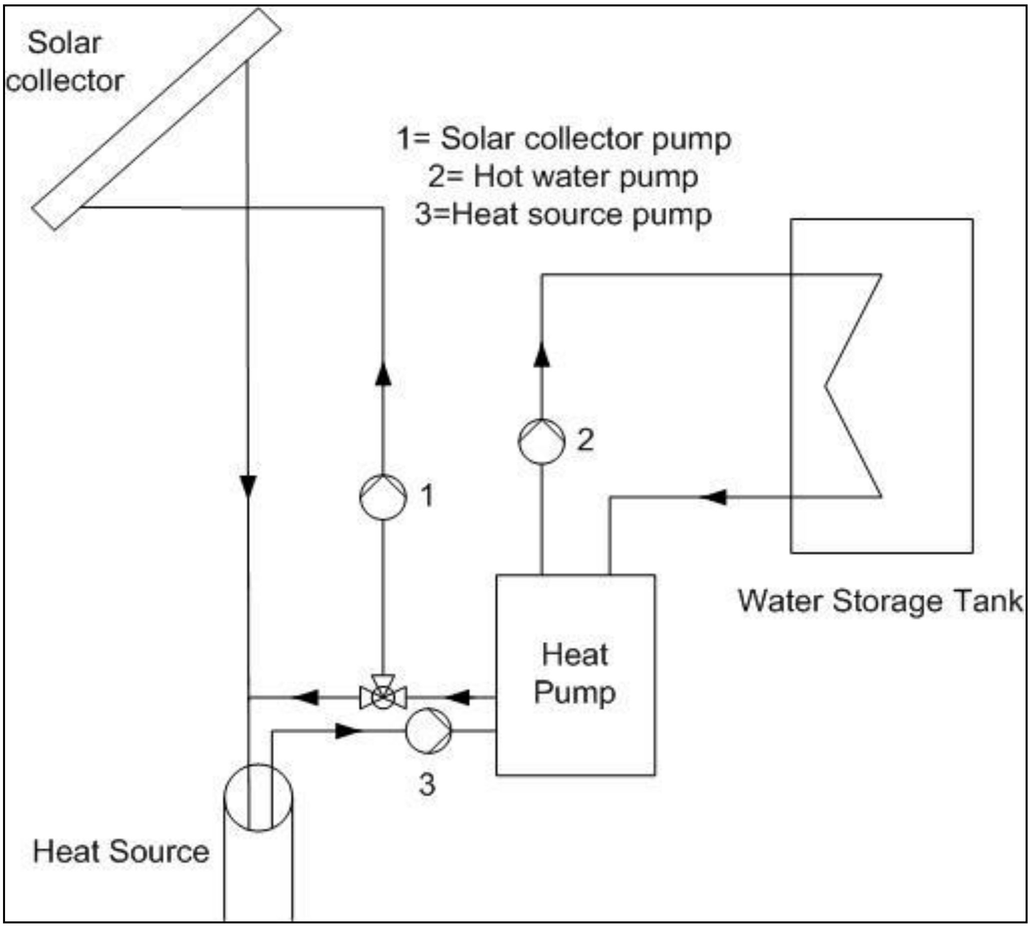

Figure 5: *A combined system where the solar collector are used to recharge the heat source*

In this case the solar collectors are connected to the same pipe as the heat transfer fluid from the heat pump. Other ways to do it is by have a separate loop for the solar collectors but then additional piping is needed for that loop. In this kind of schematic it is important to control that the pumping power does not exceed the benefit from recharging. In well dimensioned heat sources this system is often not profitable due to the cost for running the pump but when the heat source is under dimensioned or in an area where there is multiple heat sources nearby this solution could compensate for these problems. It is important to make sure that the temperature at the inlet of the compressor is not too high, although because the solar collectors runs through the heat source before entering the heat pump this should not be a problem during normal operation. Sensors that measures the temperature in the collectors and on the heat transfer fluid tells the circulation pumps when to start and stop.

4.6.2 The solar energy is used for preheating the evaporator side of the heat pump and for recharge the heat source.

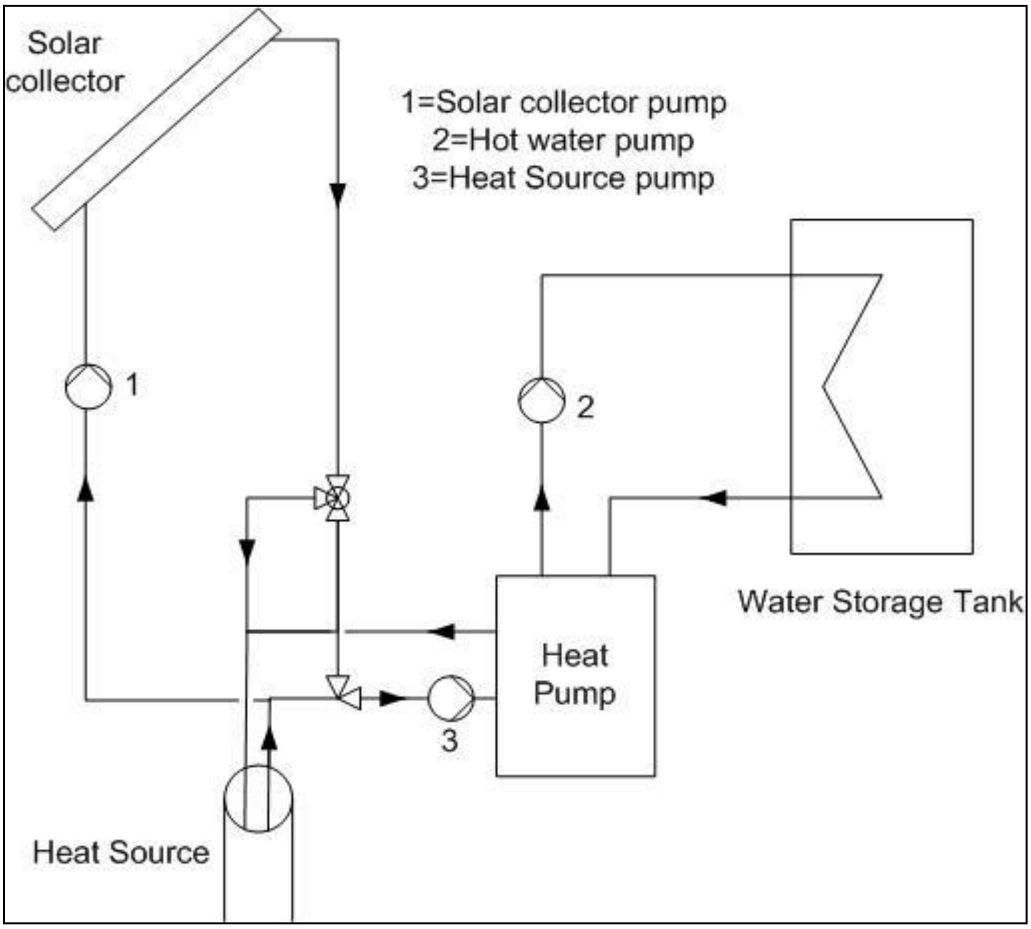

Figure 6: *A combined system where the solar collector are used to recharge the heat source or boost the evaporator side of the heat pump*

In this schematic the hot fluid from the solar collector are connected to both the inlet and the outlet of the heat pump. As shown in section 4.4, with an increased temperature of the heat source, better heat performance can be achieved. This can be done by boosting the temperature from the heat source with the solar collectors as shown in figure 6 with the hot fluid from the solar collector connected with the fluid from the heat source at the inlet of the heat pump. When boosting the inlet to the evaporator the outlet temperature will increase as well and recharging will occur when the fluid is transferred back to the heat source.

4.6.3 When the solar energy is sufficient it is used for domestic hot water

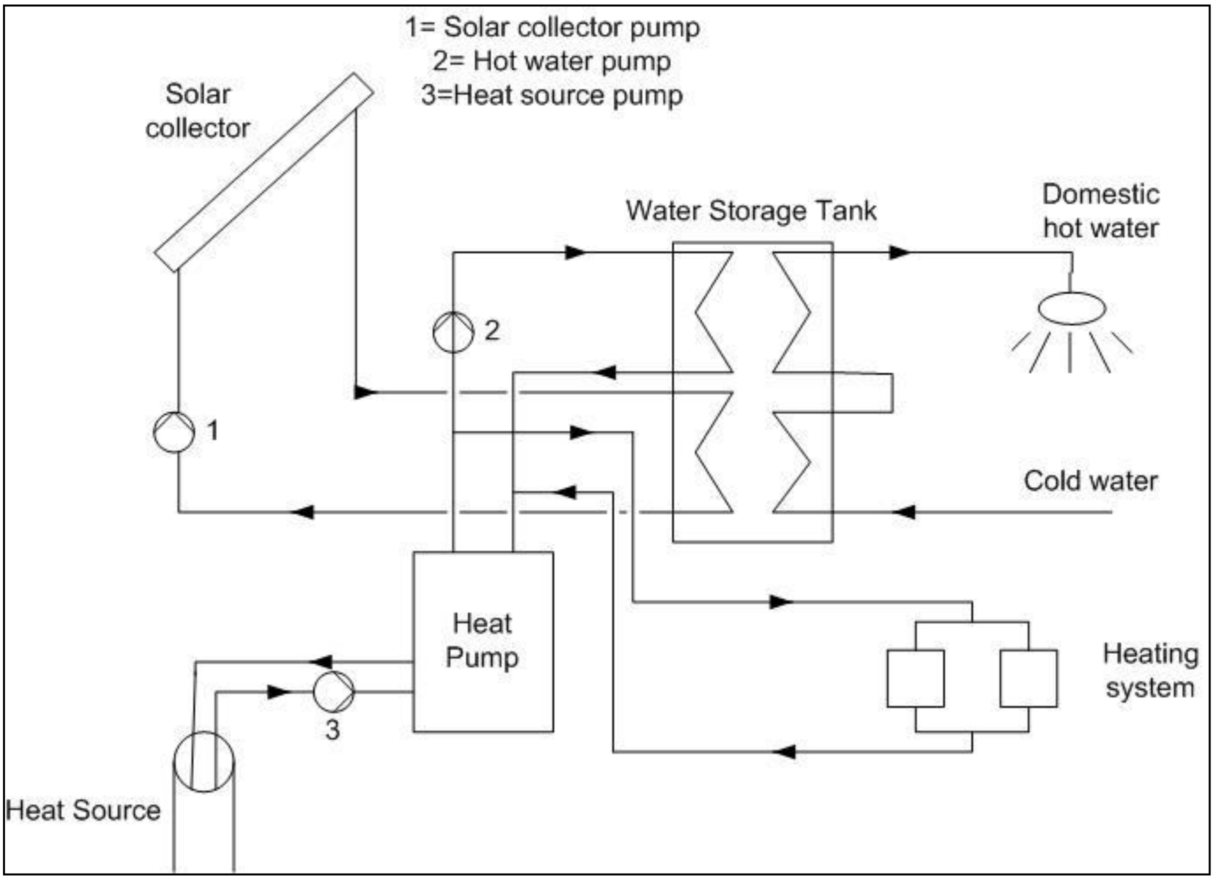

Figure 7: *A combined system where the solar collector are used for domestic hot water*

The heat pump is providing the house with heat and as long as the solar collectors provide a sufficient amount of heat they will provide the house with domestic hot water, in this way, the solar power is helping the heat pump and in this way decrease the operation time for the heat pump. In this specific schematic one water storage tank is used but there are other solutions with two or more tanks being used, e.g. one for domestic hot water and one for the radiator system. The solar collectors have the largest capacity during the summer (see section 2.1) and are often able to cover the demand for domestic heat water during the summer months. Because it is warm outside there is no or small heating demand, the heat pump can then be shut off during this months and increase the efficiency of the system. Because there is no outtake from the heat source for this period the natural recharging will be faster in this case than during constant operation.

