Efficient Cooling in Hot Climates

A Thermodynamic Systems Approach to Quadrupling Average Efficiency

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Project Objectives were to ***identify*** complementary cooling technologies and associated controls,
Develop ***component and subsystem models*** including optimal control algorithms,
Prepare ***MatLab scripts for simulating*** and comparing baseline and several advanced HVAC configurations,
Estimate savings over a representative ***grid of building, climate and HVAC configuration***:
·Basic-, Medium- and High-Performance building (efficient envelope, lighting, and user-equipment)
·Five climates from hot-and-humid (Houston) to northern-temperate (Chicago), and for
·All combinations of the identified efficient cooling technologies
Extrapolate simulated energy use and savings to estimate national ***energy savings potential*** based on application of the best-performing combination of technologies and controls to all new construction projects.

System Integration: Chiller · Ventilation · Control

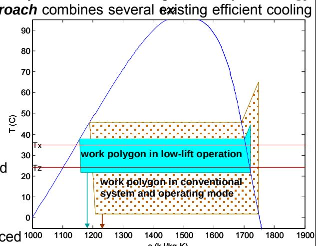
Envelope Measures: Efficient cooling in any climate means first minimizing sensible and latent cooling loads… ***sensible*** loads by insulation, by reflective wall and roof surfaces, by control of solar gains to just what is needed for daylighting, and by use of efficient office (or other user) equipment and efficient lighting…***latent*** loads by preventing infiltration, by conditioning only as much ventilation air as is needed, and by using exhaust enthalpy recovery. In short, investment in an efficient cooling system should not outpace investment in the building envelope, lighting, and other efficiency measures--cost-effective performance is achieved by balanced investment.

HVAC Measures: **Even in hot climates,** there is ample room for improving mechanical cooling and distribution performance by approaching the building and equipment as an integrated system. The main targets, ***transport energy*** and ***compressor energy***, must both be addressed. The ***systems approach*** combines several existing efficient cooling and distribution ***technologies that are particularly complementary:***

- **Peak-Shifting by precooling the building mass at night**
- **Radiant Cooling Panels (RCP) for sensible cooling load**
- **Dedicated Outside Air Supply (DOAS) and conditioning**
- **Low-Lift vapor compression cooling equipment.**

What is Low-Lift? Vapor-compression and absorption devices move heat against a temperature difference…which can be reduced by:
- Using RCP & DOAS to ***increase evaporator temperature***
- Cooling at night to ***decrease condensing*** temperature and average load
- Using variable speed fans and pumps to ***reduce transport energy***
- Using a variable speed compressor to ***reduce flow losses*** and…
- Achieve ***closer condenser and evaporator approach*** temperatures

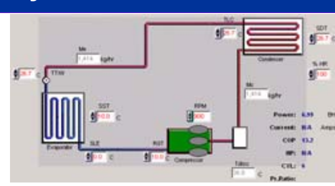
Compressor work (blue area) for given cooling effect is thus greatly reduced



Chiller Component and System Models

Optimal Control of Chiller-Distribution

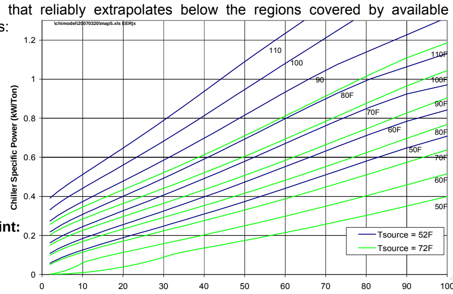
Chiller and distribution components to be modeled include:
- **Compressor**
- **Evaporator**
- **Condenser**
- **Transport (fan and pump) power**
- **Radiant Cooling Panel**
- **CV and VAV Fan-Coils**



Compressor performance must be accurately characterized at very low pressure ratios (1:1 to 1:1.5) and low displacement rates (to 25% or less of rated speed). This is accomplished by fitting a **semi-physical model** that reliably extrapolates below the regions covered by available performance data (above right). The model reflects:
- Polytropic Compression
- Suction gas heating decreases with speed
- Flow and Friction Losses increase with speed
- Volumetric Efficiency = f (ρ, Pd/Ps, rpm)

Compressor Performance Map (right) is based on **Effectiveness-NTU models** of
- Evaporator (assume zero superheat)
- Desuperheater (Hiller effectiveness model)
- Condenser (assume zero subcooling)

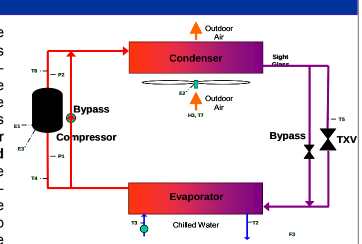
Chiller Solver: **given load and conditions** must determine the **maximum-COP operating point:**
- Compressor, fan and pump **shaft speeds**
- Suction and Discharge **pressures**
- Refrigerant mass **flow rate**
- Condenser fraction for **desuperheating**



