DAIKIN

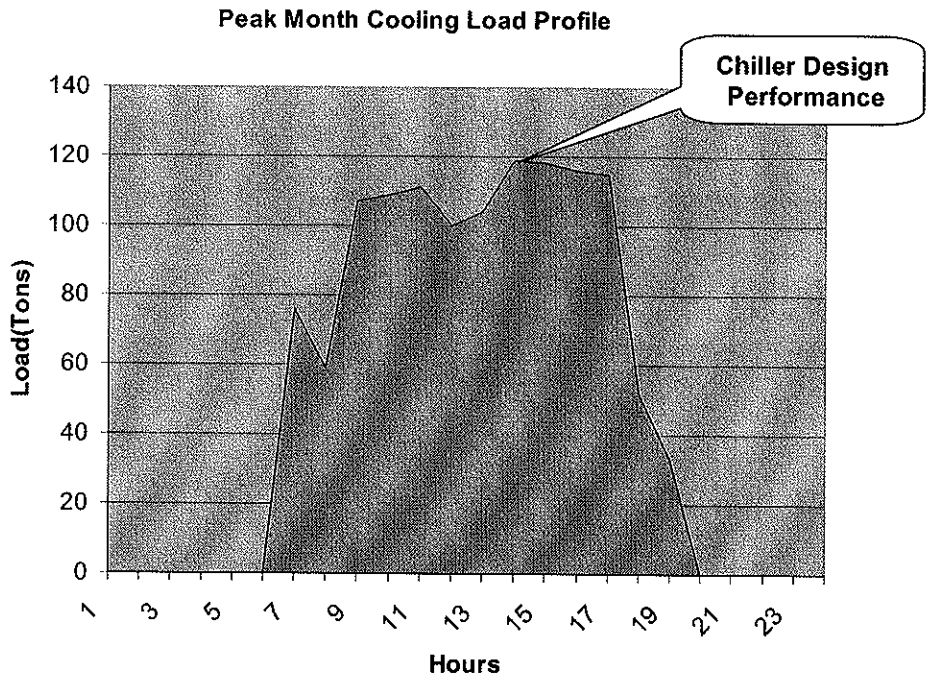
McQuay International

HOW TO OPTIMIZE CHILLED WATER SYSTEMS

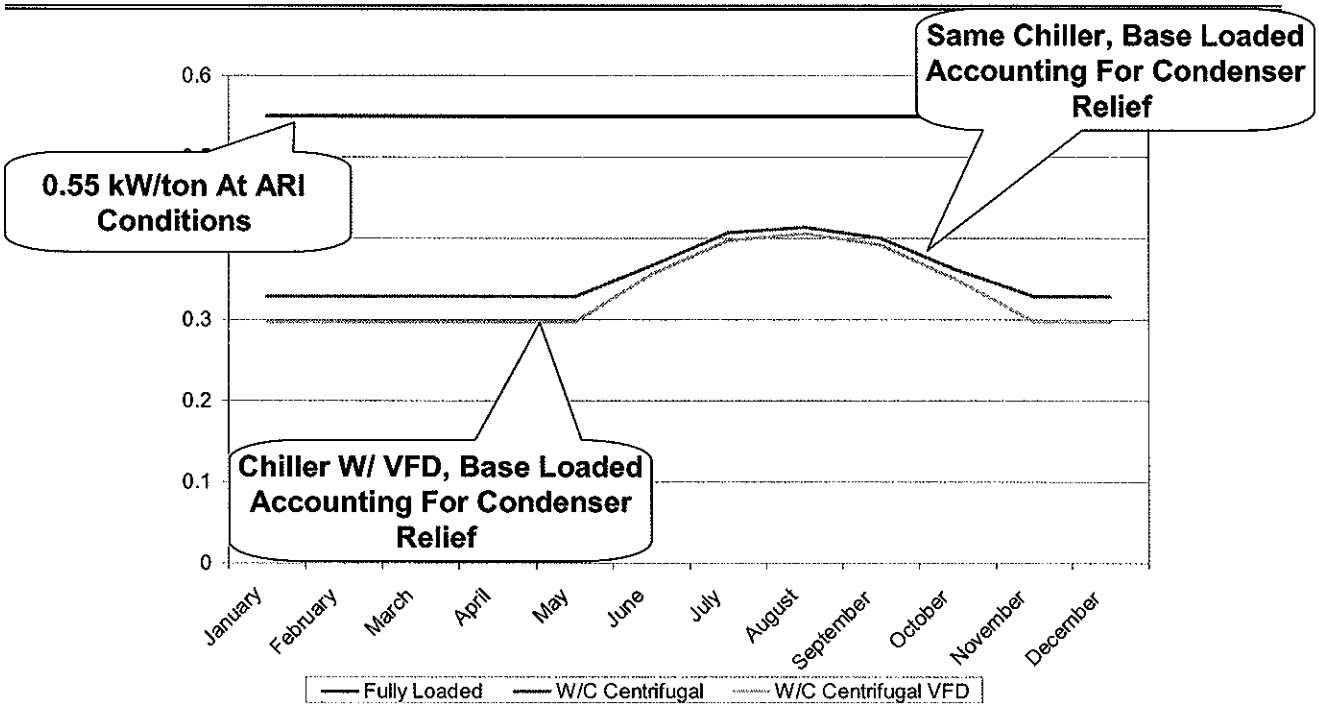
How To Optimize Chilled Water Systems

Julian de Bullet
Director of Industry Relations
McQuay International
703-395-5054

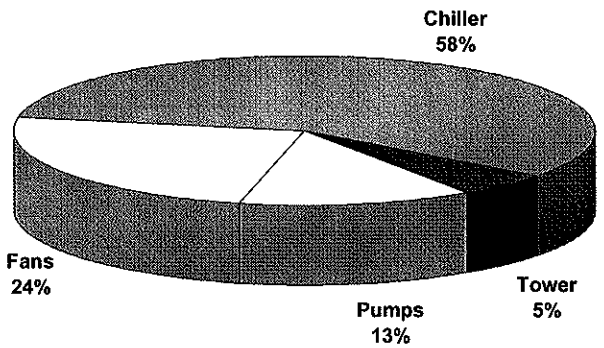
Full Load Vs. Annual Load



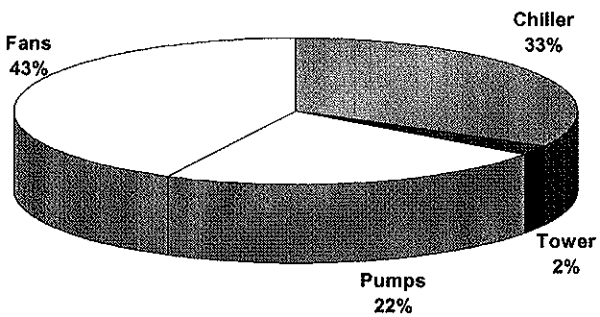
Full Load Vs. Annual Load



Full Load Vs. Annual Load



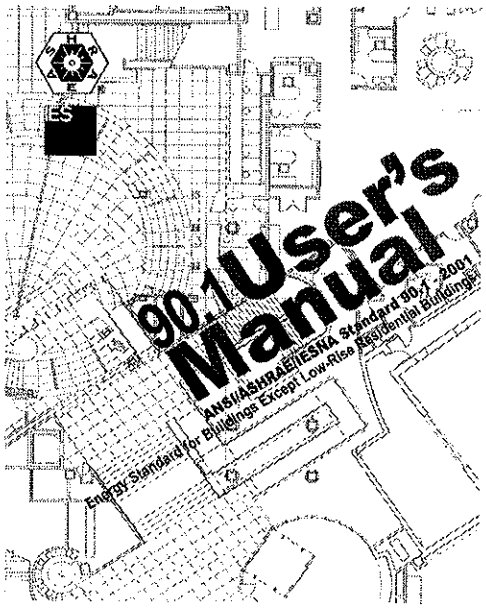
Design Performance



Annual Energy Usage

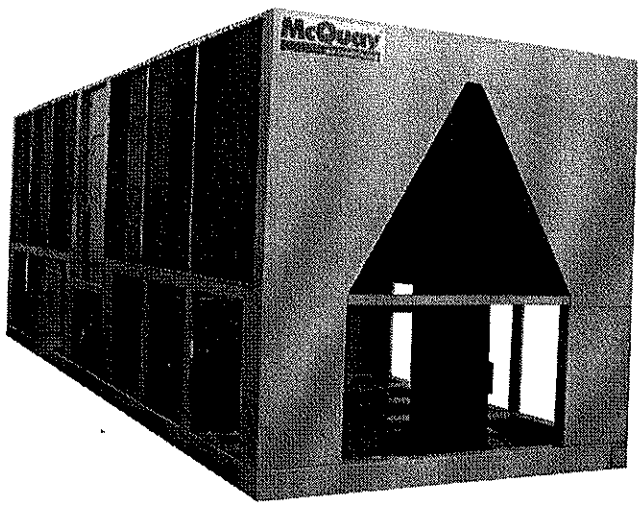
ASHRAE Standard 90.1





Air Cooled Vs. Water Cooled

- Avoid Tower, Pump And Piping
- No Water Cost For Tower
- Higher kW Than Water Cooled Chillers
- Compressor Work Tracks Drybulb Not Wetbulb
  - Very Good NPLVs
- Excellent Choice For Schools With Reduced Summer Hours