4.6.4 When the solar energy is sufficient it is used for both domestic hot water and heating

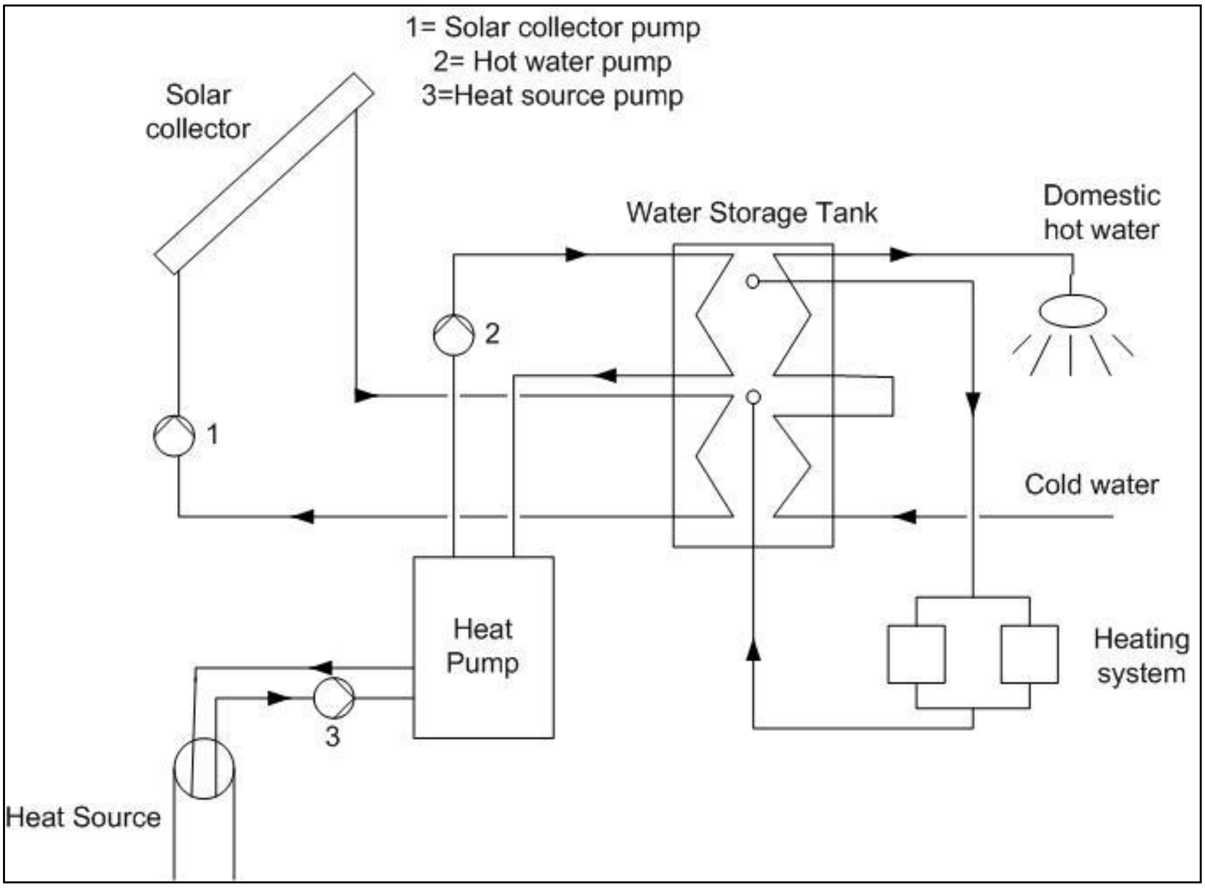

Figure 8: *A combined system where the solar collector are used for domestic hot water and for the radiator system*

This system is almost as system nr 3 except that the solar collectors can be used for heating as well. The connection circuit can be made in various ways in these kinds of system, for example with multiple tanks and/or an external heat exchanger for the solar collector loop. This kind of system implies for a larger solar collector system but can unburden the heat pump in larger extent.

As seen in section (3.3.3) the outtake temperature of the domestic hot water and the water for the radiator system are in the same range.

4.6.5 When the solar energy is sufficient it is used for both domestic hot water and domestic heating, otherwise it is used to boost the evaporator or conderser side of the heat pump and for recharging of the heat source.

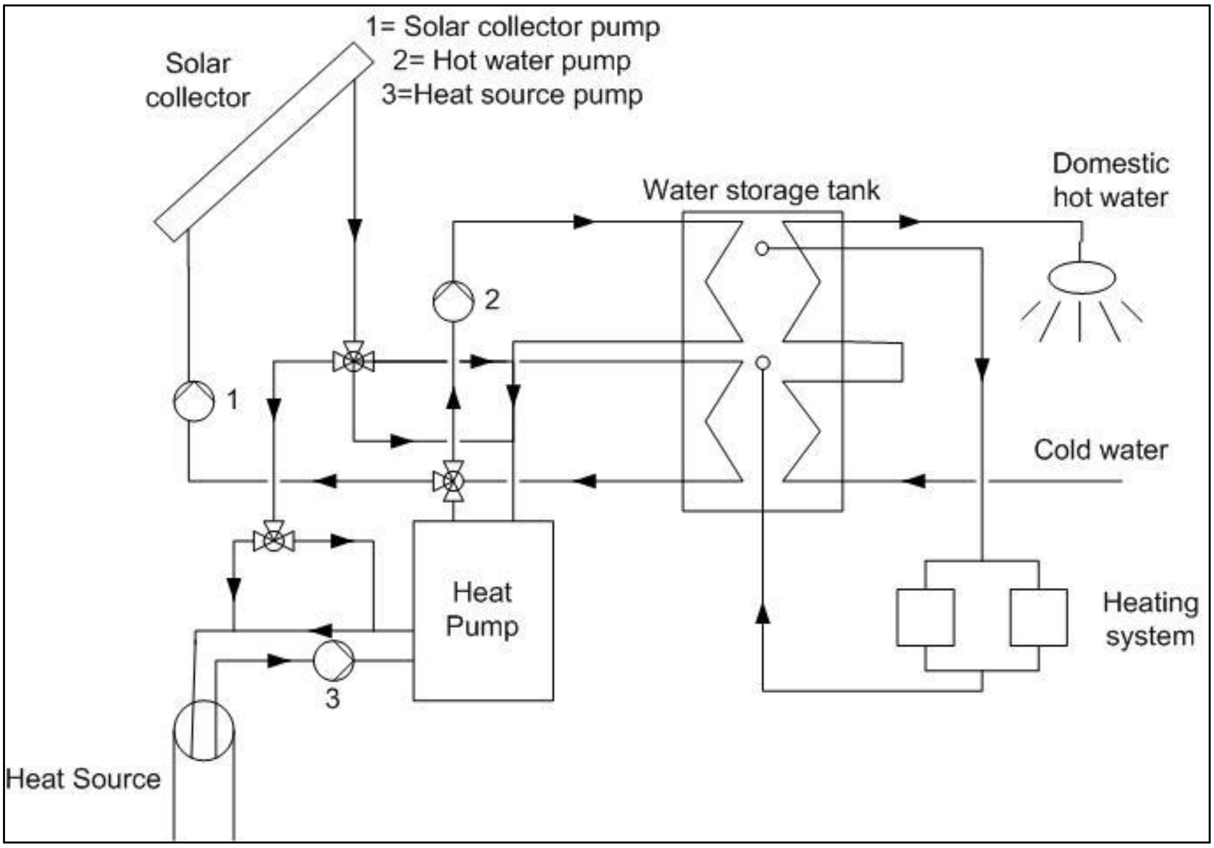

Figure 9: *A combined system where the solar collector can be used for domestic hot water, radiator system, recharging the heat source and/or boosting the evaporator or the condenser side of the heat pump*

This is the most flexible system where the solar collectors can be used in many different ways depending on the available temperature from the collectors. At high temperatures they can be used for heating and domestic hot water, when the temperature is not enough for that they can be used to increase the ingoing temperature on the condenser side of the evaporator. At even lower temperatures they can be used for boosting the evaporator side of the heat pump or/and recharge the heat source.

4.6.6 Complexity

The schematics mentioned shows just the general idea of how to use the solar energy and the possible connection alternative is many. These systems are very complex and needs accurate and detailed investigations before deciding for one specific solution, both when it comes to control and optimization of utilization of the solar energy but also how to connect the components in the system in order to minimize investments and maximize efficiency.

4.7 On market solutions

There is some companies that offers combined system solutions for single family dwellings. This section mentions some solutions that different companies offers on the market.

4.7.1 Eviheat

A Swedish company that offers a solar heat pump, a combination solution between solar collectors and heat pump. The solar collectors can be used in multiple ways, if the temperature of the collector water is over 50 °C it can be used to heat the house and for domestic heat water. If the water gets colder it can first be heat exchanged to preheat the fluid from the heat source (and therefore decrease the energy usage) and thereafter sent down to recharge the drilled well.

When the temperature in the collectors gets to high during summer heat can be “dumped” in the drilled well causing recharge of the well and cooling of the collectors.

When there is no sun insulation the heat pump is working on its own but since the well is charged with heat the heat capacity increases.

(Eviheat, 2010)

4.7.2 Sol & Energiteknik

Another company that offers a lot of different connections in combined solar and heat pump systems.

Some of their system is:

1. Solar collector system is used separately in a closed loop to be heat exchanged in a storage tank. Heat is used from the collectors for domestic hot water whenever there is sufficient heat from the collectors and a ground source heat pump provides the heating of the house and domestic hot water when the collector temperature are insufficient.
2. Same as 1. except that the collectors are used for both heating and domestic hot water by mixing solar heated water and heat pump water to get the desired temperature for heating.
3. Same as 2. except that excess heat is used to charging the heat source.
4. Same as 1. except that excess heat is used to charging the heat source.
5. Solar collector system is used for domestic heat water whenever possible but also to increase the temperature of the heat source fluid.

They have also some specific applications, e.g. swimming pool heating.

(Solenergiteknik, 2010)

4.7.3 Grup Romet

Grup Romet is a Romanian company that offers a solution where the solar collectors are directly coupled to the domestic water heat exchanger. The remaining heat energy is then exchanged with the space heating system that is coupled to buffer tank. To this buffer tank an external heat source is connected. This company offers a solution with an air-water heat pump

but also have a suggestion with a gas burner. In the description it also states that an outside temperature of -20 °C would be sufficient to heat a house with this technique.
(Grup Romet, 2010)

5. Modeling/Simulations

A simplified model containing a combined system with heat pump and solar collectors has been developed. This section deals with the model documentation, the development of the model and the simulation of the model. The software used to develope the model is MATLAB.

5.1 Purpose:

The purpose of the model is clarify the potential with solar collectors combined with heat pumps, to give general directions on when different temperatures occurs from the solar collector and show different ways to use them.
The model is in steady state and is calculated for every hour in one year.

5.2 The model

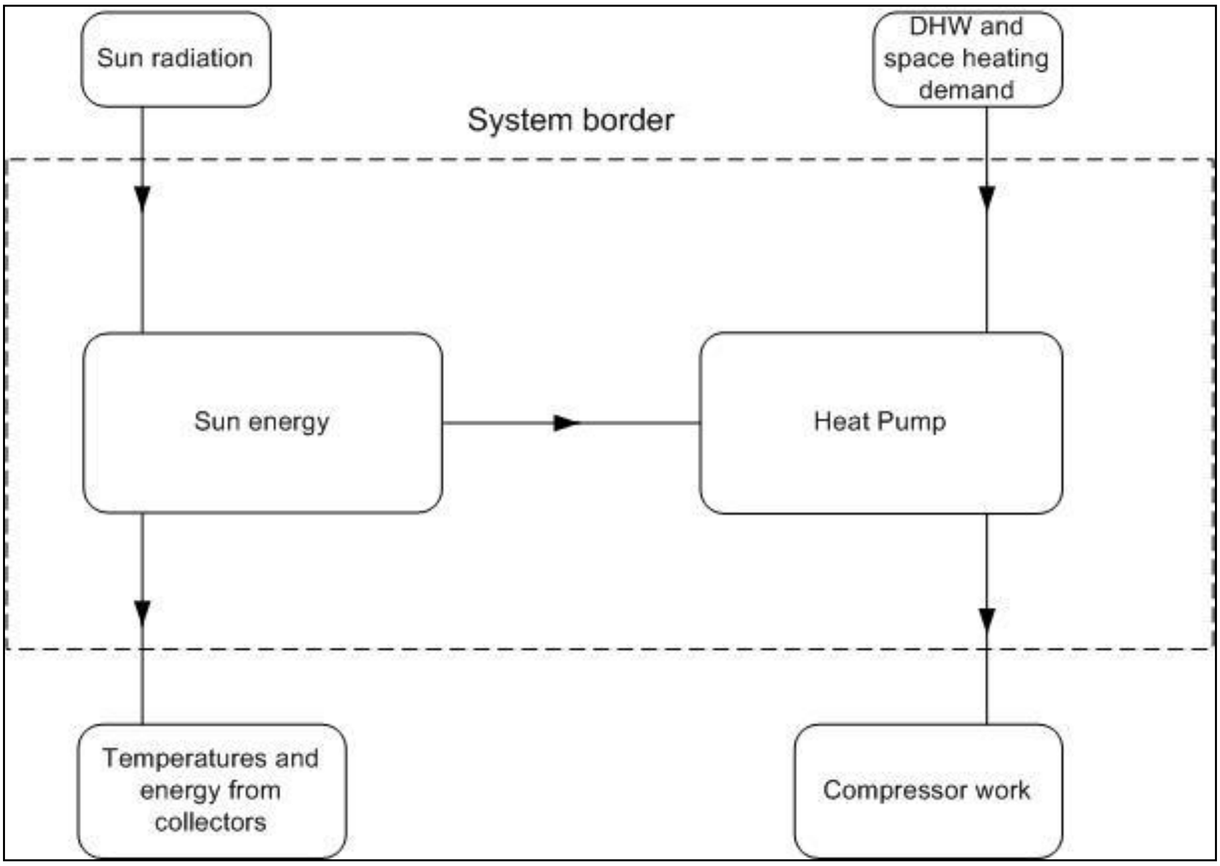

Figure 10: *Schematic of the model*

A schematic of the model is shown in figure 10. Solar radiation, DHW and space heating demand are inputs to the system while compressor work and temperatures and energy from the collectors are output.

Information about the model

- If nothing else is mentioned, the basic simulation case is with solar collector area of 10 m^2, collector inlet temperature of 5°C, tilt of the solar collectors is 45° and Stockholm is the location of the system.
- No storage is considered in the model. That means that no solar energy can be stored during the day and used for example during the night.
- The heating load from the house is a given input and not calculated. Values are taken from (Kjellsson, 2009) and is average for each month.
- The domestic hot water usage is assumed to be the same for all days during the year. Values are taken from (Energimyndigheten 2009b)
- The heat pump in the system is assumed to work at constant condensing and evaporating temperatures with fixed subcooling and superheat.
- Solar radiation and ambient temperature data is created in TRANSYS.
- The temperature of the water leaving the solar collectors are divided into three different levels.
    1. If the temperature is higher than 55 °C it can be used directly to cover the DHW.
    2. If the temperature is between 30 °C and 55 °C it can be used for covering the space heating demand.
    3. If the temperature is lower than 30 °C it cannot be used to cover any heating demands, but can be connected to either the heat pump to rise the COP or to recharge the borehole.

