OPTIMIZATION OF GROUNDSTORAGE HEAT PUMP SYSTEMS FOR SPACE CONDITIONING OF BUILDINGS

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TARGETS

• Targets for the research:
  - A calculation tool for complete system optimization
  - Identification of key efficiency factors (for optimization)
  - Identification of energy efficient system designs for a few (to be specified) applications

• Target levels for market penetration (conditions to be decided):
  - Space conditioning of office buildings with < 20 kWhel/m²/year and < 5 kWhheat/m²/year
  - Space conditioning of apartment buildings with < 40 kWhel/m²/year
WORK PLAN (1)

• Previous work
  - Data bases (Fridoc, compendex …), personal contacts (project group)

• State-of-the-art review (analysis and synthesis)
  - Practical experience, measured results, estimate of future potential
  - Modelling tools, identification of the need for development and measurements for validation; suitable ways to proceed

• Overview and classification of current system designs
  - Inventory of review, classification and selection of systems for further investigations (modelling and measurement)

• Field tests
  - Inventory of existing and need for new tests, planning and start of measuring program, analysis and synthesis
WORK PLAN (2)

• Theoretical analysis
  - Development of component models
    (e.g. collector, heat pump, heat exchangers, pumps, fans etc.)
  - Comparison of calculations and measurements
  - Simulation of selected system solutions with variation of design values
    (sensitivity analysis: how critical is the system design for the technical
    and economic result?)
  - Identification of strengths and weaknesses of different designs

• Reporting, presentation and dissemination of results
  - Intermediate work reports for each sub-task
  - International conferences: IEA, IIR, ASHRAE
  - National conferences: Effsys Annual Meeting, Energitinget
  - Scientific articles: Journal of the IIR, Energy and buildings, ASHRAE
  - Popular articles: ScanRef, Energi och Miljö, VVS-Forum
  - LICENTIATE THESIS
# TIME PLAN

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RELATED PROJECTS (1)

- **Optimized operation of heat pump systems (VSD)**
  - Fredrik Karlsson, SP/CTH Building Services Engineering

- **Efficiency of building related pump and fan operation (VSD)**
  - Caroline Markusson, CTH Building Services Engineering
  - Johan Åström, CTH Electric Power Engineering

- **Energy efficient cooling coils and brine systems (laminar flow)**
  - Caroline Haglund Stignor, SP/LTH/CTH Building Services Eng.

- **Control-on-demand ventilation – Intelligent supply-air devices (VAV)**
  - Mari-Liis Maripuu, CTH Building Services Engineering

- **Environmental assessment of air-conditioning systems**
  - Katarina Heikkilä, CTH Building Services Engineering

- **Free-cooling (general, air, sorptive, desiccant, cooling towers)**
  - Torbjörn Lindholm, CTH Building Services Engineering
RELATED PROJECTS (2)

- **Recharging of boreholes using ventilation air**
  - Per Fahlén, CTH Building Services Engineering

- **GSHP with integrated liquid-loop HRV**
  - Per Fahlén, CTH Building Services Engineering

- **Dynamic thermal networks**
  - Johan Claesson, CTH Building Physics

- **Control-on-demand heating, cooling and ventilation**
  - Per Fahlén, CTH Building Services Engineering

- **Energy efficient shopping centres**
  - Sofia Stensson, SP/CTH Building Services Engineering

- **Supermarkets**
  - Ulla Lindberg, SP/CTH Building Services Engineering

- **Integrated, intelligent control-on-demand**
  - Mohsen Soleimani Mohseni, CTH Building Services Engineering
APPLICATION
Ground-source with heating cooling and storage (free cooling)

- Heat source
- Heat sink
- Heat storage
BOREHOLE SYSTEM
– Important design factors

- **Building load profile**
  - Relation between heating and cooling

- **Borehole load**
  - Specific heat uptake in winter [kWh/m/year]
  - Specific heat input in summer [kWh/m/year]

- **Bore hole system geometry**
  - Number of holes and their depth
  - Borehole pitch
  - Ratio of length/width (rectangular or linear configuration)

\[
\text{Antal hål } N = m \times n
\]

Linjär geometri, antal hål \( N = m \times n \)
**INFLUENCE OF LOAD PROFILE ON TEMPERATURE**

Rectangular systems with storage require more careful sizing than linear.
EXAMPLE OF TEMPERATURE VARIATION

Three different borehole loads with the same ratio between heat uptake and heat rejection in the borehole (rectangular geometry)
HEATING – COOLING - VENTILATION

• **Recharging:**
  - exhaust-air coil **ON**
  - supply-air coil **OFF**

• **Cooling**
  - exhaust-air coil **OFF**
  - supply-air coil **ON**

• **Heat recovery**
  - exhaust-air coil **ON**
  - supply-air coil **ON**
DEMAND FOR HEATING AND COOLING