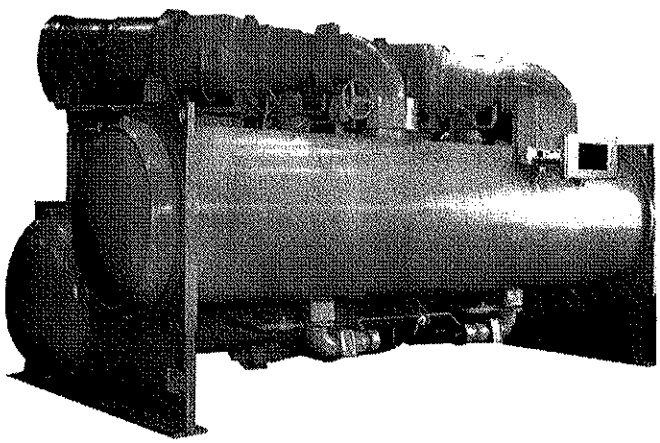
Air Cooled Vs. Water Cooled

Summary

- Water Cooled Is More Energy Efficient
- May Not Be More Cost Effective
  - Water And Maintenance Costs
- May Not Have Acceptable Life Cycle Analysis

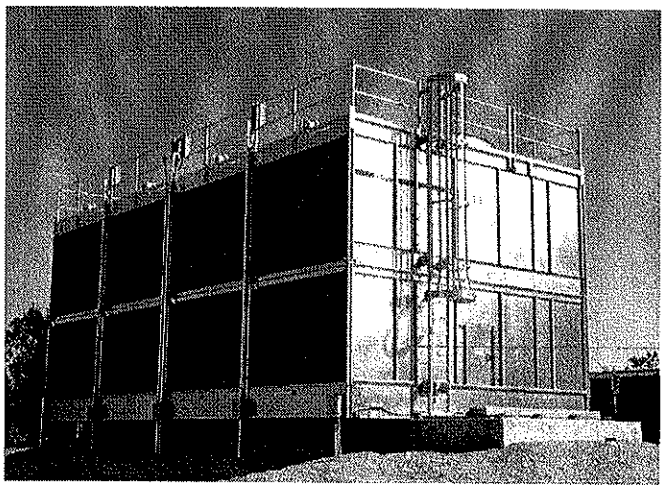
Chiller Basics

- Air, Water Or Evaporatively Cooled
- Reciprocating, Scroll, Screw Or Centrifugal Compressors
- DX or Flooded Evaporators

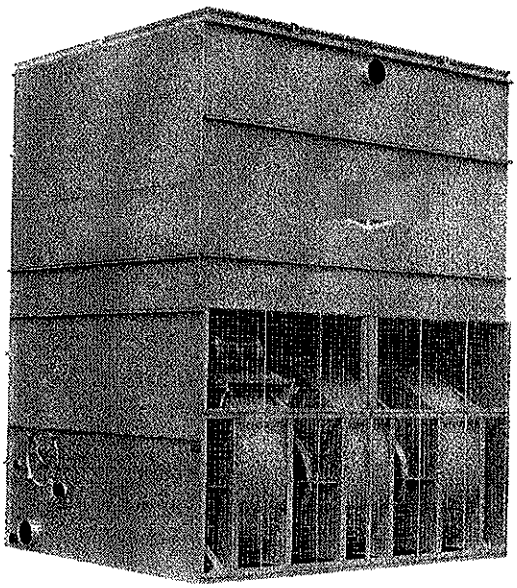


Cooling Tower Basics

Induced Draft Tower

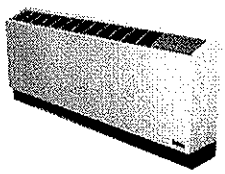


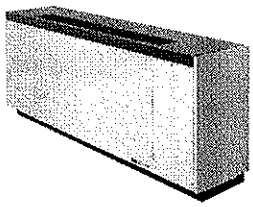
Forced Draft Tower



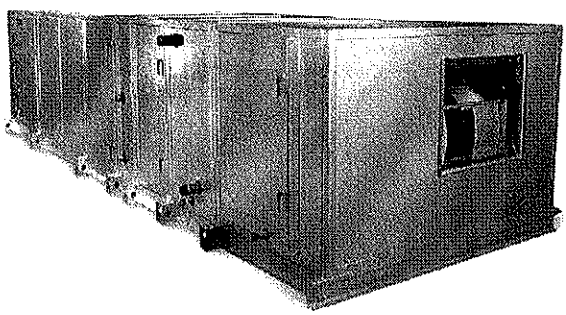
Load Basics

- **Chilled Water Coils Transfer Heat From Building Air To Chilled Water**





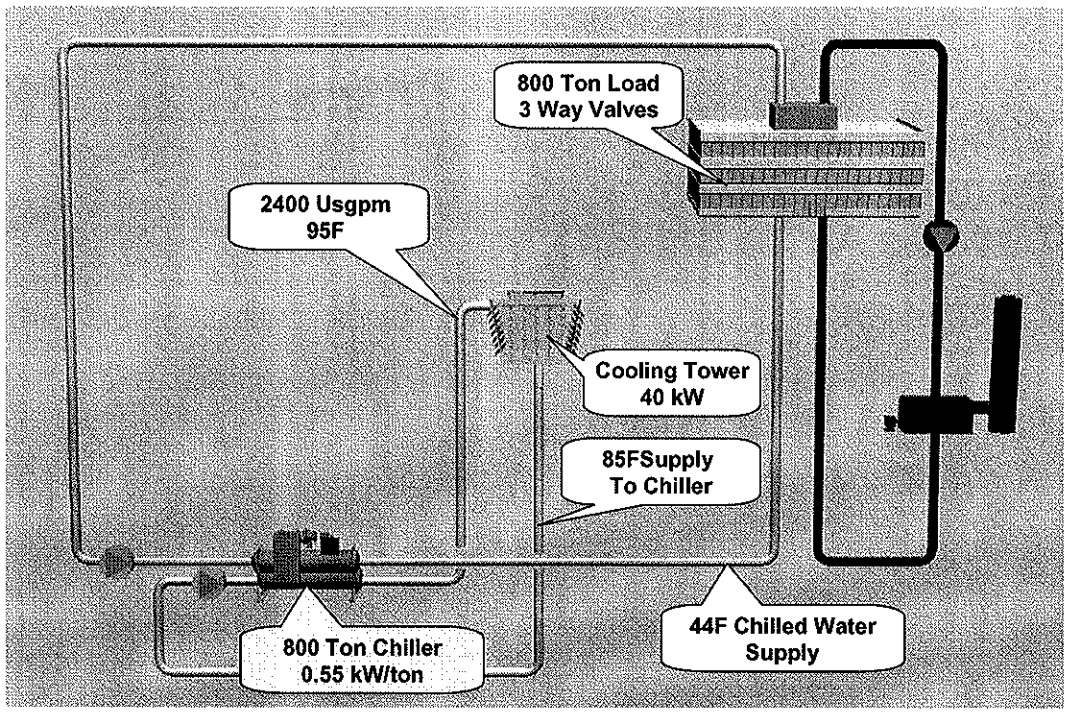
- **Process Loads**
  - **Cooling Jackets**