5.3 Subsystems

The combined solar collector and heat pump system is divided in these subsystems:

5.3.1 Sun radiation

This model has direct, diffuse solar radiation on horizontal surface as an input and is calculating the solar radiation on a tilted surface for every hour in a year. The calculations are made in accordance with eq. 1.1 – 1.6 (Williams, 1983).

Assumptions:

- Stockholm is the location of the solar collector during simulations
- The solar collector is pointing south in the simulations
- The reflection from the ground is assumed to be 30 percent.
- The sky is seen as a uniform radiator of diffuse solar radiation.
- The zenith angle cannot be larger than 88 ° in the model because higher values of zenith angle causes unreasonable high values of the solar radiation due to division with numbers close to zero.

Subsystem results:

The distribution between direct, diffuse and reflective is shown in figure 11. As expected, the direct solar radiation has the highest tops while the diffusive is more evenly distributed

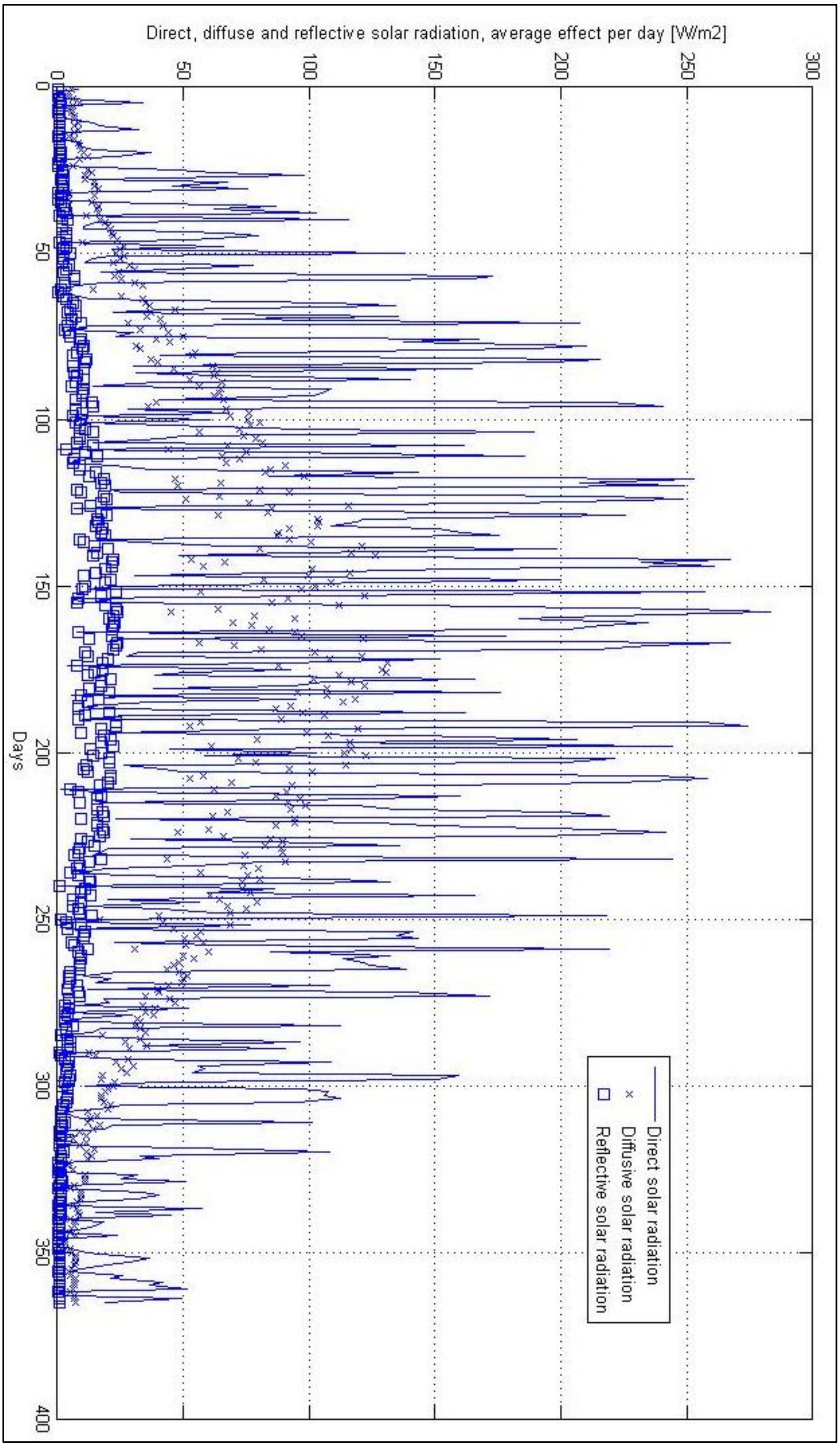

Figure 11: *Direct, Diffuse and reflective solar irradiation on a tilted surface of 45 °*

The tilt of solar collector makes it possible to collect more energy from the sun and when optimizing the tilt of the collector based on maximum energy output for a year it is shown in figure 12 that 45 degrees is the best option. This doesn't mean that it is the optimal tilt for every system because other factors e.g. when and how much the energy extraction should be during throughout the year or how the load patterns has to be taken under consideration.

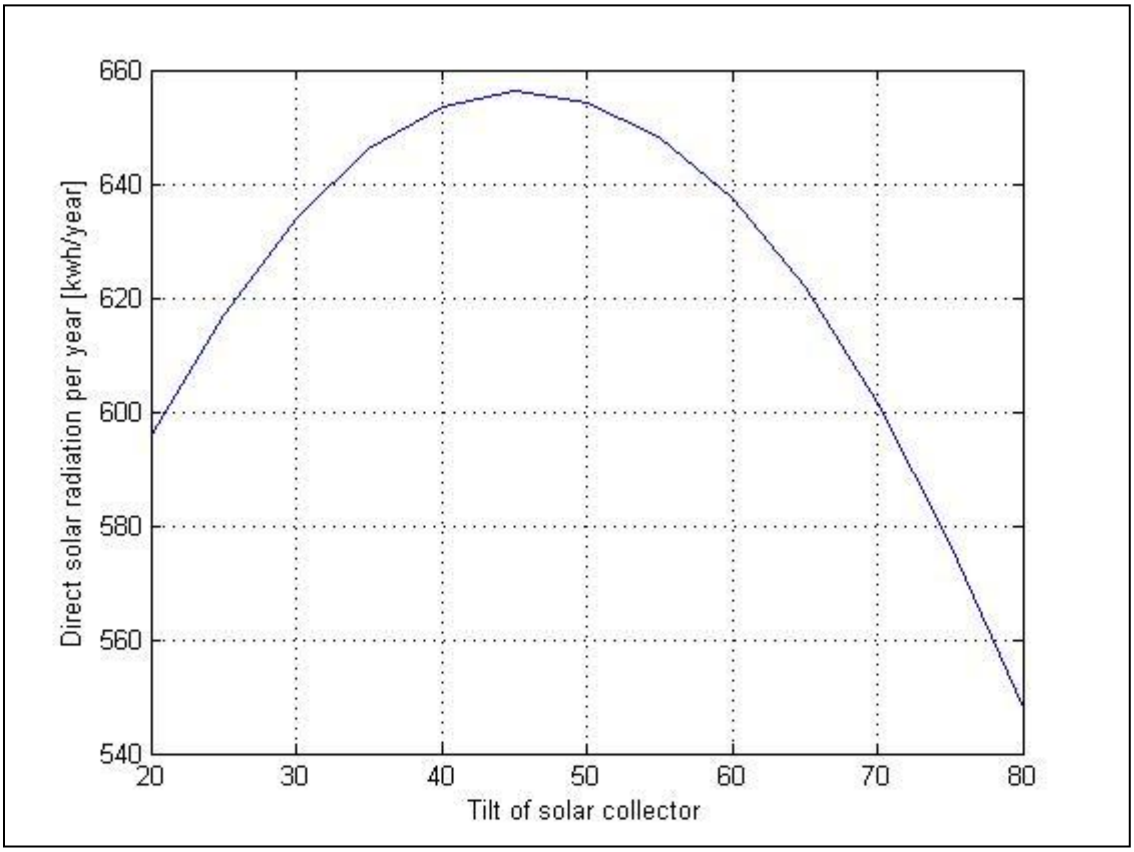

Figure 12: *Direct solar radiation as a function of the tilt of the solar collector*

In figure 13 it is shown that the diffuse solar radiation is decreasing with increasing tilt of the collectors. This is due to the fact that the view factor from the solar collectors to the sky decreases with decreasing diffuse solar radiation as a result. The opposite can be seen with the reflective solar radiation as its view factor to the ground increases with increasing tilt.

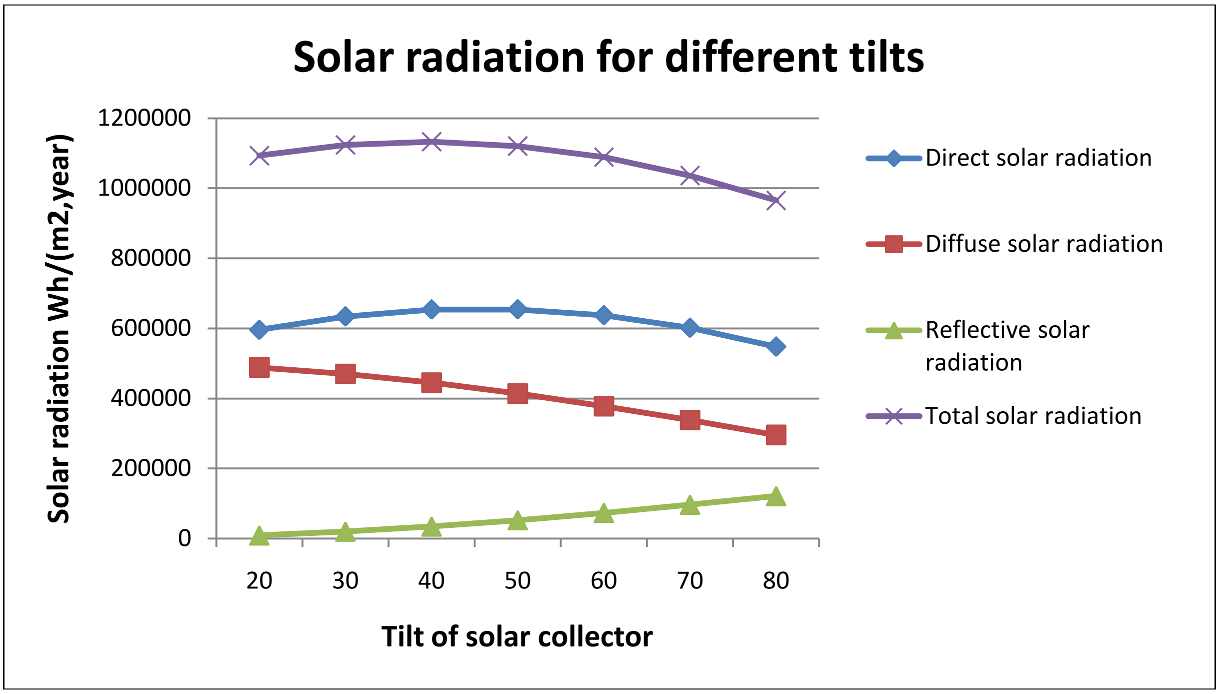

Figure 13: *Solar radiation as a function of different tilt of the solar collector*

5.3.2 Sun Energy

Input to this model is the solar radiation that is calculated in the sun radiation model and also outside temperature data. The outputs are the temperature and energy from the solar collector for every hour.

The heat transfer fluid is assumed to be rapeseed oil with properties from table 1. It is assumed that it has constant specific heat and this gives:

$$Q_{out} = \dot{m}_{htf} Cp_{htf} (T_{out} - T_{in}) \qquad (1.44)$$

where $\dot{m}_{htf}$ = mass flow of the heat transfer fluid

Cp_{htf} = specific heat of the heat transfer fluid

T_{out} = the temperature out of the solar collector

T_{in} = the temperature in to the solar collector

This equation together with equation 1.13 makes it possible to calculate the temperature out of the solar collector and the useful energy output.
A solar collector is chosen from literature (Williams, 1983) but values can be changed in the model.

Assumptions:

- A low flow system with a mass flow of 3 g/(s m^2) is chosen. The mass flow in fixed during one simulation (one year).

- The inlet temperature for the solar collector is assumed to be a known input and constant during one simulation.
- The heat transfer fluid has a constant specific heat through the solar collectors.
- No energy is extracted from the solar collectors if the difference between the inlet and the outlet temperature is 5°

Subsystem results:

As shown in table 3 temperatures reaches over 55°C for only 3,6 % of the hours in a year. As seen in the table the percent of time when there is no solar energy is increasing with the tilt. In a real system the output temperatures can be regulated by increasing or decreasing the mass flow through the solar collector. This means that depending on the demand, different temperatures can be produced. E.g. if the DHW demand is high a lower mass flow can be circulated through the solar collectors in order to produce high temperatures (T>55°C).