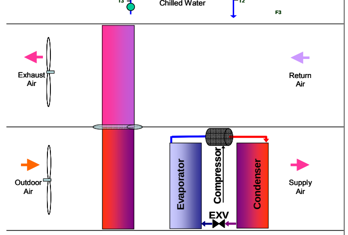
A bicubic response surface of COP is fit to data computed on a grid of part-load fraction and outdoor temperature. The bicubic form, which evaluates very fast, is critical to the peak shifting control algorithm within which chiller performance is evaluated millions of times. The response surfaces for load-side temperatures of 22°C (radiant panel case) and 11°C (fan coil case) are shown above. Note the inflections at low part-load fraction for 22°C load-side temperature on the 10°C and 15.6°C outdoor temperature lines; the compressor is bypassed and refrigerant-side economizer mode is invoked below these inflection points.

Special Subsystem and Control Models

Refrigerant-Side Economizer: All-air systems use cool outside air for "free cooling" when conditions permit; this is not possible with RCP systems. Connecting to a maintenance-intensive cooling tower is costly. An alternative--which may make RCP-based cooling more attractive to building owners--is to use air-cooled chillers with refrigerant-side economizers. This approach is illustrated to the right. Our **model solves condenser fan** and **chilled water pump speeds** that satisfy a **given load** with **minimum transport energy**. Separate bicubic response surface is fit to the performance data computed on a grid of part-load fraction and outdoor temperature. When outdoor temperature is above room temperature the vapor-compression chiller map applies; when outdoor temperature is below room temperature both are evaluated and the less power intensive result is applied.



DOAS Conditioning: A dedicated outdoor air system circulates about 15% of the air required by an all-air system—just enough to remove and maintain acceptably low air contaminant concentrations. 100% of this reduced air flow is outside air. Enthalpy recovery equipment has been shown capable, in tight buildings, of saving 50-80% of the heating and cooling energy needed to condition outside air. In cooling mode the DOAS must satisfy any remaining latent cooling (dehumidification) load. A variable-speed vapor-compression machine—as shown at right downstream of the supply-air-conditioning enthalpy wheel—can be used to convert the latent load to sensible load by cooling the supply air to the required dew point and then reheating the air by rejecting all of the heat produced in the dehumidification process. Thermally regenerated desiccant dehumidification will also be evaluated.



Thermal Energy Storage (TES): Several technical options (right) exist for a TES sub-system to enable load-shifting schemes such as night pre-cooling. For the scoping task at hand, an idealized room-temperature phase-change material applied to all room surfaces, is postulated. This idealized model has the intrinsic TES properties of low transport energy and low losses, but is limited to one-day storage.

Common TES Options:
Water in stratified tank
Ice (not suitable for low-lift)
Building mass (intrinsic TES)
PCM floor, ceiling, walls
PCM in tank (discrete TES)

Peak-Shifting Controls: The peak-shifting algorithm for an idealized TES determines the 24-hourly chiller loadings that satisfy the total daily load (sensible plus latent plus DOAS compressor heat) with minimum input energy, J (right). This type of advanced control requires a 24-hour-ahead cooling load forecast and an embedded chiller-distribution system performance model. Building-specific transient thermal response models, one of the objects of ongoing research, will be incorporated into next-generation peak-shifting algorithms to prevent unnecessary overcooling or undercooling of building mass.

Minimize $J = \sum_{t=1}^{24} \frac{Q(t)}{\eta_{chiller}(t)}$

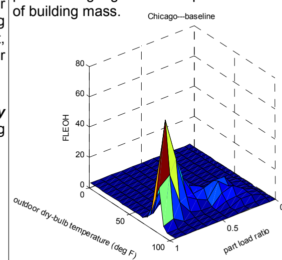
subject to the daily load requirement:

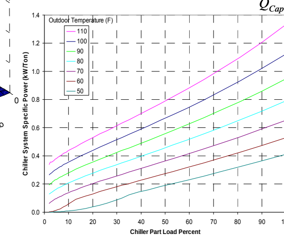
$$\sum_{t=1}^{24} Q_{Load}(t) = \sum_{t=1}^{24} Q(t)$$

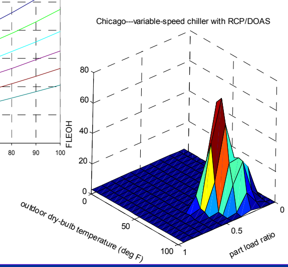
and to the capacity constraints:

$$0 \le Q(t) \le Q_{Cap}(T_X(t), T_Z(t)) \quad t = 1:24$$

where
$\eta_{chiller}$ = chiller efficiency (kW/kBtuh or kW/ton)
T_X = outdoor dry- or wet-bulb temperature,
T_Z = zone temperature,
Q = evaporator heat rate (positive for cooling),
Q_{Load} = building cooling load, and
Q_{Cap} = chiller cooling capacity.