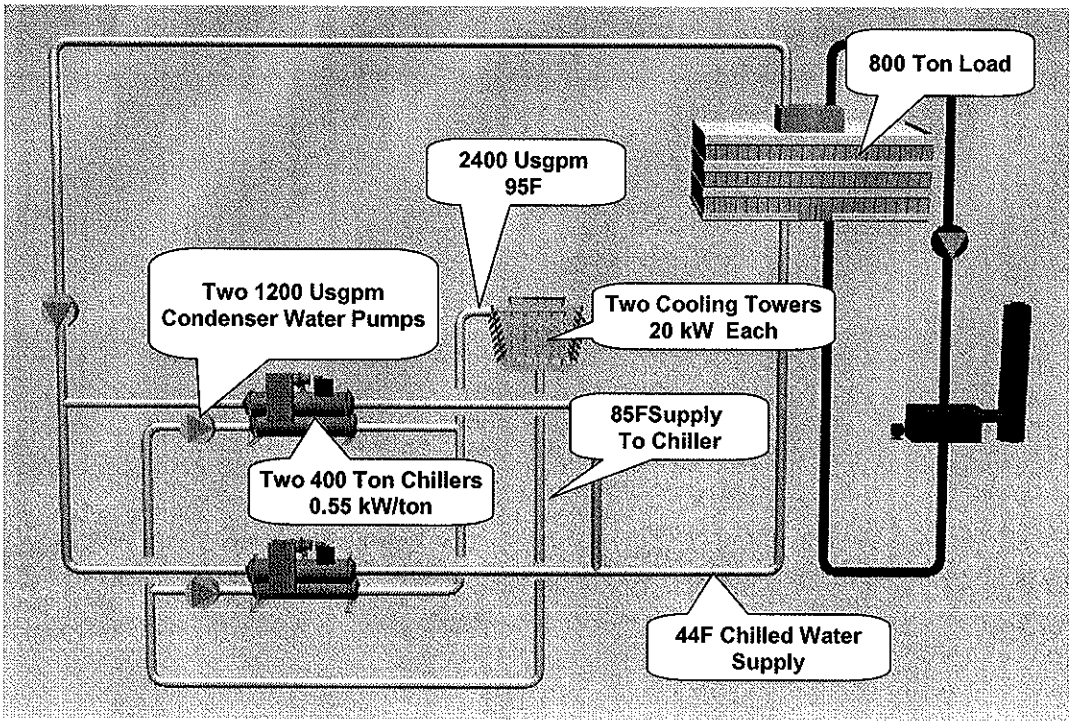
Single Vs. Parallel Vs. Series

- All Constant Flow Systems
- All Have Same Total Pump Power

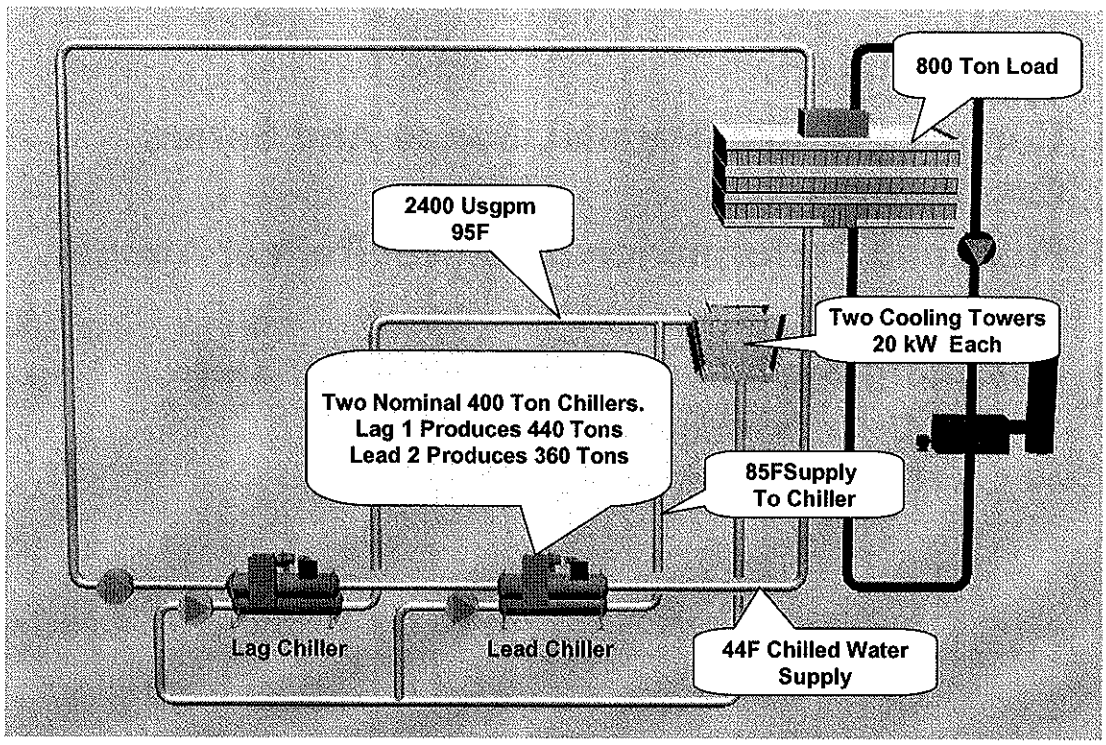
Single Chiller Design



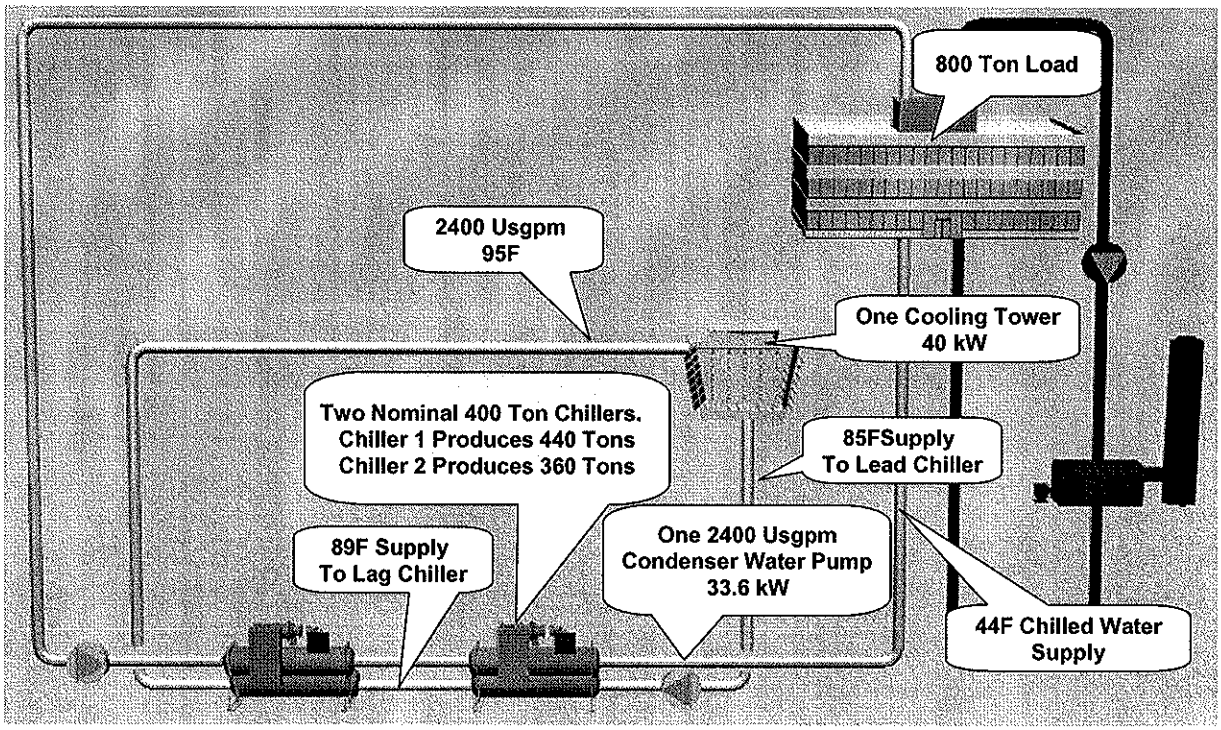
Parallel Chiller Design



Series Chiller Design



Series-Counterflow Chiller Design



Single Vs. Parallel Vs. Series

- Compare Single, Parallel, Series And Series Counterflow
- Full Load Performance Does Not Reflect Annual Performance
- Changing System Delta T Affects Ranking
  - High Delta Ts Promotes Series
  - Low Delta Ts Promotes Parallel

Single Vs. Parallel Vs. Series

- Annual Energy Analysis
- 44F Supply, 8 °F Delta T
- Series Best
- Order Changed From Full Load To Annual Analysis
- Series Counterflow Best Chiller Performance, Higher Pump Energy Usage

Systems	Chiller	Pumps	Cooling Tower	AHU Fans	Total
Single Chiller	258,344	175,689	16,597	334,237	784,867
Parallel Chillers	258,344	175,689	16,597	334,237	784,867
Series Chillers	237,607	160,762	19,243	334,237	751,849
Series Counterflow Chillers	227,718	173,564	19,153	334,237	754,672

HVAC - Prescriptive Method (6.3)

Hydronic Systems (6.3.4)!!!!!