Beta	T>55	45<T<55	30<T<45	10<T<30	No solar
20	2,9 %	3,4 %	7,7 %	21,7 %	64,3 %
30	3,4 %	3,5 %	7,8 %	20,9 %	64,4 %
40	3,6 %	3,6 %	7,6 %	20,7 %	64,7 %
50	3,6 %	3,4 %	7,4 %	20,7 %	64,9 %
60	3,2 %	3,4 %	7,4 %	20,8 %	65,2 %
70	2,5 %	3,5 %	7,4 %	21,0 %	65,5 %
80	1,5 %	3,4 %	7,3 %	21,8 %	65,9 %

Table 3: *Outlet temperature from the solar collector for different solar collector tilts [Percentage of hours in a year with corresponding temperature outlet]*.

Different inlet temperature to the solar collector affects the system in different ways. In Table 4 it is seen that an increasing inlet temperatures results in a decrease in both total solar energy for a year and solar collector efficiency. The losses from the solar collectors is governed by the difference between the average temp in the solar collectors and the ambient temperatures and with higher inlet temperatures the average temperature gets higher which lead to higher losses and lower efficiency. Although with higher inlet temperatures, the percentage of high output temperatures from the solar collectors is increasing.

In the model the inlet temperature is constant during one simulation (one year) in a real system, the inlet temperature will be determined by the temperature of the temperature at the bottom of the hot water storage tank. The storage tank is therefore a important factor in a real thermal solar system as the efficiency as well as the output energy is depending on how well stratified the tank is.

T_in	T>55	45<T<55	30<T<45	10<T<30	No solar	Solar energy Wh/year	Solar efficiency
0	2,6 %	3,0 %	6,8 %	26,9 %	61,5 %	9 240 452	85 %
5	3,6 %	3,5 %	7,5 %	20,7 %	64,7 %	8 577 833	76 %
10	4,7 %	3,8 %	8,5 %	15,8 %	67,2 %	7 995 192	71 %
15	5,9 %	4,4 %	9,6 %	10,8 %	69,3 %	7 459 264	67 %
20	7,5 %	4,7 %	10,8 %	5,6 %	71,4 %	6 955 120	64 %
25	8,9 %	5,3 %	12,8 %	0,0 %	73,0 %	6 497 536	63 %
30	10,8 %	5,9 %	15,2 %	0,0 %	68,1 %	6 211 207	64 %

Table 4: *Total solar energy for a year, solar efficiency and outlet temperature from the solar collector for a year for different solar collector inlet temperatures [Percentage of hours in a year with corresponding temperature outlet]*.

As seen in table 5 the solar collector produces most energy with a tilt of 30-50° and are in the same range producing the most of the high temperature energy, T>30. As the high temperatures can be used for heating and DHW(if T>55) this energy is the most useful energy and as seen in the table this energy is about 70 % of the total energy for 50<Beta>30.

Beta	Solar energy Wh/(year)	Solar energy T>30	Solar energy T<30
20	8 345 122	5 750 165	2 594 957
30	8 572 133	6 020 264	2 551 870
40	8 634 213	6 136 329	2 497 884
50	8 543 677	6 058 664	2 485 013
60	8 309 714	5 836 010	2 473 704
70	7 919 782	5 456 337	2 463 445
80	7 391 802	4 837 141	2 554 661

Table 5: *Total solar energy and solar energy for a year when the temperature is above and below 30 °C, with different inlet temperatures to the solar collector. [Wh/year]*

Figure 14 shows when there is a possibility to boost the evaporator side (30>T<10) during the year. As the figure shows it is possible to boost almost all year around with the maximum during the summer. If this energy can be used it will increase the performance of the system since this energy is not useful for heating or DHW. The average temperature in this temperature range is about 18°C in the model. That means that the evaporator can be boost up against 15°C.

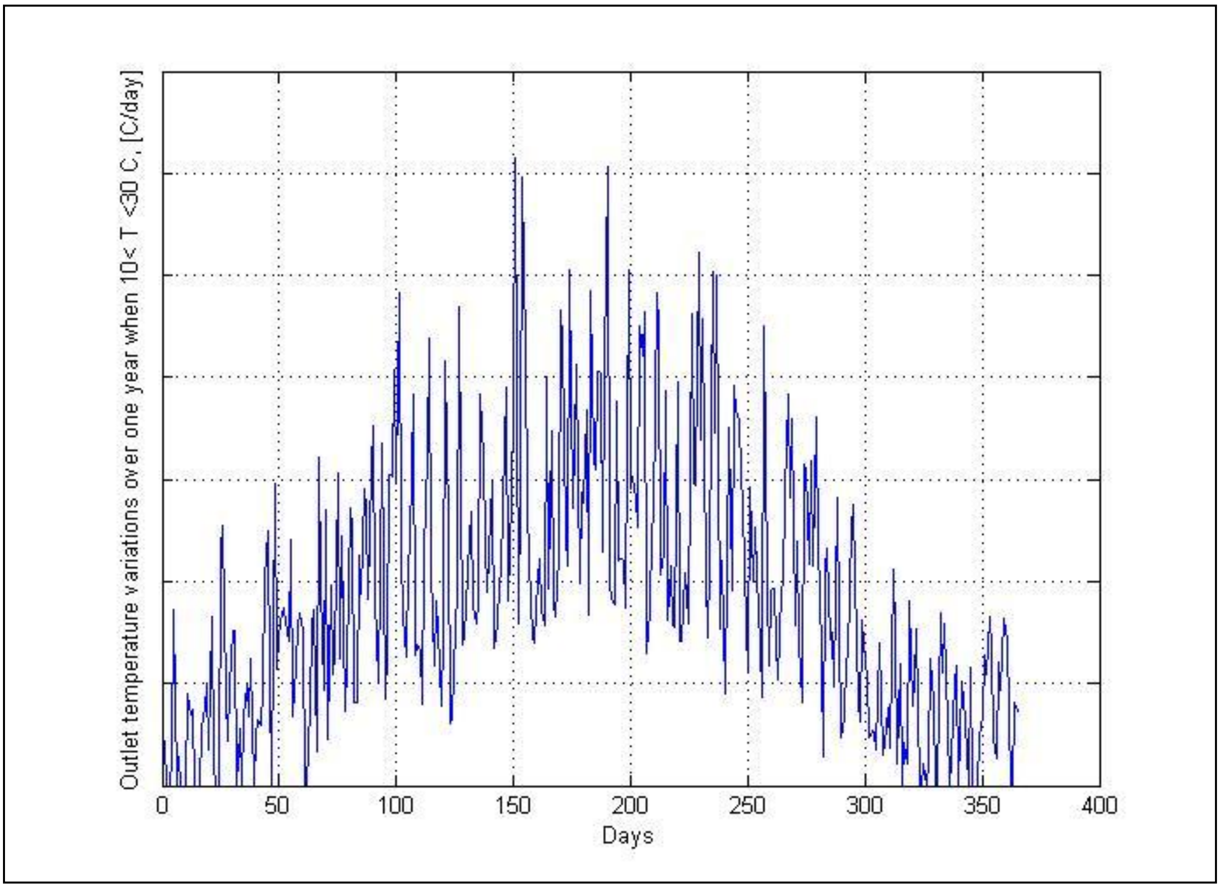

Figure 14: *Yearly variation of output temperature between 10 and 30 °C from the solar collectors*

When the output from the solar collector is high temperature energy (T>55°C) it is possible to use this energy for both DHW and heating. Figure 15 shows when the output from the solar collector is >55°C during a year and as seen there is no high temperature output during the winter and a lot during the summer.

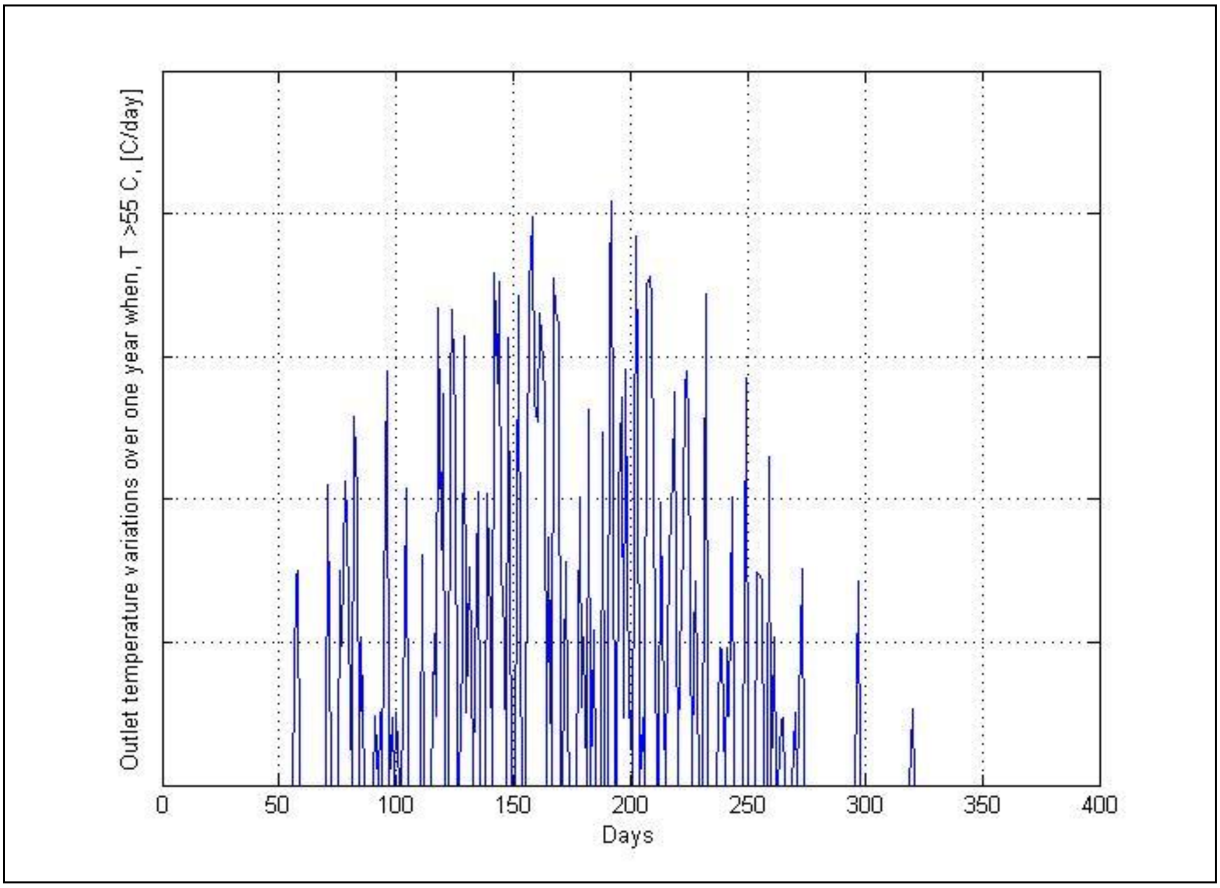

Figure 15: *Yearly variation of output temperature over 55 ° from the solar collectors.*

Figure 16 shows the amount of energy that is available for boosting the evaporator side during the year. In the winter this amount is approximately 5 kWh/day and 15 kWh/day in the summer. Depending on the load, this energy can be used as boosting or in the case of no load, for recharge.

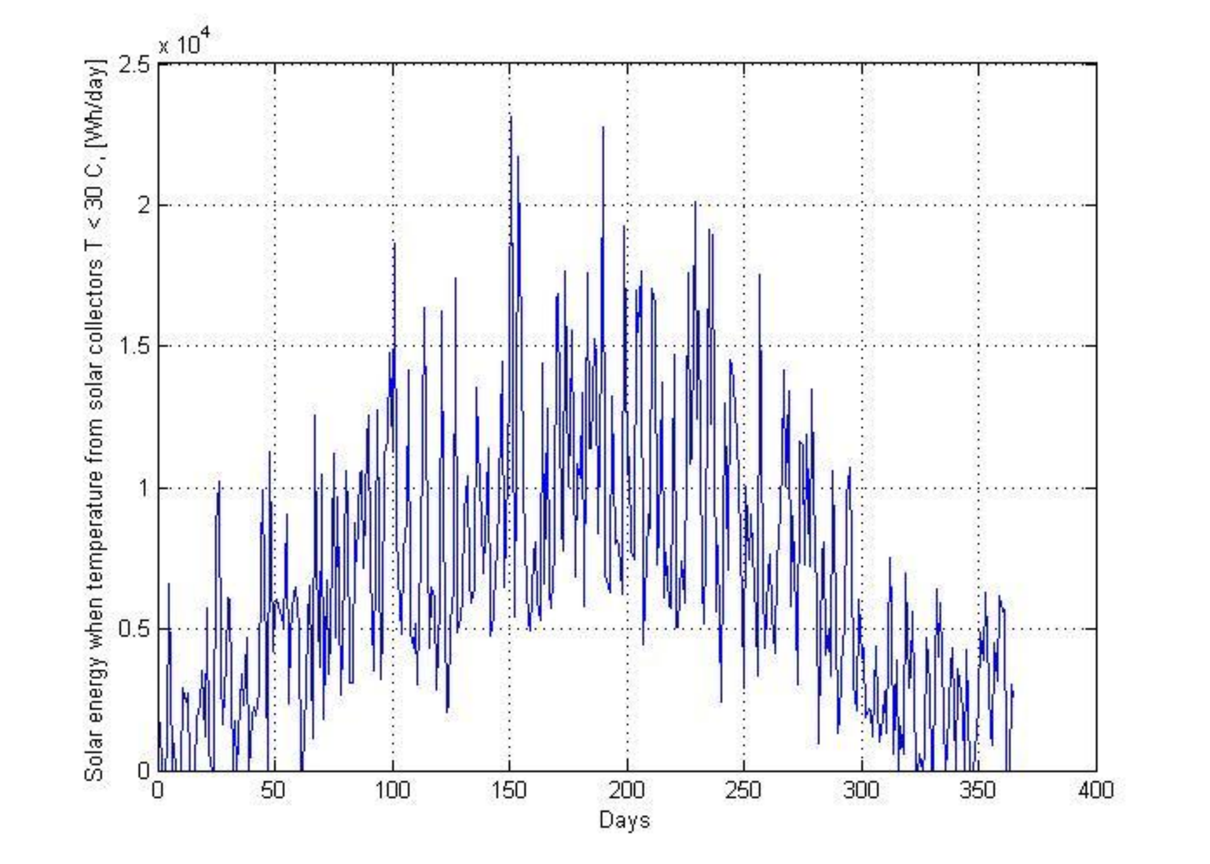

Figure 16: *Output energy from the solar collector when the outlet temperature is under 30℃.*