• **Building level**
  Balance between surplus and deficit of heat within the building (own supply unit)

• **Block level**
  Balance between adjacent buildings (block centrals)

• **Community level**
  Balance between groups of buildings (district heating, district cooling)
SUMMARY

• Non-residential buildings with large heat surplus ⇒ demand for cooling

• Heat pumps can handle heating and cooling simultaneously (natural relation approx. 3:2)

• Borehole system provides stable temperature and possibility of storage (free cooling)

• GSHP systems may achieve very high efficiency but require careful analysis of real heating and cooling demand

• Design and sizing of heat pump, heating and cooling system (temperature!) and borehole system important for efficiency and economy
KEY FACTORS FOR EFFICIENCY

- **Temperature level** *(COP changes by 2-3 % per °C!)*
- **Heat exchangers** *(temperature difference, pressure drop, material, pressure level)*
- **Compressor** *(type, sizing, capacity control)*
- **Pumps and fans** *(efficiency, operating time, capacity control)*
- **Refrigerant** *(efficiency, long-term acceptability, cost, safety)*
- **Brine** *(efficiency, corrosivity, stability, cost, safety, environment)*

\[
\text{COP}_{vpa,medel} = \frac{\bar{Q}_{vpa} + \frac{R_{p1}}{R_{vpa}} \cdot \dot{W}_{e,p1}}{\dot{W}_{e,m} + \frac{R_{p1}}{R_{vpa}} \cdot \dot{W}_{e,p1} + \frac{R_{p2}}{R_{vpa}} \cdot \dot{W}_{e,p2}}
\]
TEMPERATURE LEVEL AND PARASITIC POWERS

• **Influence on COP**
  - Drive units and temperature levels:

  \[
  \frac{\Delta COP_1}{COP_1} = -\frac{\Delta \dot{W}_{e,vp}}{\dot{W}_{e,vp}} = \left[ \frac{\Delta T_{kb}}{T_1 - T_2} - \frac{T_2}{T_1} \cdot \frac{\Delta T_{vb}}{T_1 - T_2} - \frac{\dot{W}_{e,p}}{\dot{W}_{e,vp}} \right]
  \]

• **Example with recharging**
  - \(\Delta T_{kb} = +4\) K should give \(\Delta COP/COP \approx +10\) %
  - But \(\Delta T_{vb} > +4\) K, \(\Delta w_{e,p}/w_{e,vp} = -9-10-24 \approx -43\) %
  - Total reduction by 40 - 60 %!
EXAMPLE: FAN COIL

- **Control**
  - Coil fan on
  - Heat pump on-off

- **Temperature**
  - On-temperature > mean temperature
  - Example: $t_{out} = 4.4 \, ^\circ C$

$$\bar{t}_{w1, on} - \bar{t}_{w1, cycle} = 5.7 \, ^\circ C$$

$$\Delta COP \approx -10 \text{ till } -15 \%$$
PARASITIC POWERS

- Heat gain ("cooling loss")
- Heat transfer resistance
- Heat transfer pressure drop
- Heat transport pressure drop

\[ \dot{W}_{ep} = \frac{\dot{V} \cdot \Delta p}{\eta_p} \]
TEMPERATURE DISTRIBUTION

- Two transfer differences
- One transport difference
CONTROL OF AIR FLOW

- **Thermal comfort: E.g.** $t_{room}$

  $$t_{room} = t_{sa} + \frac{\dot{Q}_{int}}{K_{tot}}$$

  $$K_t = -\frac{\dot{Q}_{int}}{C_a \cdot (U \cdot A / C_a + \dot{V}_{nom})^2}$$

  $$\Delta t_{room} = \Delta t_{sa} + K_t \cdot \Delta V_{vent}$$

- **Air Quality: E.g.** $CO_2$

  $$c_{room} = c_{sa} + \frac{\dot{V}_{CO2}}{\dot{V}_{vent}}$$

  $$K_{CO2} = -\frac{\dot{V}_{CO2}}{\dot{V}_{nom}^2}$$

  $$\Delta c_{room} = \Delta c_{sa} + K_{CO2} \cdot \Delta V_{vent}$$
SUPPLY AIR COOLING CAPACITY

- Depends directly on
  - air flow rate
  - supply-air temperature
  - room temperature

\[ \dot{Q}_{sa} = K_{sa} \cdot (t_{sa} - t_{room}) \]

\[ \frac{\Delta \dot{Q}_{sa}}{\dot{Q}_{sa}} = \frac{\Delta V_{sa}}{V_{sa}} + \frac{\Delta t_{sa}}{(t_{sa} - t_{room})} - \frac{\Delta t_{room}}{(t_{sa} - t_{room})} \]
LOWER TEMPERATURE OR MORE FLOW?