- Variable Flow Required For Systems Over 10 HP (6.4.3.1)
  - Modulate Down To 50%
- Exceptions
  - Where Minimum Flow Is Less Than Flow Required By Equipment And < 75HP
- Individual Variable Flow Pumps
  > 100 Feet *And* 50 hp Motor
  - 30% Design Wattage At 50% Flow
  - Controlled As A Function Of Flow Or Pressure Differential

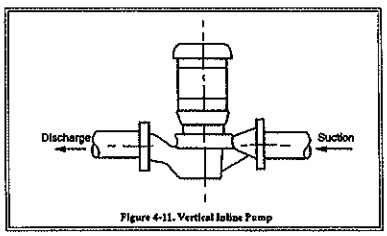
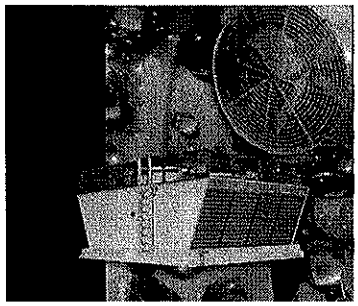


Figure 4-11. Vertical Inline Pump

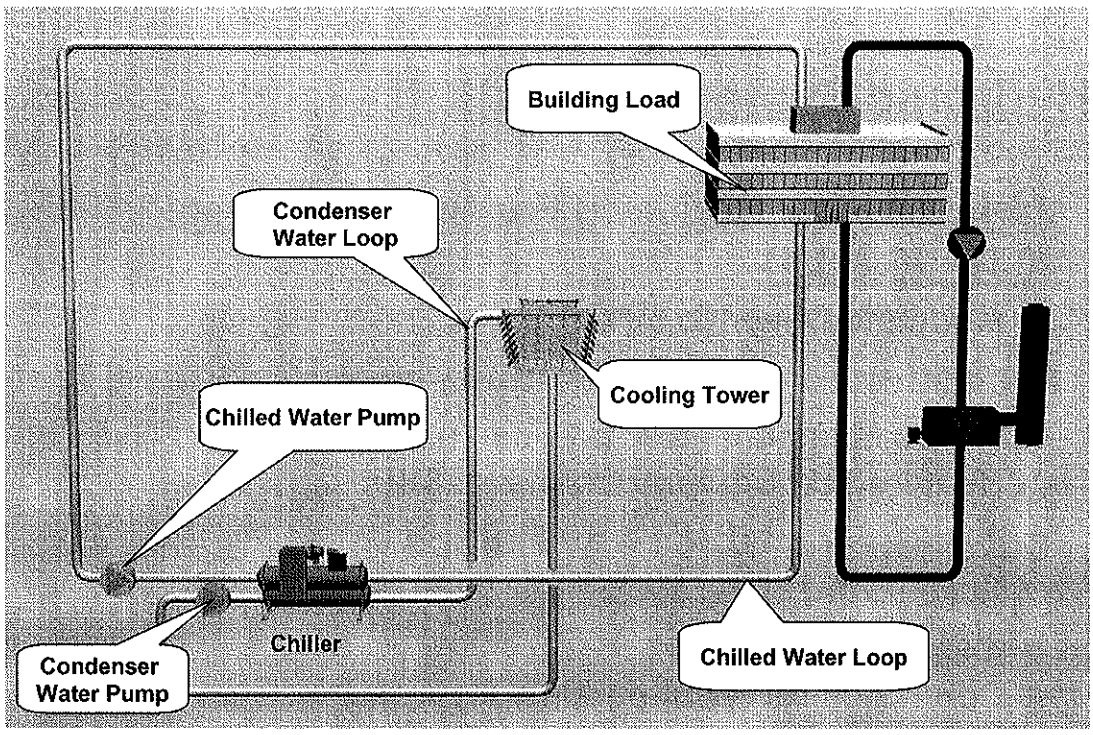
HVAC - Prescriptive Method (6.3)

Cooling Towers (6.3.5)

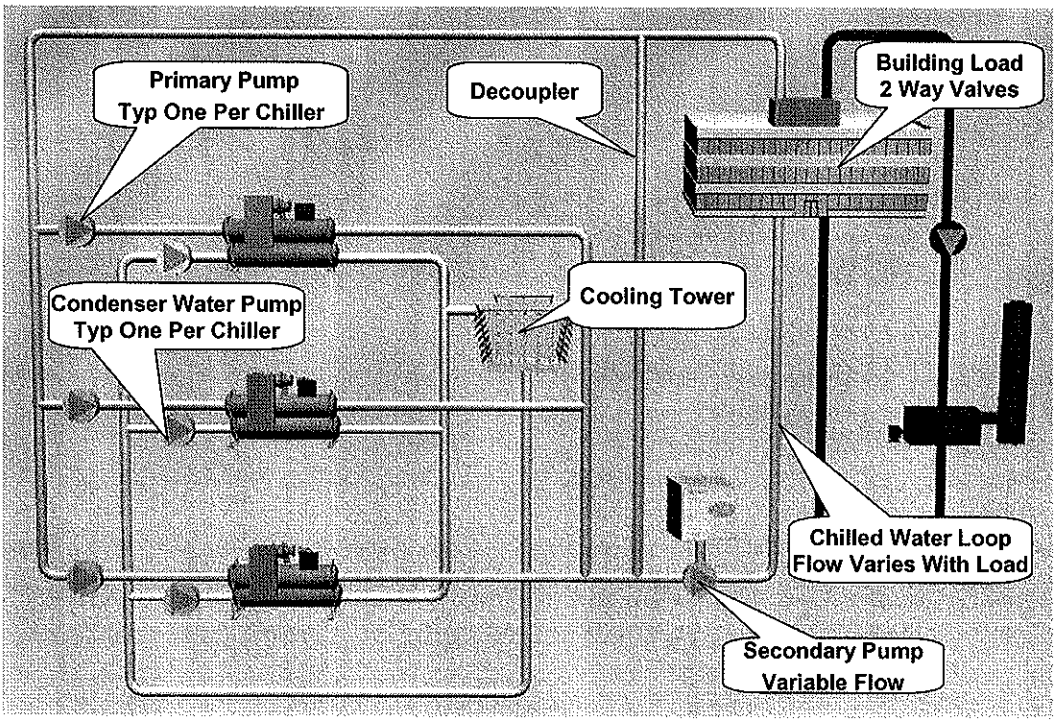


- **Fan Speed Control 7.5 hp And Up**
  - **Capability To Operate At 2/3 Speed Or Less (6.3.5.2)**
- **Exceptions**
  - **Condenser Fans On Multiple Circuits**
  - **Condenser Fans On Flooded Condensers**
  - **Climates With A Lot Of Cooling**
  - **1/3 Of The Fans On A Multiple Fan Application**