The amount of energy that is available for heating is when the outlet temperature from the solar collectors in over 30°C. In figure 17 it is shown that there is a small amount of solar energy available for heating during the winter months (0 and 15 kWh per day) and a large amount during the summer months (40-60 kWh/day)

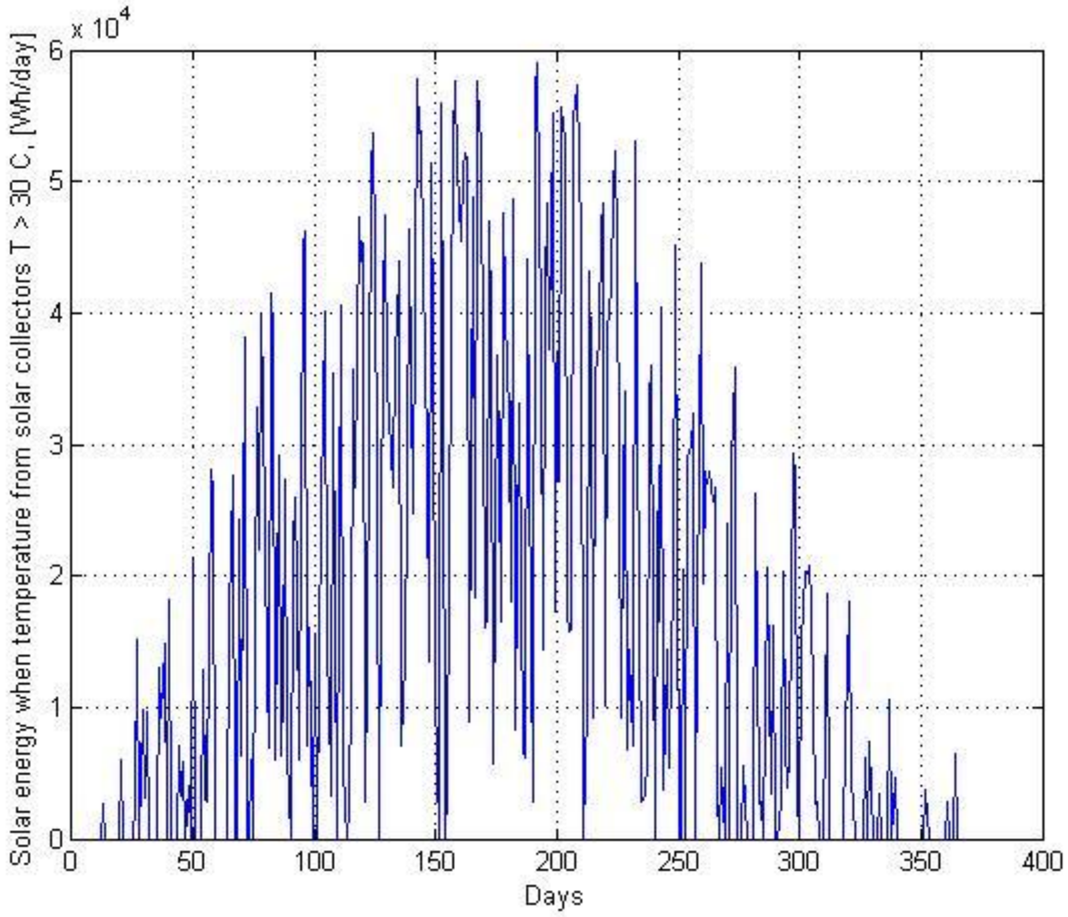

Figure 17: *Output energy from the solar collector when the outlet temperature is over 30℃*

As seen in figure 18 the amount of energy output when outlet temperatures is over 55°C is zero during the winter months and during the rest of the year it varies between 1,5-4,5 kWh/day.

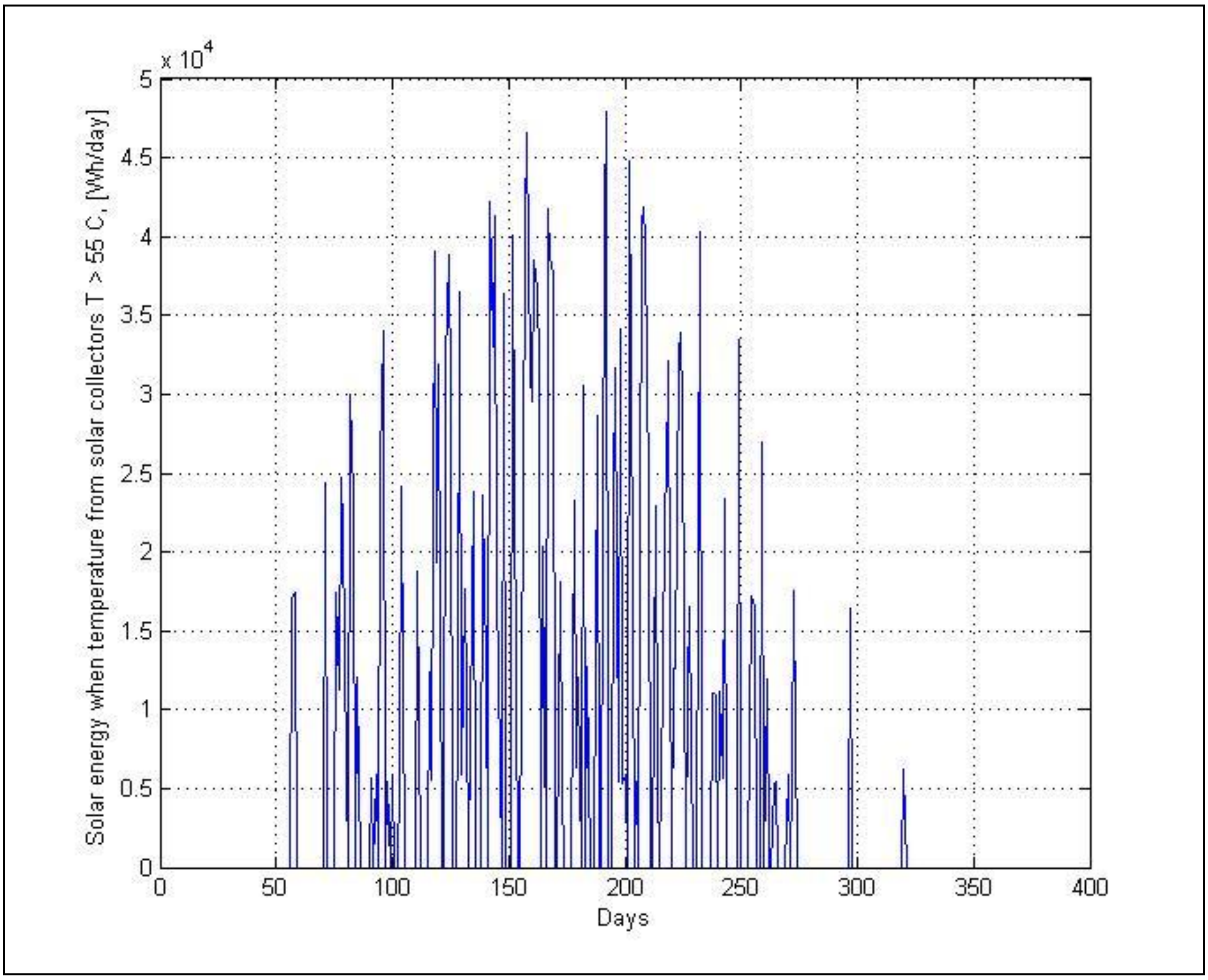

Figure 18*: Energy from the solar collectors when the outlet temperature is larger than 55℃*

5.3.3 Energy Demand

This model has monthly averaged heating demand and a daily DHW demand as input and is calculating the heating and DHW demand for every hour of the year. DHW demand is 144 liters/day (Energimyndigheten, 2009b) and the daily space heating demand varies between 500 Wh during the summertime and 5500 Wh during the wintertime. (Kjellsson, 2009)
These values are only valid for a house situated in Sweden.

Assumptions:

- Temperature for DHW is assumed to be 55°C (Energimyndigheten, 2009b))
- Cold water inlet to the DHW is assumed to be 8,6 °C (Energimyndigheten, 2009b))
- Constant specific heat of water
- Constant density for water
- The space heating demand is assumed to be constant over the day for the whole month, so it only varies monthly.

Subsystem results:

Figure 19 and 20 present the values that were used in the model regarding DHW and space heating demand. No calculations have been made in this section, except for that the values for space heating demand have been created to be constant over one month.

In figure 18, the daily fluctuations in domestic hot water usage can be seen. These variations have been created with help from statistics made by the Swedish energy agency. As the figure shows, the highest usage is during the evening and during the night it is assumed to be no usage. This profile is constant over the whole year, which means that there will always be a heating demand for temperatures over 55 °C, for the system to cover.

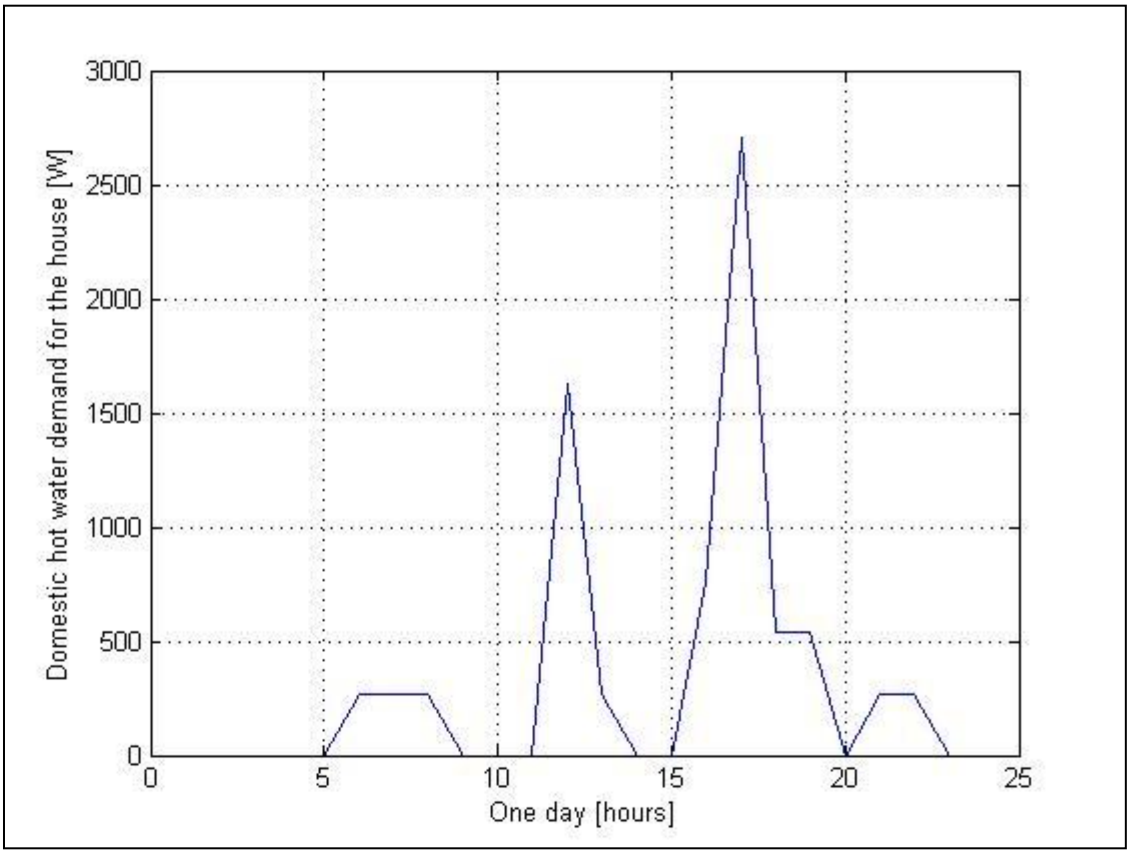

Figure 19*: Domestic hot water demand for 24 hours*

Figure 19 presents the yearly variations in space heating demand, which is created with values taken from (Kjellsson, 2009). As explained in the assumptions, there are only monthly variations that are taken into account, which also explains the plateaus in the graph. The figure is also showing that there is a small space heating demand during the summer.