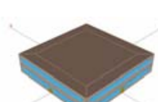


System (chiller, condenser, radiant slab) performance map (center) informs the cost function (top right) to optimally shift load (initial distribution shown at left) from day to night (lower outdoor temperature→higher efficiency) and to shave peaks and fill valleys (transformed distribution shown at right) when thermal energy storage is used.

Evaluation Schema and Results

Case/Building/Climate Evaluation Grid: The base HVAC system is modeled as a variable-air-volume (VAV) no-reheat system fed by a central chiller to condition the occupied spaces. Hourly building cooling loads from DOE-2.2 simulation are input to a separate simulation model of the chiller and distribution system. The previously described system performance maps ensure an apples-to-apples comparison by using identical chiller components for the baseline, partial, and full TSC configurations. In addition to the basic HVAC system (Case 1, right), seven alternative HVAC systems (six partial and the full TSC configuration) were analyzed. The evaluation grid also covers three building energy performance levels (below) and five climates.

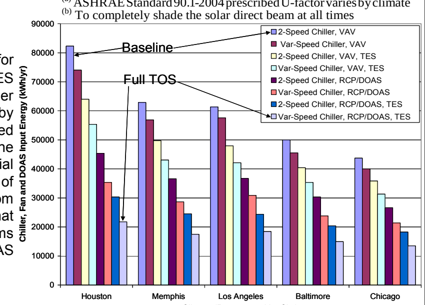
Technology Subset Combinations (TSC)
Base Case: VAV with 2-Speed Chiller
Technology Elements Applied One at a Time
• VAV with Variable-Speed Chiller
• Radiant Cooling and 2-Speed Chiller
• VAV, 2-speed Chiller, and Thermal Storage
Two-Element Combinations
• Radiant Cooling and Variable-Speed Chiller
• Variable-Speed Chiller and Thermal Storage
• Thermal Storage and Radiant Cooling
All Three Elements
• Radiant Cooling, Variable-Spd Chiller, TES



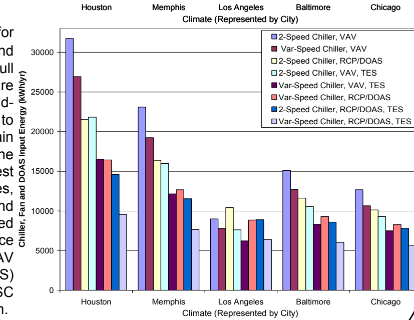
	Non-HVAC Energy Performance Levels		
Component	**Standard**	**Mid-Performance**	**High-Performance**
Wall-Roof U-Factor[a]	90.1-2004	2/3 of 90.1-2004	4/9 of 90.1-2004
Window U-Factor and SHGC[a]	90.1-2004	2/3 of 90.1-2004	4/9 of 90.1-2004
Window-to-Wall-Ratio	40%	20%	20% + Shading[b]
Light and Plug Loads (W/m^2)	0.121+0.059	0.081+0.039	0.054+0.020
Fan Power (W/(m^3/s))	0.471	0.314	0.210

[a] ASHRAE Standard 90.1-2004 prescribed U-factor varies by climate
[b] To completely shade the solar direct beam at all times

Standard Building: Annual energy savings for the RCP system with variable-speed chiller and TES compared to the VAV system with two-speed chiller range from 74% for a hot climate (represented by Houston) to 70% for milder cooling climates (represented by Los Angeles and Chicago). Note, moreover, that the savings for the full TSC compared to the next best partial TSC–in which the chiller operates in 2-speed in-stead of full variable-speed mode–are significant ranging from 27% (Houston) to over 32% (Los Angeles). Note also that RCP/DOAS performs the best of partial TSC systems involving one element, and TES with RCP/DOAS performs the best of systems involving two elements.



High-Performance Building: Savings for the full TSC are 71% for Houston, 57% for Chicago, and 34.5% for Los Angeles. The percent savings for the full TSC compared to the next best partial TSC are significantly better than those of the standard and mid-performance buildings, ranging from 30% (Chicago) to 35% (Houston). The RCP/DOAS configuration again performs best of the partial TSC systems involving one element, and TES with RCP/DOAS still performs the best of systems involving two elements. For Los Angeles, however, VAV is retained in the best-performing one- and two-element configurations. This reflects the reduced specific-fan-power design of the high-performance building, which benefits the air-side economizer (VAV cases), while refrigerant-side economizer (RCP/DOAS) performance is unchanged. Thus the best partial TSC involving one element in Los Angeles is the TES system.



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