Basic System



Primary/Secondary Design

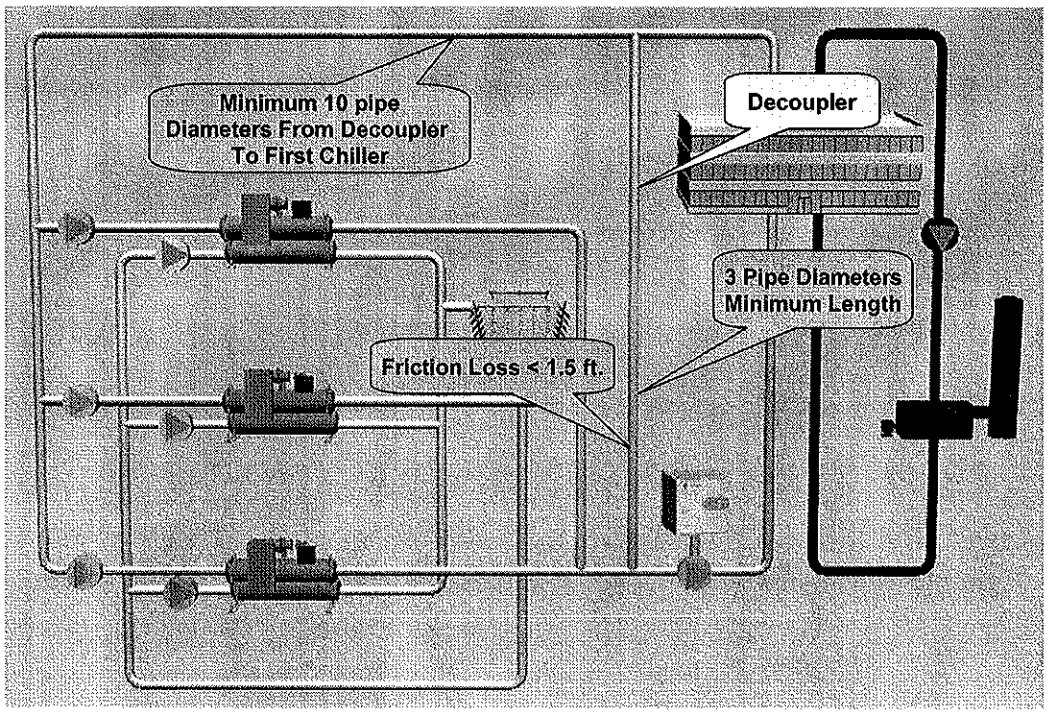


Primary/Secondary Design

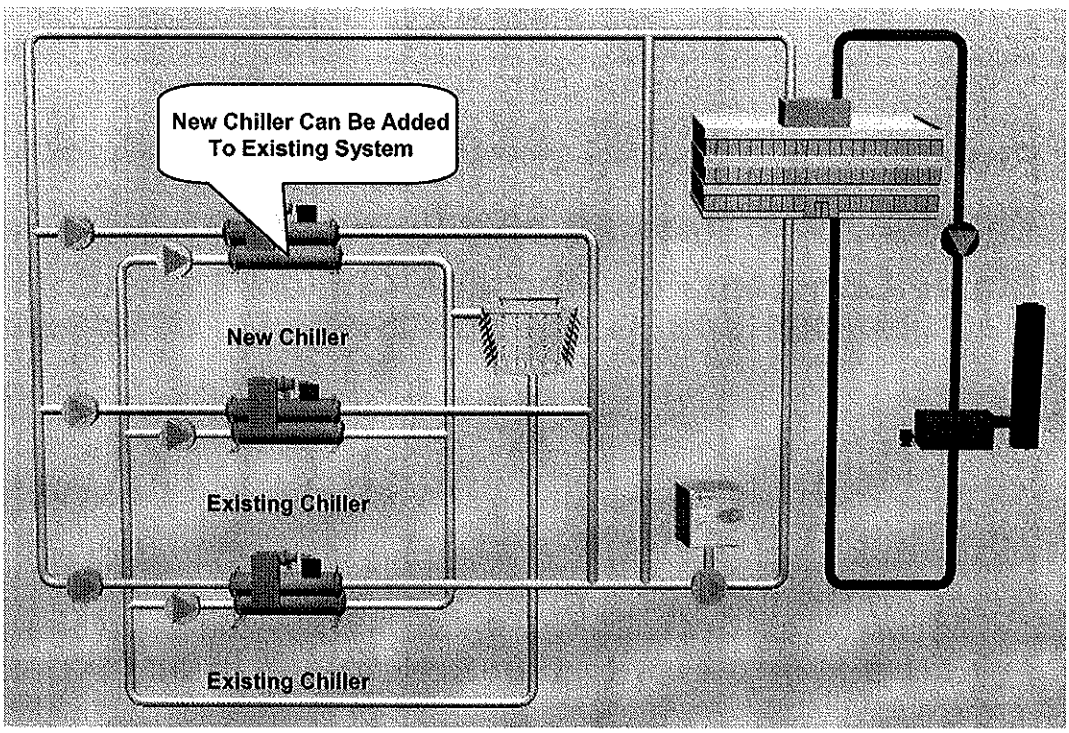
- Pros
- Variable Flow Through Secondary Loop
- Stable Constant Flow through Chillers

- Cons
- Complexity
- Low Delta T Syndrome
- Stepped Primary Flow
  - (More Pump Work Than Variable Primary Flow)

Decoupler Design



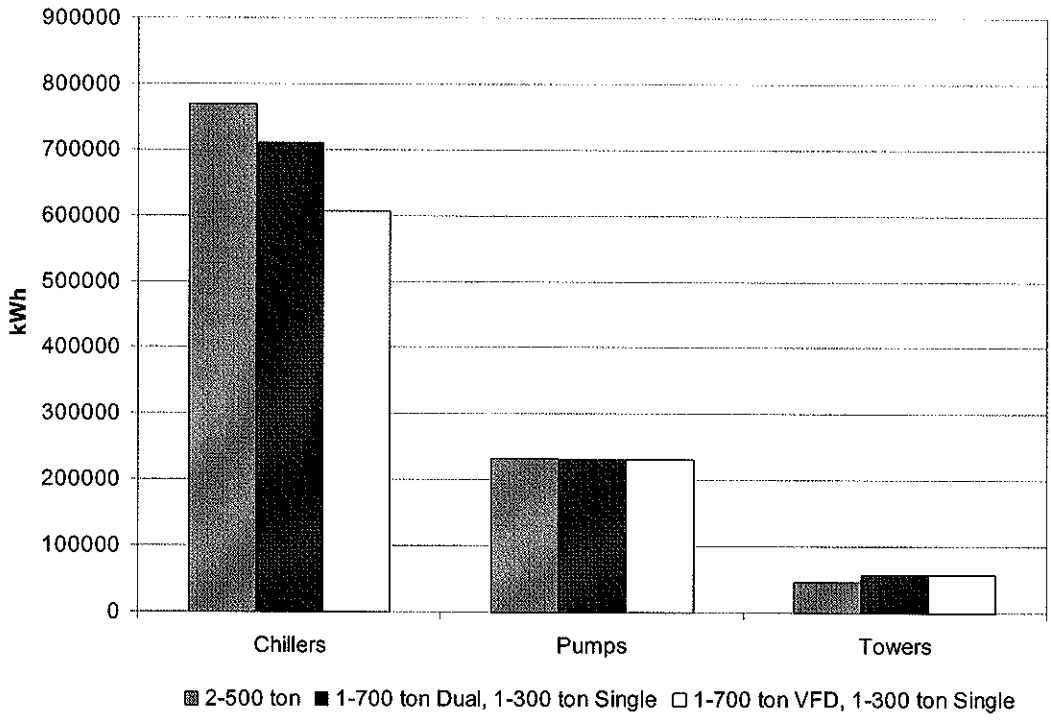
Expanding The System



Varying Chiller Sizes

- Different Sized Chillers May Be More Effective Than Even Sized Chillers
  - Depends On Load Profile
- Can Optimize Chiller By Operating 1 Large Chiller Rather Than 2 Smaller Chillers
- Possible Penalty From Ancillary Devices
- Constant Volume With Reheat Generally Works Well

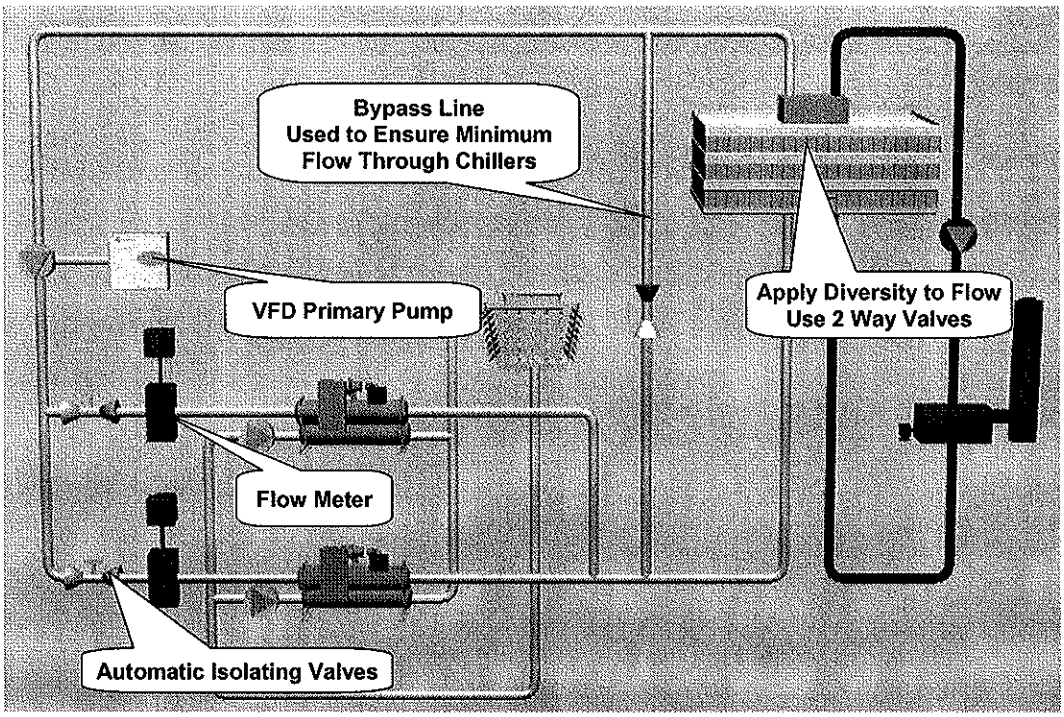
Varying Chiller Sizes



Why Variable Flow?

- **Piping And Pumps Based On Design Load Rather Than Connected Load**
  - **Lower First Cost**
  - **Lower Operating Cost**

Variable Primary Flow Design



Variable Primary Flow Design

- Design Flowrate Determined by Tube Velocity
  - Minimum 1.5 FPS (Based on a Reynolds # of 7500)
  - Maximum 12 FPS
- At Typical Conditions, 6-7 FPS
- Select Evaporator with More Passes & Higher Pressure Drop
- Minimum Flow Typically 50% or Less Of Design
- Bypass Must Be Sized to Maintain Minimum Flow Rate Of Largest Chiller

When to consider VPF ?

When:

•System flow can be reduced by at least 30% of design.

•Design affords greater cost savings than a "de-coupled" system.

•Operators will understand how the system works and will run it properly.