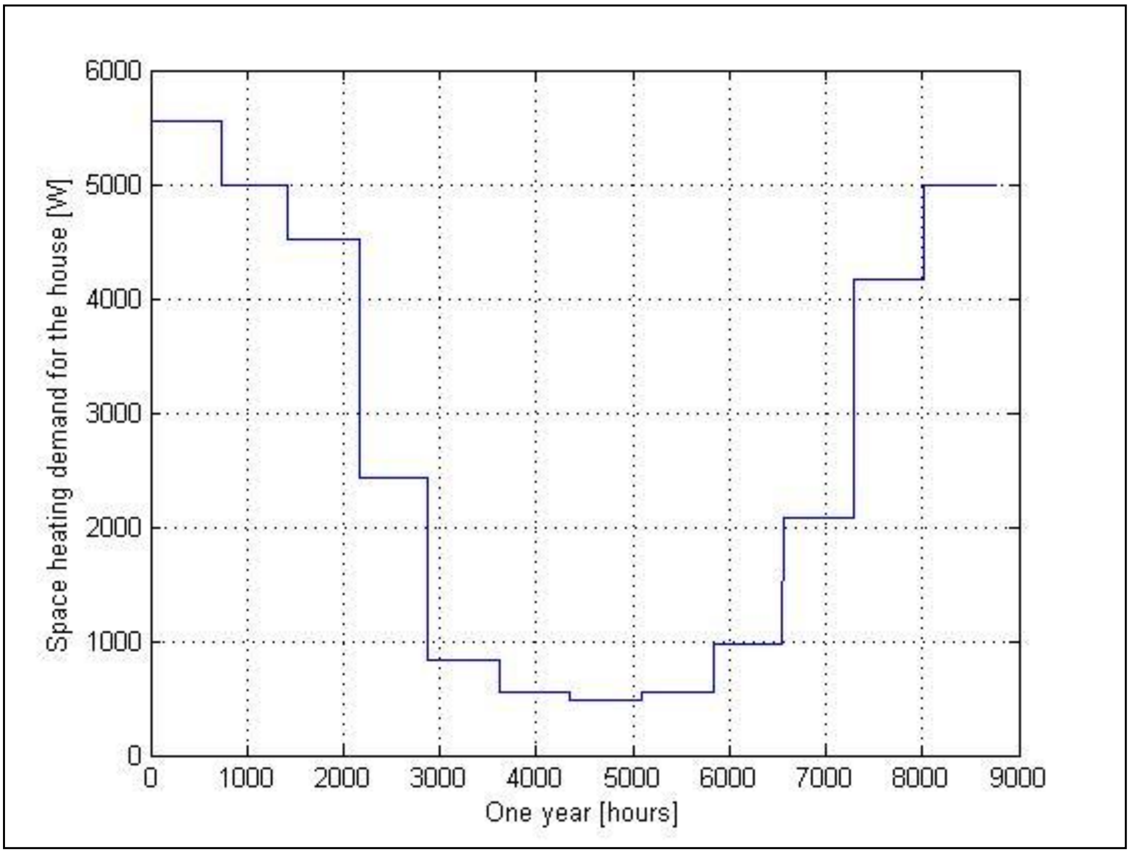

Figure 20: *Space heating demand for one year*

5.3.4 Heat pump

This sub-model has sun energy and total energy demand as input and is calculating the compressor work and excessive solar energy. The heat pump has R134a as a working fluid and it is working with fixed temperatures. The condensing temperature is 60°C with 5°C subcooling. The evaporating temp is 0°C with 10°C superheat.
The heating capacity for each hour depends on the energy demand from the house and the available sun energy. Depending on the temperature from the solar collector this energy can be used for DHW, space heating, boosting the temperature on the evaporator side or recharge of the borehole.

Assumptions:

- Constant operating temperatures for the heat pump
- Isentropic efficiency of the compressor is 0.9

Submodel results.
In figure 21, the compressor work required is presented as a function of days over a whole year. It is easy to see the connection between the amount of heat needed from the heat pump and the trend of the annual, total heating demand. This could be explained by the fact that the available sun energy during the summer is much higher than during the winter, and which is

almost sufficient to cover the space heating demand, which itself almost is negligible. Still though, there is a constant requirement for DHW, which the solar collectors seldom could cover.

In this model a heat storage tank is not considered, whereby there would be no heat available from the solar collectors during the nights and cloudy days as this heat not can be stored, so the heat pump will always be needed at least to cover the DHW demand.

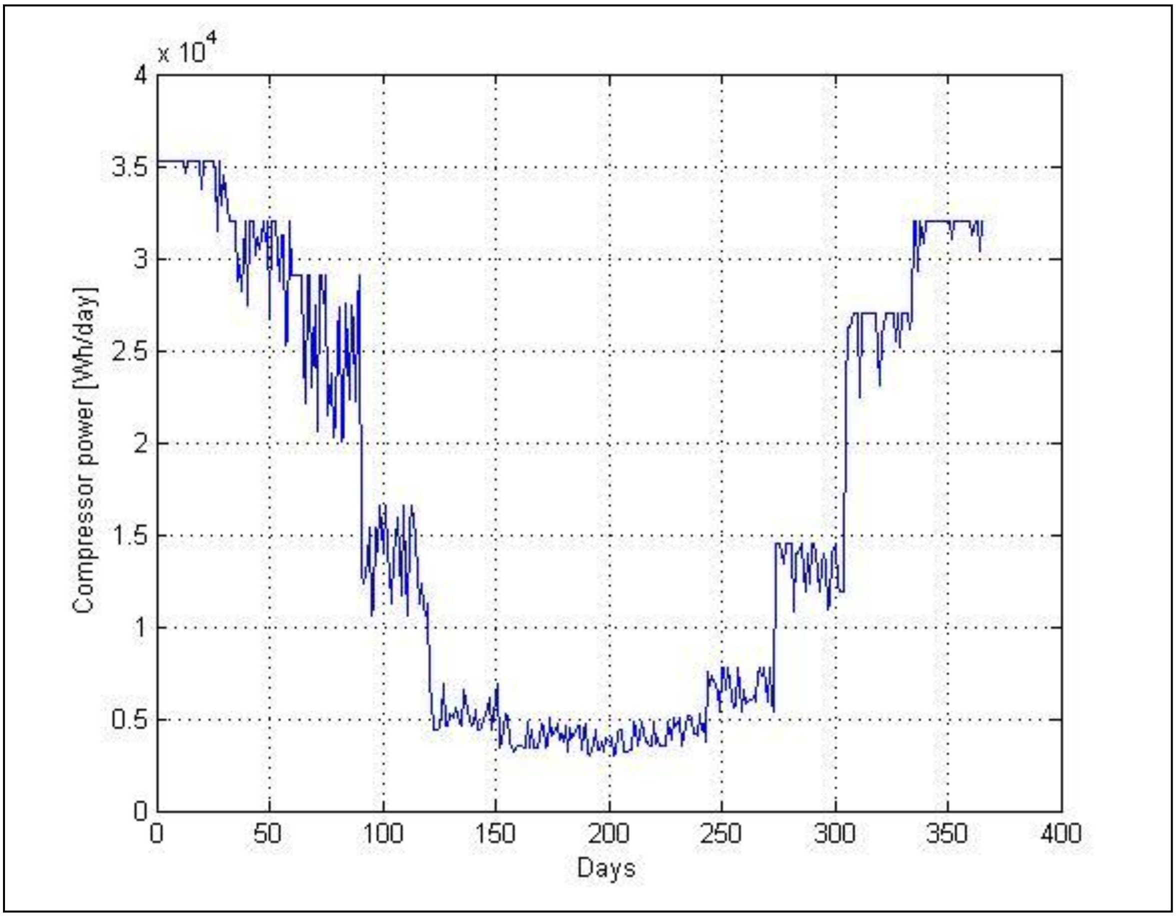

Figure 21: *Compressor work during one year*

During the days where the available solar energy is larger than the total heating demand, see figure 22, there will be some excessive energy that could be used in different ways in order to increase the efficiency of the system. For example, if the heating demand is very small than the excessive energy could be used to recharge the borehole, especially if there also is a small demand on the heat pump.

Another situation could be that when there is excessive solar energy, but the temperatures leaving the solar collector is not sufficient for heating, this temperature could be used to boost the temperature of the heat source before entering the evaporator. By doing this the evaporation temperature is increased with a higher COP of the system as a following advantage.

A third technique is to store all the excessive heat in a storage tank, as explained earlier. That way the usage of the heat pump could be decreased very much during the nights and cloudy days.

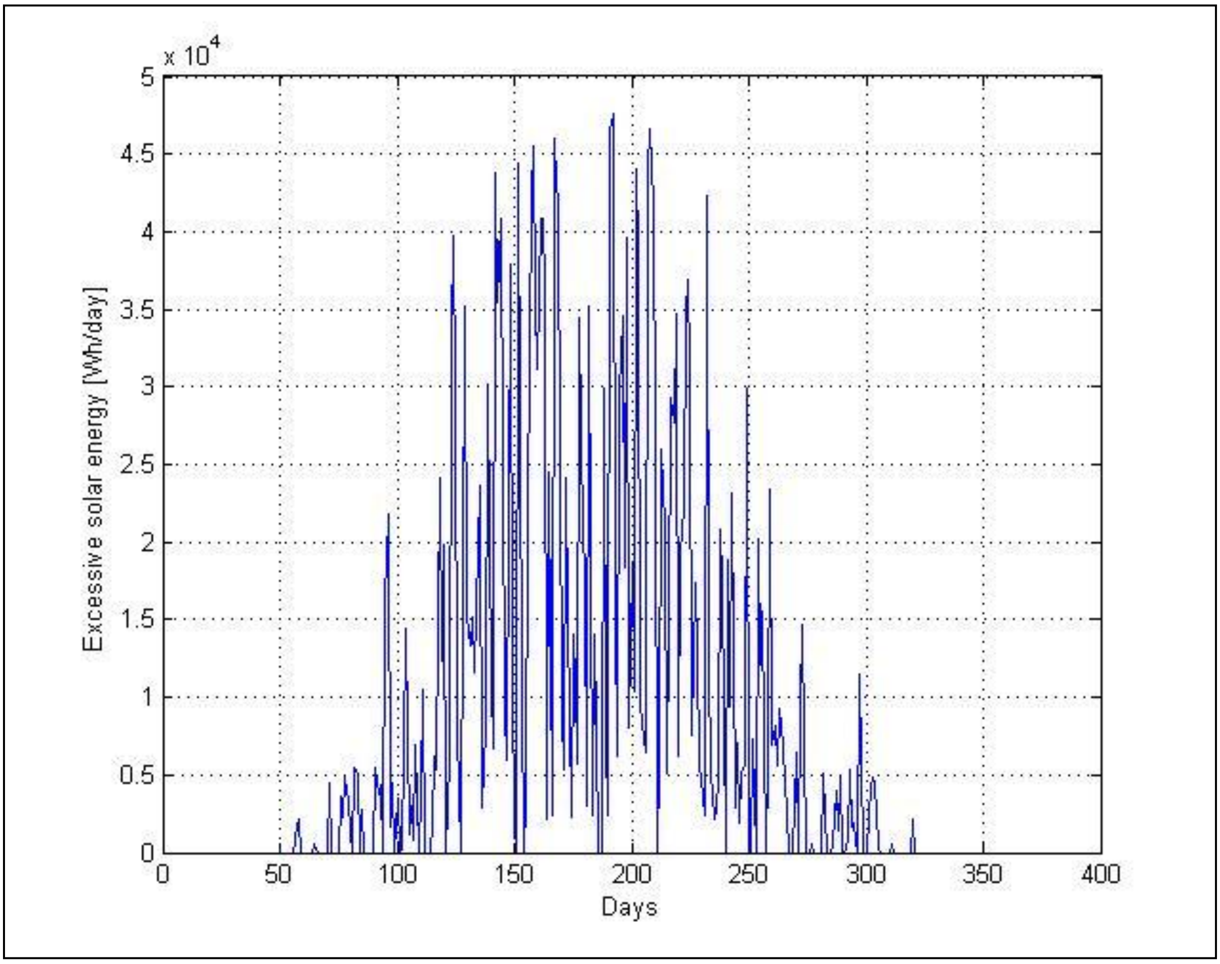

Figure 22: *Excessive solar energy during one year*

In table 6, the compressor and excessive solar energy is presented as a function of the angle of the solar panel, and an area of which that is $10m^2$.
By looking at the values for the compressor power, it is seen that the optimum angle where the least compressor work is required is around 60°, compared to table 5, where the optimum angle was found to be around 40°. So there is clearly an optimization to account for, where the system could work at its optimum. Which means that, the compressor work is small and that the produced solar heat is as high as possible.
But as in this model, where a storage tank is not considered, the main factor to consider is the compressor work, so for this system the angle should be around 60°.
Table 7 shows the same as table 6, but with twice as large collector are which is 20 m^2. It can be seen that the excessive solar energy is around twice as large as for an area of 10 m^2, but the compressor is almost the same. But again, if there is no storage tank considered, this would not be a profitable investment, but if a storage tank is considered it could absolutely be profitable. Especially as the required compressor work most certainly would decrease.

Beta	Compressor power	Excessive solar
20	6 185 427	1 301 886
30	6 131 726	1 503 957
40	6 102 569	1 563 446
50	6 089 202	1 521 110
60	6 083 896	1 297 294
70	6 100 273	941 931
80	6 131 346	513 402

Table 6*: Compressor work and excessive solar for one year vs. tilt of the solar collector with a solar collector area of 10m^2 [Wh]*

Beta	Compressor power	Excessive solar
20	6 164 193	2 905 124
30	6 099 551	3 392 649
40	6 064 187	3 580 225
50	6 050 756	3 544 225
60	6 042 328	3 083 566
70	6 057 313	2 314 341
80	6 087 292	1 326 001

Table 7: *Compressor work and excessive solar for one year vs. tilt of the solar collector with a solar collector area of 20m^2 [Wh]*

Table 8 presents the same as table 6 and 7, but is now a function of the temperature entering the solar collector. It shows what is expected, and that is that the compressor power will decrease with higher temperature of T_in and the excessive solar energy will increase. Both is a consequence of that the heat produced from the solar collector and the temperature leaving it, is increased.