- Lower temperature → higher compressor drive power
- Higher flow rate → higher fan power

\[
\Delta \dot{W}_e = \Delta \dot{W}_{hp} + \Delta \dot{W}_{e,f} = \\
= (k_{hp} \cdot \dot{W}_{hp} + k_f \cdot \dot{W}_{e,f}) \cdot \Delta t_{sa}
\]

\[
\frac{\Delta \dot{W}_e}{\Delta t_{sa}} = k_{hp} \cdot \dot{W}_{hp} \cdot \left(1 + \frac{k_f}{k_{hp}} \cdot \frac{\dot{W}_{e,f}}{W_{hp}}\right)
\]

\[
\frac{\Delta \dot{W}_e}{\Delta t_{sa}} = 0.03 \cdot \dot{W}_{hp} \cdot \left(1 - 25 \cdot \frac{\dot{W}_{e,f}}{W_{hp}}\right)
\]
EXTERNAL OR INTERNAL ROOM COOLING

- Requirement on SFP to make TC-controlled air flow more electricity efficient than a chiller (n = 1.5 to 2):

\[
SFP_{nom} < \left( \frac{\dot{q}_V}{COP_c} \right) \cdot \left( \frac{\dot{Q}_{sa}}{\dot{Q}_{nom}} - 1 \right) \cdot \left( \frac{\dot{Q}_{sa}}{\dot{Q}_{nom}} \right)^{n+1} - 1
\]

Volumetric fan power:

\[
SFP = \frac{\dot{W}_{e,f}}{\dot{V}_a} \quad [\text{kW/m}^3/\text{s}]
\]

Volumetric cooling capacity:

\[
\dot{q}_V = \frac{\dot{Q}_{sa, nom}}{\dot{V}_{sa, nom}} \quad [\text{kW/m}^3/\text{s}]
\]

\[
(\dot{q}_V \approx 1.2 \cdot (t_{sa} - t_{room}))
\]

Relative cooling capacity:

\[
\frac{\dot{Q}_c}{\dot{Q}_{nom}} \quad [\text{kW/kW}]
\]

\[0,0\quad 0,2\quad 0,4\quad 0,6\quad 0,8\quad 1,0\quad 1,2\quad 1,4\]

\[1,0\quad 1,5\quad 2,0\quad 2,5\quad 3,0\]

Relative cooling capacity [kW\text{/kW}_{nom}], \quad \text{SFP}_{nom} [\text{kW/m}^3/\text{s}]

COP = 2, Dt = 4 K
COP = 4, Dt = 4 K
COP = 2, Dt = 8 K
COP = 4, Dt = 8 K
FAN POWER

- **Reduce SFP**
  - Raise fan efficiency
  - Reduce distribution pressure drop
  - Reduce AHU pressure drop
  (FI, HR, AH, AC)

**Typical pressure drop Pa**

\[
\Delta p_{01} = -150 \quad \Delta p_{78} = -150 \\
\Delta p_{12} = -100 \quad \Delta p_{89} = -100 \\
\Delta p_{23} = -30 \quad \Delta p_{90} = 500 \\
\Delta p_{34} = -50 \quad \text{plus ducts, dampers, terminals units} \\
\Delta p_{45} = 600
\]

\[
SFP = \frac{\dot{W}_{e,f}}{\dot{V}_a} = \frac{\Delta p \cdot 10^{-3}}{\eta_f}
\]

\[
SFP = SFP_{nom} \cdot \left( \frac{\dot{V}_a}{\dot{V}_{a,\text{nom}}} \right)^n
\]
CONTROL OF AIR CONDITIONING

- Air flow based on AQ-requirement (e.g. $c_{\text{CO}_2} < 1000$ ppm)
- Room temperature based on TC-requirement (e.g. 21 °C)
- Keep $t_{\text{room}}$ at the comfort minimum (e.g. 21 °C)
- Reduce heat recovery until $t_S = t_{S,\text{min}}$ (e.g. 17 °C)
- Use deadband; $t_{\text{room,min}} < t_{\text{room}} < t_{\text{room,max}}$ (e.g. 21-25 °C)
- Raise air flow based on max. of AQ- or TC-demand
- Use night cooling but watch the fan energy
- Start chiller when the marginal increase of fan power is larger than the chiller drive power
OFFICE ECONOMY?

- Staff cost: 100 000 SEK/m²/year
- Rent: 2 000 SEK/m²/year
- Capital cost of HVAC: 200 - 1 000 SEK/m²/year
- Energy cost: 100 - 500 SEK/m²/year
GROUND STORAGE – PHASE II

- Connecting the developed models
- Optimized design and size
- Optimized control
  - Short-term: Feedback feed-forward
  - Medium-term: Predictive (day-night cooling-heating)
  - Long-term: Predictive (summer-winter balancing)
- Case-studies