•The system can tolerate a modest variation in supply water temperature.

•A single chiller is being replaced and the primary flow can be varied.

•Variable Flow can be applied to Parallel as well as Series Flow.

Low Delta T-Definition

A Condition Whereby a Low Chilled Water Return Temperature Causes an Excessive Amount of Chilled Water to Circulate to Meet System Cooling Loads and Chillers Receiving the Low Temperature CHR CANNOT be Loaded to Their Design Capacity

Low Delta T Syndrome

Low Delta T Symptoms And Solutions

- 3 Way Valves
  - Don't Use Them
- Supply Air Setpoints Lowered Beyond Design
  - Valves Go Wide Open - No Control
  - Ensure Valves Are Tracking
- Valves Not Closed When Not Required
  - Ensure Valves Close When AHU Not In Use
- All System Components Not Designed For Same Delta T

Low Delta T Syndrome

Low Delta T Symptoms And Solutions Cont'd

- Coils And Valves Not Properly Selected
  - Select Correctly
- Coils Piped "Backwards"
  - Coils Must Be Piped So Water Is Counterflow To Air
- Improper Tertiary Piping
  - Ensure Tertiary Setpoint Is Above Chilled Water Setpoint
- Dirty Coils- Clean the Coils

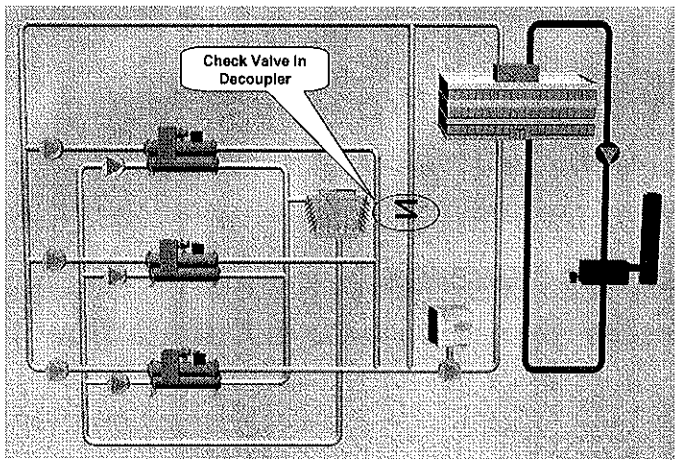
Low Delta T Syndrome

Low Delta T Symptoms And Solutions Cont'd

- Airside Economizers And 100% MUA
  - At Part Load, Reduced LMTD (Cooler Air) Reduces Coil Effectiveness And Return Water Temperature
  - Reduce Design Delta T (Not Necessarily A Good Thing)
  - Two Coils In Series
- Chilled Water Reset
  - Raising Chilled Water Temperature At Part Load Has Same Effect As Above
  - CWR Favors Systems With Smaller pumping Costs
  - Systems With Larger Pumping Losses Will Pay More In Pumping Costs Than Gained In Chiller Savings

Low Delta T Syndrome

Low Delta T Retrofit Solutions ?



- Check Valve Puts Pumps In Series
- Potentially Over-pump Chiller
- Can Starve Building
- Doesn't Fix Real Problem

Low Delta T Syndrome

Low Delta T Retrofit Solutions Cont'd

- VFD Or Dual Compressor Chillers
  - Excellent Part Load Performance Allows Two Chillers To Operate More Efficiently Even With Parasitic Losses
- Oversize Primary Pumps
  - Oversized Primary Pumps With VFDs Can Over Pump Chillers And Avoid Starting Additional Machines
- Variable Primary Flow
  - Easily Accommodates Over Pumping Chillers

Low Delta T Syndrome

Low Delta T Retrofit Solutions Cont'd

- **Reduce Delta T On Primary Side**
  - Larger Pumps And Piping Will Increase Capital Cost
  - Penalty At Full Load
- **Add Flow Control Valves At Each Coil**
  - Ensures Terminal Device Doesn't Exceed Design Flow
  - Space Cooling Not Satisfied
  - Increase System Pressure Drop
  - Adds Cost

Any More Fixes???

Is it this Simple?

Eliminate De-coupling Lines:

Modern Chillers and Boilers can work with variable flow

Employ Pumps in Series:

Modern DDC controls can Coordinate Series Pumping

Load Flow Control:

Avoid any Overflow by using local Temperature Sensors on the Return Line and Limit the Valve

The Hartman Company

Range Vs. Supply Water Temperature

- Flow (USgpm) = Load (tons) x 24 / Temp. Range (Delta t)
- Increasing Range Reduces Flow
  - Reduces Pipe, Pump And Motor Size

- Pump Power (hp) = Flow (USgpm)x Head (ft) / 3960 x Eff.
- Reducing Flow Reduces Pump Work
- This Is A Good Goal
- It Will Affect Every Part Of the Chilled Water System
  - Everything Must Be Considered

Range Vs. Supply Water Temperature

- Fan Work Savings For Small Changes (2 to 4°F) Don't Save Enough To Offset Chiller Penalty
- Especially For VAV
  - 20% Airflow Decrease
  - 35% Static Decrease
  - 49% Power Decrease
- Don't Lower Supply Water Temperature Just To Save Fan Work
- Don't Lower To Ensure Design Water Temperature Will Be Available At Coil
  - If You Assume Water Will Be 2°F Warmer At Coil Then You Assume 20% Of Chiller Capacity Lost To Heat Gain!

Myth # 1 - 42F Water Saves Money

Myth

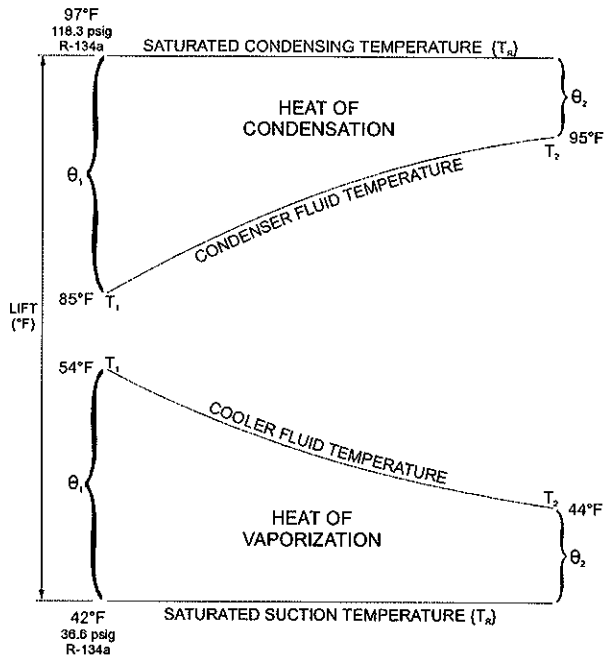
- *Colder* Water Is Always More Efficient

Fact

- 42F Water Doesn't Automatically Save Money
- Increasing The Range Saves Money
- Water Temperature Is A Balancing Act

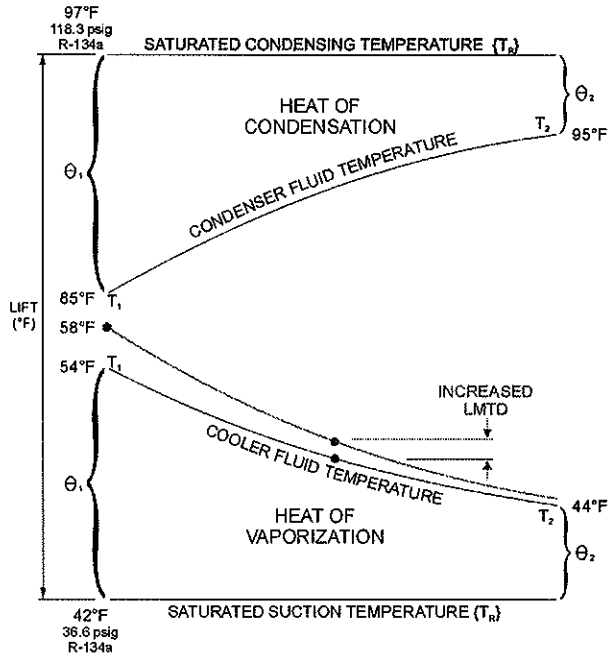
Range Vs. Supply Water Temperature

- Standard ARI Conditions
  - 54- 44F Chilled Water
  - 85 - 95F Condenser Water
- 10F Range
- 2F Approaches In Heat Exchangers
- 55F Lift On Compressor