T_in	Compressor power	Excessive solar
0	6165407	1230489
5	6095682	1568182
10	6023196	1844710
15	5954922	2091524
20	5887323	2381370
25	5838244	2491210
30	5826581	2587424

Table 8: *Compressor work and excessive solar for one year vs. tilt of the solar collector.*

During the winter there is a low solar radiation and as seen in figure 23 there is only output from the solar collectors 3 times in the first week of January. The energy output is low temperature output which means that it cannot be used for DHW or heating. Since the load is constant at about 6kW the heat pump has to work all the time. This makes boosting of the evaporator to a good alternative for the low temperature energy from the collectors.

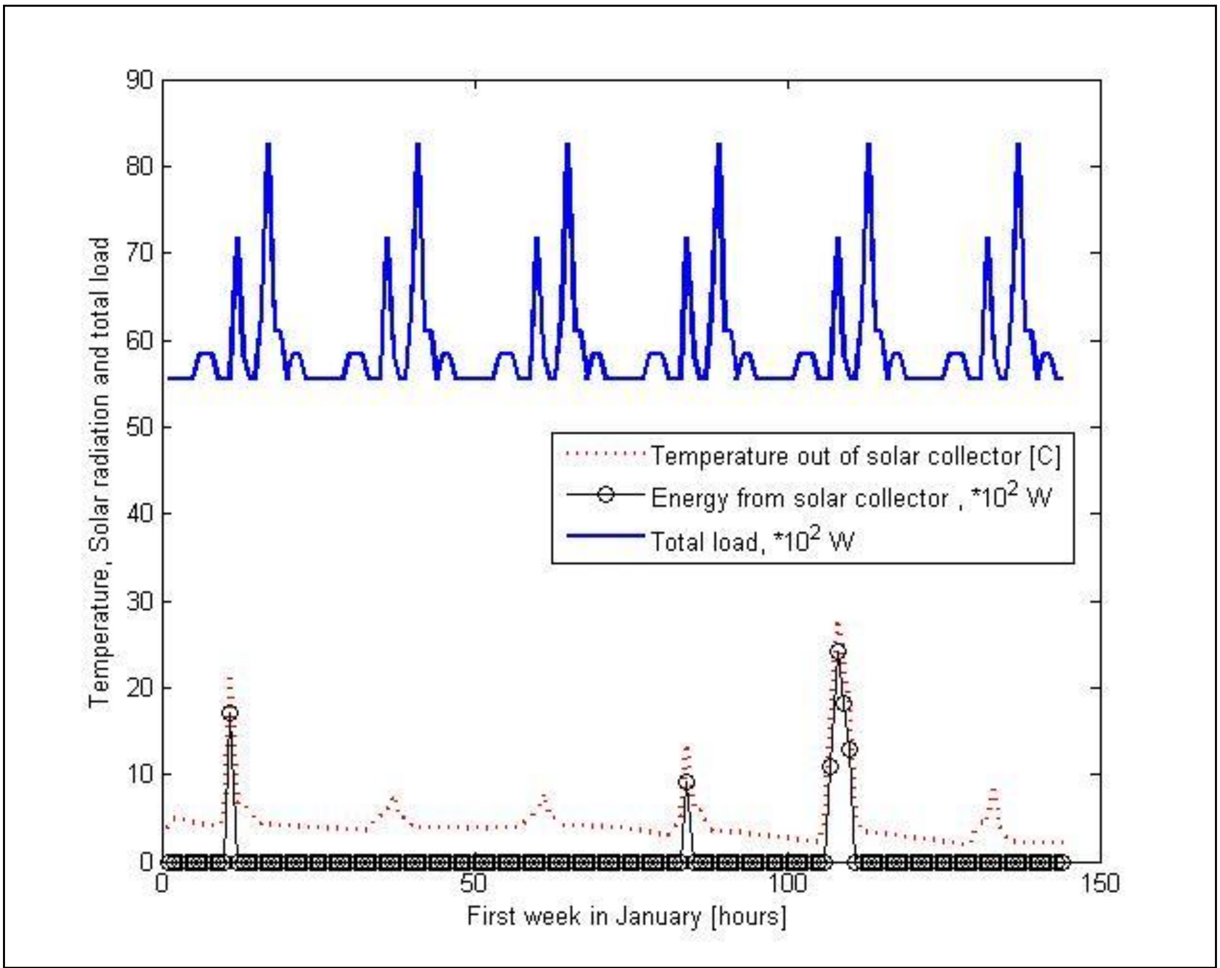

Figure 23: *Temperature and energy out of the solar collector and total load for one week in January*

Due to higher solar radiation during spring and autumn there are other possibilities. As seen in figure 24 the solar energy exceeds the total load around midday but it less than the load during the rest of the day. This brings a lot of different operating conditions for the system. When there is no solar energy the heat pump does all the work by itself, but as solar radiation produces low energy heat this energy can be used to boost the evaporator (e.g. point 1-3 for the energy from solar collector in figure 24). As the solar radiation increases during the day, high energy heat can be extracted from the solar collectors, and this energy can be used both for heating of the house and DHW (if temp >55°C, e.g. point 4 for solar energy). In this model the difference between the solar energy and the load is considered as a redundancy and can only be used to recharge the borehole. In a real system with storage this energy can be saved and used later that day when/if the load exceeds the solar energy. When the solar energy temperatures if below 55°C but above 30°C this energy can be used only for heating of the house (e.g. point three day two).

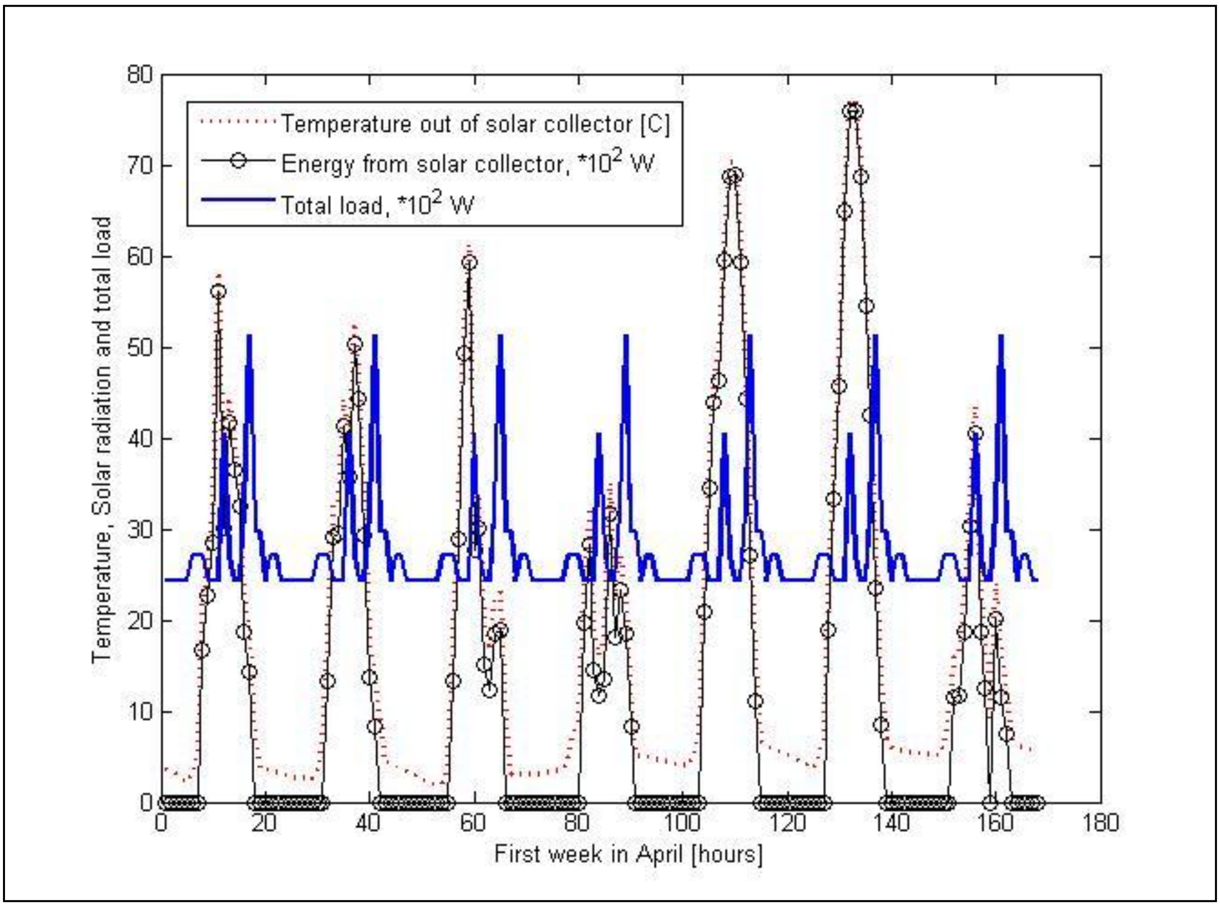

Figure 24: *Temperature and energy out of the solar collector and total load for one week in April*

Since pump work is not included in this model the electrical input to the system consists only of the energy needed for the compressor. The energy output from the system is always the demand from the house. This relation is shown in figure 25 and it can be seen that the maximum compressor work during the winter is about 40 kWh/day and the load is at that time approximately 140 kWh/day. This results in an energy ratio of 3,5. During the summer months the compressor work is about 0,5 kWh per day and the load around 2,5 kWh/day and this results in a energy ratio of 5. The higher ratio in the summer months is a result of the high solar energy during this period. The fact that the load is constant at about 0,5 kWh/day is an affect of the absence of storage, since this storage could store the redundant solar energy during the day and use it when there is a demand. This would probably reduce the compressor work to almost nothing during the summer months.

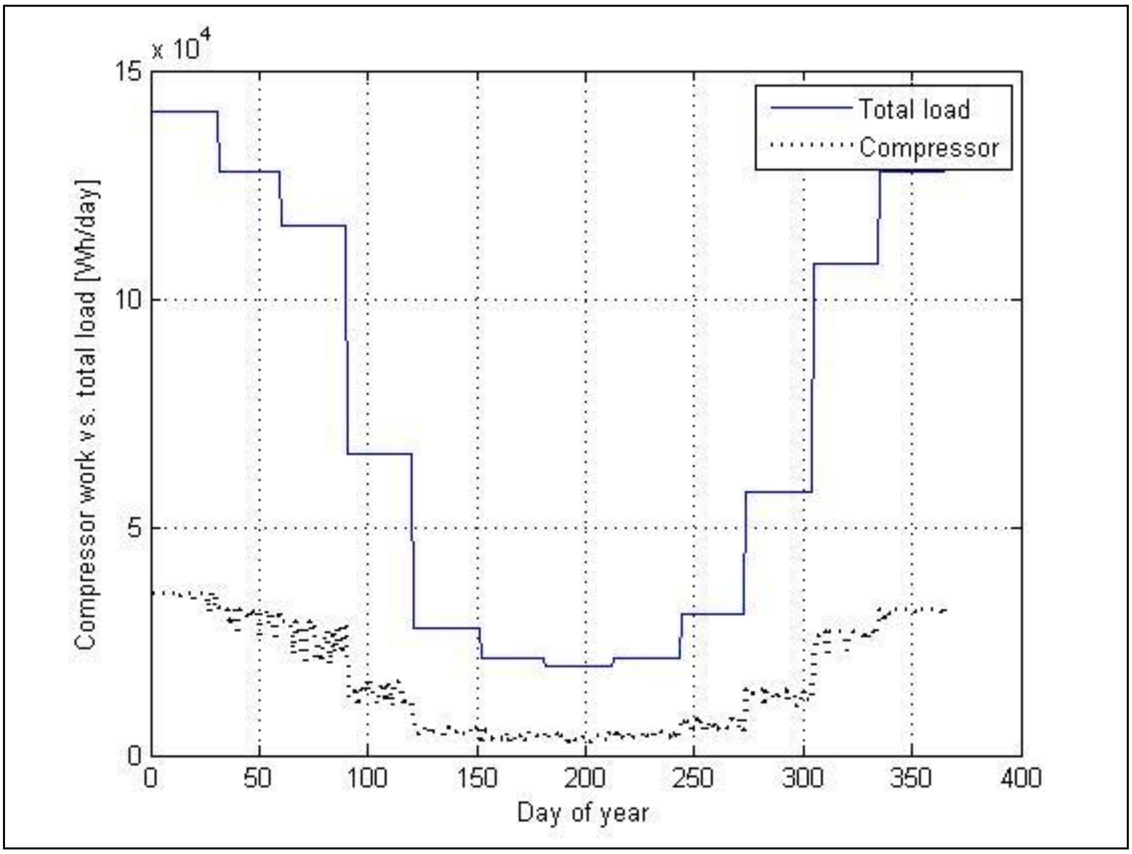

Figure 25*: Compressor power vs. Total load for a year.*

In figure 26 the total load is plotted versus the total amount of solar energy and as seen the solar energy exceeds the load from approximately the beginning of May to the end of August. If all of the solar energy could be utilized in the system it would be possible to reduce the compressor work to zero during these months. This has to include high performance storage and solar energy extraction at high temperatures.