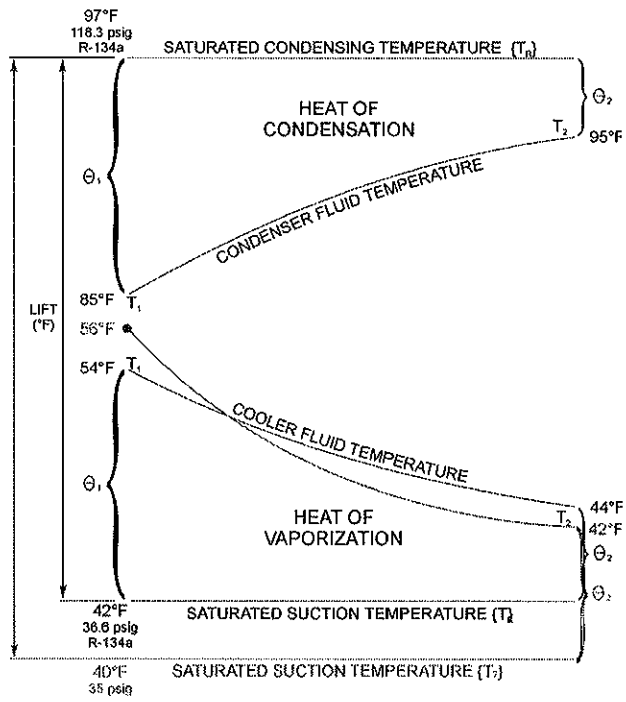
Range Vs. Supply Water Temperature

- Change To 14F Range
  - Smaller Pumps, Pipes etc.
- Maintain Supply Water Temperature
- LMTD Increases
  - Improves Chiller Performance
- Hurts Chilled Water Coil Performance
  - Deeper Coils Required
  - Increased Fan Static Pressure



Range Vs. Supply Water Temperature

- Maintain 14F Range
- Lower Supply Water Temperature To 42F
- 4% Increase In Compressor Lift
  - Chiller Performance Suffers
- Chilled Water Coil Performance Improves



Range Vs. Supply Water Temperature

- VAV Office Building In New York City
- Fixed Supply Water Temperature-44F
- Design Conditions
- Increase Chilled Water Range From 10 To 24F
- Fan Motor Goes From 94.8 HP To 114.7 HP (21%)
- Pump Goes From 38.5 HP to 16 HP (58%)

Run	Chiller Capacity Tons	Perform KW/ton	Chilled Water Temp Range (°F)	Pump HP	Coil APD (in. w.c.)	Rows/fins	TSP (in. w.c.)	Fan Motor size (HP)	Total Power (HP)
1	400	0.546	10	38.5	0.62	5/10	3	94.8	426.1
2	400	0.546	12	32.1	0.66	5/11	3.04	96	420.9
3	400	0.547	14	27.5	0.7	6/10	3.08	97.3	417.6
4	400	0.547	16	24	0.79	6/12	3.15	99.5	416.3
5	400	0.543	18	21.4	0.87	8/9	3.25	102.7	415.3
6	400	0.543	20	19.2	0.94	8/11	3.32	104.9	415.3
7	400	0.543	22	17.5	1.1	10/10	3.48	109.9	418.6
8	400	0.543	24	16	1.25	12/10	3.63	114.7	421.9

Range Vs. Supply Water Temperature

- Fixed Supply Water Temperature (44F)
- Increase Chilled Water Range From 10 to 24F
- Annual Energy Analysis
- System Peaks At 16F Range

Run	C.W. Range (°F)	Chiller ($/yr)	Pumps ($/yr)	Tower Fan ($/yr)	S.A. Fan ($/yr)	Total ($/yr)
1	10	26,074	15,175	1,591	28,275	71,115
2	12	26,096	13,784	1,593	28,560	70,033
3	14	26,167	12,792	1,594	28,846	69,399
4	16	26,211	12,055	1,597	29,350	69,213
5	18	26,081	11,489	1,601	30,070	69,241
6	20	26,126	11,034	1,604	30,574	69,338
7	22	26,259	10,784	1,619	31,726	70,388
8	24	26,358	10,487	1,625	32,810	71,280

Range Vs. Supply Water Temperature

- **Switch To Constant Volume With Reheat**
- **Increase Chilled Water Range From 10 to 24F**
- **Annual Energy Analysis**
- **System Peaks At 14F**
  - **Fan Penalty Outweighs Pump Savings**

Run	C.W. Range (°F)	Chiller ($/yr)	Pumps ($/yr)	Tower Fan ($/yr)	S.A. Fan ($/yr)	Total ($/yr)
1	10	40,035	19,842	2,821	70,957	133,655
2	12	40,034	18,013	2,821	71,954	132,822
3	14	40,224	16,728	2,831	72,396	132,179
4	16	40,327	15,765	2,839	73,657	132,588
5	18	40,174	15,025	2,852	75,455	133,506
6	20	40,285	14,429	2,863	76,715	134,292
7	22	40,526	13,963	2,884	79,595	137,193
8	24	40,772	13,692	2,912	82,283	139,659

Range Vs. Supply Water Temperature

- **Declining Supply Water Temperature (44 To 38F)**
- **Increase Chilled Water Range From 10 To 24F**
- **Annual Energy Analysis**
- **System Peaks At 16F Range And 42F SWT**

Run	C.W. Range (°F)	C.W. S.T. (°F)	Chiller ($/yr)	Pumps ($/yr)	Tower Fan ($/yr)	S.A. Fan ($/yr)	Total ($/yr)
1	10	44	26,074	15,175	1,591	28,275	71,115
2	12	44	26,096	13,784	1,593	28,560	70,033
3	16	44	26,211	12,055	1,597	29,350	69,213
3	14	42	27,733	12,790	1,593	28,573	70,689
4	16	42	27,779	12,039	1,593	28,570	69,981
5	18	40	29,371	11,462	1,594	28,584	71,011
6	20	40	29,351	11,002	1,596	28,872	70,081
7	22	38	30,365	10,623	1,596	28,881	71,465

Myth #2 - 2 GPM/Ton Is Good

Myth

- *Increasing* Condenser Delta T Saves Energy

Fact

- It Almost Always Raises The Operating Cost

Condenser Water Range

- Increase Condenser Water Range From 10 To 15F
- Annual Energy Analysis
- System Peaks At 10 Range
- It Costs More To Operate A System At Higher Ranges

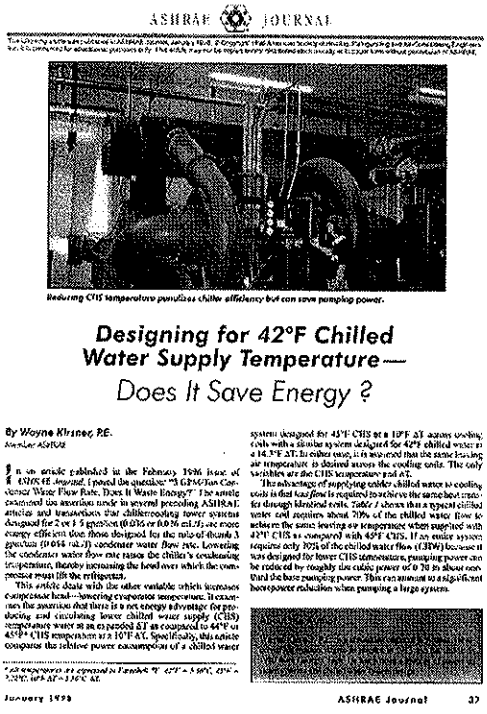
Run	Cond .W. Range (°F)	Chiller ($/yr)	Pumps ($/yr)	Tower Fan ($/yr)	S.A. Fan ($/yr)	Total ($/yr)
1	10	26,074	15,175	1,591	28,275	71,115
2	11	27,084	14,562	1,592	28,283	71,521
3	12	27,517	14,049	1,592	28,286	71,444
4	13	28,094	13,616	1,592	28,290	71,592
5	14	28,527	13,245	1,592	28,293	71,657
6	15	29,057	12,923	1,593	28,297	71,870

Summary

- The Savings Comes From The Increased Range
- Playing With Range And Supply Temperature Is Capital Expense Exercise
- 15F Condenser Delta T Is A Waste Of Energy

Engineer's View

"We may conclude that unless the system pumping head at 2.4 gpm/ton is well in excess of 100ft, there is an energy penalty to design chillers for 42F..."



ASHRAE JOURNAL

Reducing CHS temperature penalizes chiller efficiency but can save pumping power.

Designing for 42°F Chilled Water Supply Temperature— Does It Save Energy ?