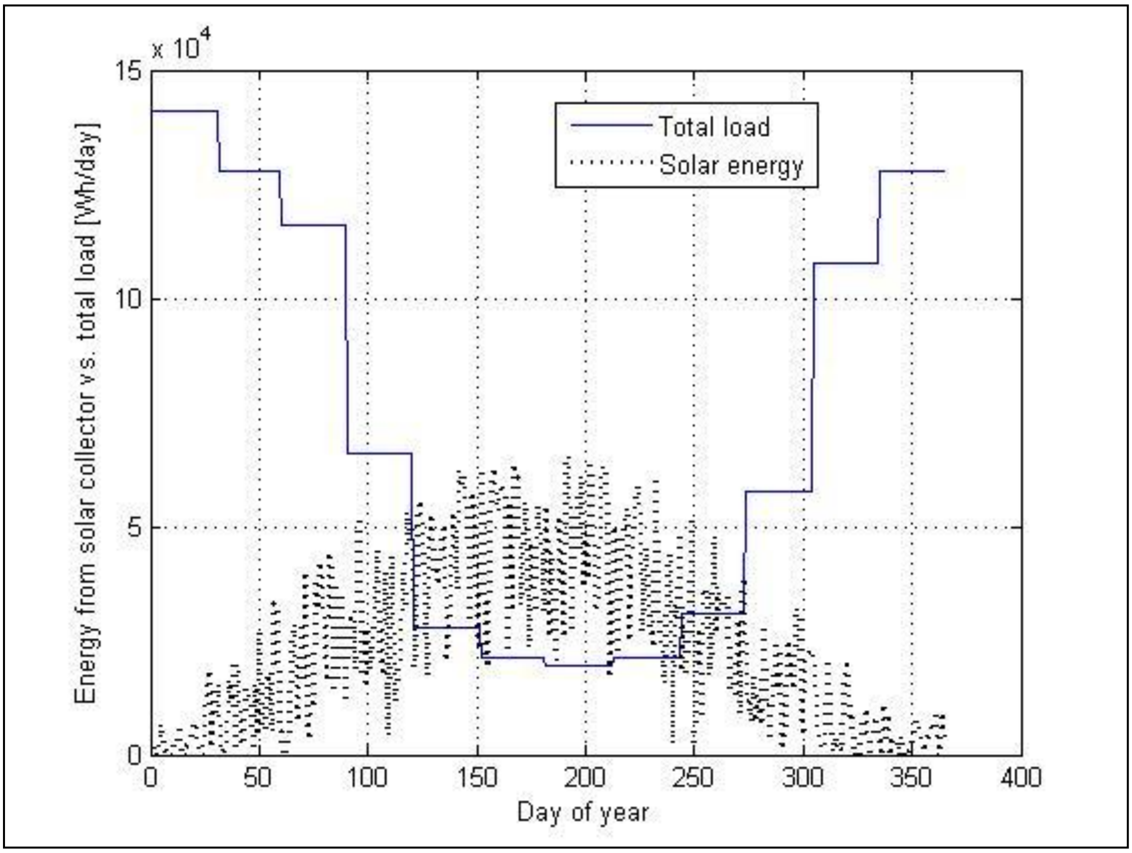

Figure 26: *Energy from solar collector vs. total load*

6. Analysis/Discussion

This section deals with the discussion and analysis of the literature study and the model.

From the literature study we found that a heat pump system with a vertical ground source heat exchanger is the best alternative for a combined system. This is because of the relative stable temperatures from the heat source throughout the year and stable temperature is desired to the evaporator side of the heat pump. In warmer climate than Sweden other sorts of heat pump might be the best option although when it comes to recharge the vertical ground source is the best. The stable temperature of the heat source is also positive when it comes to recharging because the heat losses to the environment is lower in the rock than in for example a horizontal ground heat source.
Another reason for choosing vertical ground source heat pump system is that it is a common solution on the Swedish market and there is a high level of knowledge about these systems.

For the solar collectors it is important to point out that the solar radiation varies from year to year and it is not certain that the system delivers exact the amount of energy as expected. To get as much energy as possible from the solar collector, a sun tracking solar collector would be the optimal. The investment for these solar collectors is large and they are seldom used in small scale thermal solar solutions. The optimal tilt for the collectors is depending on the amount of solar energy, the load and when the system should produce solar energy. During winter time a high tilt is wanted because the sun is low on the sky and this is also the time during the year when solar energy is needed the most for space heating. If a high energy output is wanted during the winter a large tilt is the best option, although the sun radiation is a lot smaller in the winter time compared to summer time and in this way a lower percentage of solar radiation is absorbed through one year.
When combining these systems it is important to consider what kind of solar collector that is best for the system. If, for example a heat pump is installed for a house and later on it appears that the bore hole is under dimensioned. A simple solution can be to just to buy a simple and inexpensive solar collector that can recharge the bore hole during the year. Also depending on the purpose of the system and the desired temperature level different solar collectors can give the optimal solution. For example, vacuum solar collectors can give very high temperatures and are preferred in high temperature applications.

From the study of literature and the model we found that good heat storage is of outmost importance. It is necessary to store the heat given from the collectors and the size of the storage must be optimized for the load and amount of solar energy for the specific system considered. It is possible to have two or more separate tanks in the system, this can increase the efficiency of the system but it has to be optimizing for specific conditions. Stratification in the tanks is an important factor because a low temperature is desired in the bottom of the tank where the temperature to the solar collectors in determined but also because the outtake for heating the house and DHW is made at the top of the tank. If the stratification is good the exergy losses due to mixing will be lower and the solar energy can be utilized in a better way.

Recharge can be used when the energy from the solar collectors exceeds the load or if the temperature of the energy is too low to be used elsewhere. In this case the energy can be heat transferred with the ground source heat carrier at the inlet of the heat pump and the heat carrier will go through the heat pump and transport this energy to the ground source. In this solution the only additional pump power is for transporting the solar collector fluid to the ground source.
How efficient the recharge is are depending on the properties of the rock. When the rock has a high thermal conductivity there will be large heat losses if the heat from the solar collectors has high temperature. The most efficient way to recharge is to use low temperatures from the solar collectors and increase the temperature of the rock to that extent that it has equal or a few degrees higher temperature than the surroundings. If the rock is heated further, this heat will be lost to the surrounding. On the other hand, if the rock has a high thermal conductivity it will have a high natural recharging from the surrounding rock. If the rock has low thermal conductivity it has a low natural recharge and are more likely to need recharge. The recharging will also be more efficient due to decreasing heat losses. It is hard to say if recharging is profitable in a well dimensioned system but in the case of under dimensioned bore holes recharging can be a good way to keep the temperature in the bore holes at a constant level. It might also be used to create under dimensioned system by purpose and therefore get lower costs for drilling.
When considering recharging it is important to be aware of the pump power needed to transport the fluid. If the pump power exceeds the effect from recharging it is unnecessary to do it. To be able to predict if recharging is beneficial for the system a good knowledge about the rock is needed, a well design control system that controls when recharging should takes place is also desired.

The bore hole can also be used as a heat sink. During the summer when the heating load is low and there is a lot of solar radiation problems can arise when there is nowhere to cool the fluid from the solar collectors. One way to avoid stagnation temperatures in the solar collector can be to use the bore hole as a heat sink and cool the fluid by pumping it through the ground loop.

Boosting of the condenser side is in the author’s opinion unnecessary. This is because boosting can only take place in the temperature range between the inlet and the outlet of the cooling fluid on the condenser side. The outgoing temperature in the model is 55°C and the inlet temperature is 45°C. This means that it is only possible to boost when the temperature from the solar collector is higher than 45 and lower than 55.

Boosting on the evaporator side can be beneficial because of the utilization of the low temperature energy from the solar collector. These low temperatures can be extracted all year around and improve the efficiency for the system. In the model was the average temperature available for boosting about 18 °C, assuming a boost at the inlet of the evaporator from 0°C to 15 °C this would make a 30 % decrease of the compressor work.

The complexity of these systems is very high and to optimize them a lot of information and calculations is needed. When designing a combined system it is really important to know right from the start what the system should perform, what it should be used for and when it is going to be used. Information about the rock and solar radiation is necessary to be able to optimize the system.

The model:
Many of the sub models in the model have a lot of simplifications and therefore the results from the model has are valid only with the properties of this specific system. The simulation gives general directions on how much solar energy that can be extracted from the solar collectors. Heat storage was not included in the model and this is because it is a complex component and there was not enough time to develop a good storage model. With that in mind we decided to neglect storage in the model because a model with storage would have given to insecure results.
Further work with the model is to:

- Be able to vary the inlet temperature to the solar collector, this can be done by implementing a storage tank with solar collector outlet at the bottom.
- The mass flow in the solar collector system should not be constant but change depending on the desired temperature out from the collectors.
- The U-value for the solar collector should be divided in a temperature dependent part and a temperature independent part.
- The specific for the heat transfer fluid should vary with the temperature.
- Develop the heat pump model so that the operating conditions for the heat pump can change depending on evaporator temperature.

7. Conclusions

Before considering a combination system, it is of great importance that is known how much the system actually should perform. Because there are a number of system solutions on the market, each providing different amount of heat and the cost of which wary a lot.

For a system that should work ideally, the best alternative for a system of a solar collector in combination with a heat pump contains a vertical ground source heat exchanger. The solar collector type should be decided carefully in order to get the desired temperature from the collectors, depending on these temperatures the solar energy can be used for different purposes and this depends on what the system are suppose to perform.

A combined system would only be beneficial if there is a storage tank installed in the system, where the exceeded heat from the solar collectors could be stored and used during cloudy days and during nighttime. Otherwise the compressor would need to be running in times when the heat pump actually could be switched off. So if there is no storage tank installed all the heat from the solar collectors would not be used and therefore the efficiency of the system would go down significantly and that is a huge disadvantage in an economic point of view.

In order to further increase the efficiency of the system it should be possible to perform a heat exchange between the heat from the solar collectors and the heat source entering the evaporator of the heat pump. This is preferable in order for the system to both boost the evaporating temperature and, when it is possible and if the pump power for the circulation is not too large, recharge the borehole.

It seems that the potential for such a combination system is large but there is a complexity factor that needs to be taken care of. In order for the system to work ideal, the connection between the solar collector and the heat pump should be carefully controlled in order to get the most out of it. If it would be possible to create a computer based system that is measuring the temperature levels in the solar collectors, knows the heating demand of the house and telling the heat pump exactly when to kick in, the potential would be high. If this control system also could decide when the boosting is necessary and when there is a recharging possibility the potential would be even higher. Then the economic savings could be large for a family house in Sweden.

References

Literature:

Brown, R.N. 2005. *Compressors, Selection and Sizing.* ISBN: 0-7506-7545-4

Department of Energy Technology, KTH, 2009. *Refrigeration Engineering*. Division of Applied Thermodynamics and Refrigeration, Royal Institute of Technology, KTH, Stockholm.

Duff, W.S. 1996. *Advanced solar domestic hot water systems*. International Energy Agency, Task 14 final report

Morrison G.L 2001. *Solar Collectors.* Chapter 4 in ISES 2001. *Solar energy: The state of the art.* ISBN: 1-902-916*23-9.*

Williams, R.J 1983. *Design and installation of solar heating and solar water systems.* ISBN: 0-250-40593-8.

Manuals

Caneta Research inc. 1995. *Commercial/Institutional Ground-Source heat pumps.* Engineering Manual. ISBN: 1-883413-21-4

Reports

ISES 2001. *Solar energy: The state of the art.* ISBN: 1-902-916*23-9.*

Energimyndigheten 2009b. *Mätning av kall- och varmvattenanvändning i 44 hushåll*. ID-nr: ER 2009:26

Energimyndigheten 2009c. *Solklart Solvärme*. ID-nr: ET 2009:17

Kjellsson, Elisabeth 2004. *Solvärme i bostäder med analys av kombinationen solfångare och bergvärmepump.* Report TVBH-3047 Lund 2004. Building Physics LTH.

Kjellsson, Elisabeth 2009, *Solar Collectors Combined with Ground-Source Heat Pumps in Dwellings Analyses of System Performance*. Report TVBH-1018 Lund 2009. Building Physics LTH

Internet:

Energysavers 2009. *Heat-Transfer Fluids for Solar Water Heating Systems*
http://www.energysavers.gov/your_home/water_heating/index.cfm/mytopic=12940 (2010-02-15, 13:00)

Energimyndigheten 2009a. *Cirkulationspumpar*
http://www.energimyndigheten.se/sv/Hushall/Tester/Testresultat/Cirkulationspumpar/ (2010-02-17, 15:00)

Eviheat 2010
http://www.eviheat.se/ (2010-03-01,12:00)

Grup Romet, 2010
http://www.grupromet.ro/Engleza/Panouri%20Solare/Sisteme%20combinate/Sisteme%20combinate.html (2010-02-19, 16:00)

International Energy Agency 2009. *About heat pumps*
http://www.heatpumpcentre.org/ (2010-02-17, 16:00)

Solenergiteknik, 2010
http://www.solenergiteknik.se/ (2010-03-18, 16:00)

Svenska Värmepumpsföreningen 2010a. *Värmepumpar-Berg*
http://www.svepinfo.se/varmepumpar/berg/ (2010-02-18, 15:00)

Svenska Värmepumpföreningen, 2010b. *Värmepumpar-Vatten*
http://www.svepinfo.se/varmepumpar/vatten/#Grundvatten (2010-02-18, 16:00)

Seminars:

KTH, Lecture, 2009,*Liquid Heating/Cooling Distribution Systems Part 1*
Joachim Claesson,(Prepared by Hans Jonsson)