By Wayne Kirsner, PE.
Member ASHRAE

January 1998 ASHRAE Journal 37

Condenser Water Relief

Chiller Type	Performance Improvement
	Percent kW /°F condenser water
W/C Recip.	1.1 to 1.3
W/C Scroll	1.3 to 1.5
W/C Screw	1.6 to 1.8
W/C Centrifugal	1.0 to 1.6
W/C Centrifugal VFD	2.4 to 2.6
Absorption	1.4 to 1.5

Condenser Water Relief

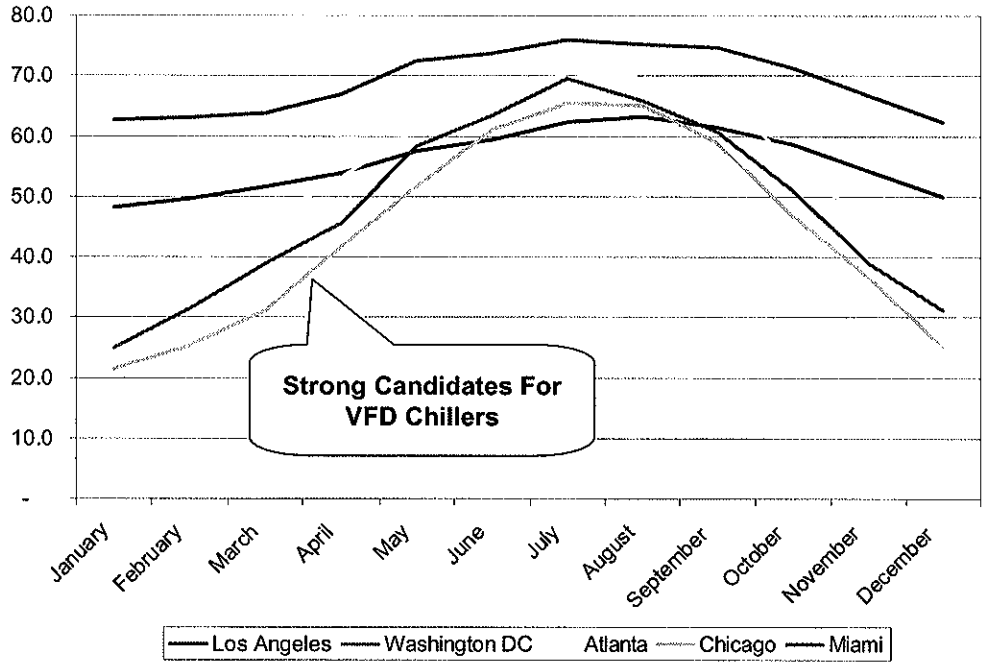
- **Goal Is To Use Least Amount Of Power Between Chiller And Tower Fan Motor**
  - **When In Doubt, Optimize The Chiller!**
- **Dictated By**
  - **Wetbulb Profile**
  - **Load**
  - **Equipment Selection**
  - **Control Sequence**

Condenser Relief

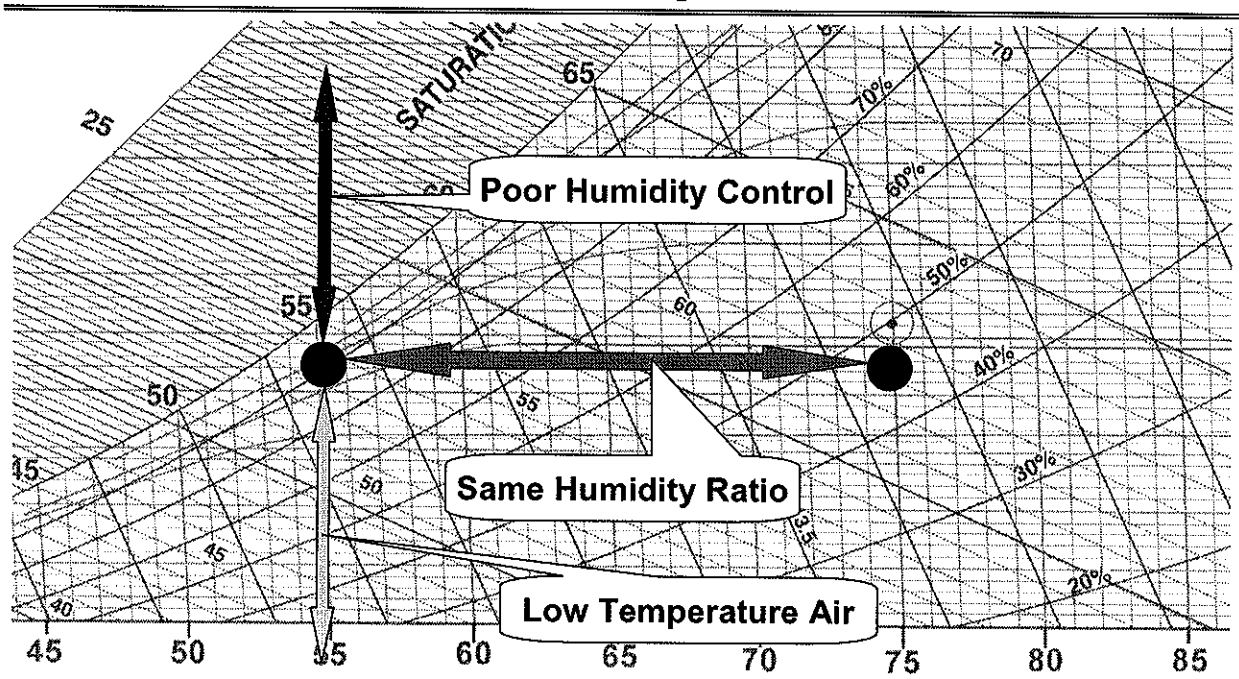
Summary

- Condenser Relief Is A Easy Way To Improve Performance
  - It May Be Necessary For Stable Operation
  - It Is Assumed In Industry Standards
- Oversizing Cooling Towers Usually Pays For Itself
- Good Controls Are A Must

Annual WB In Major US Cities



Why Use Low Temperature Air?

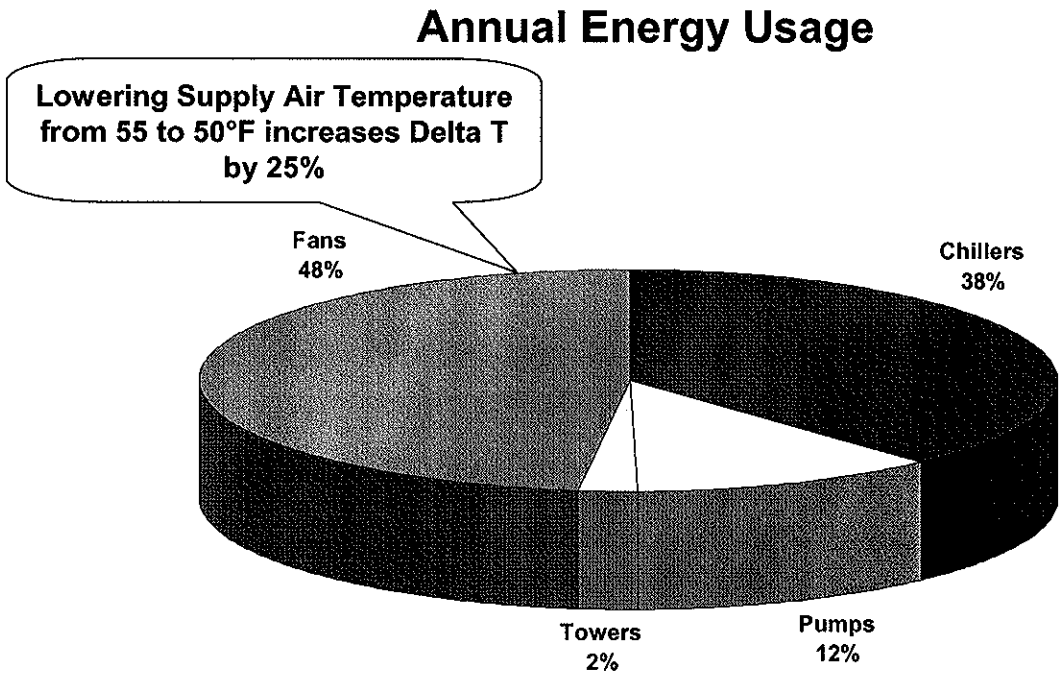


Concept: Reduce Construction Costs; Equipment Room and Ceiling Plenum Space

Low Temp Vs. Optimal Air

- **Optimal Air**
  - 45 to 52°F From AHU
  - Originally Used In Grocery Stores
  - Does Not Use More Energy Than Conventional Systems
  - Uses Conventional Design Practices
- **Low Temperature Air**
  - 39 - 42°F From AHU
  - Often Used With Ice Storage Systems
  - Will Use More Energy Than Conventional Systems
  - Requires Specialized Design

Benefits Of Optimal Air



Benefits Of Optimal Air

- Lowering Supply Air Temperature Will:
  - Increase Delta T And Lower Supply Air Volume
  - Reduce Duct Size (Capital Savings)
  - Lower Space RH (Higher Possible Space Setpoint)
- Quieter Due To Smaller Fans And Ductwork
- Lower Filter Face Velocities Can Allow Higher Efficiencies With Same APD
- Lower RH Can Help Control Mold Issues

Optimal Air

SAT From 55°F To 45°F

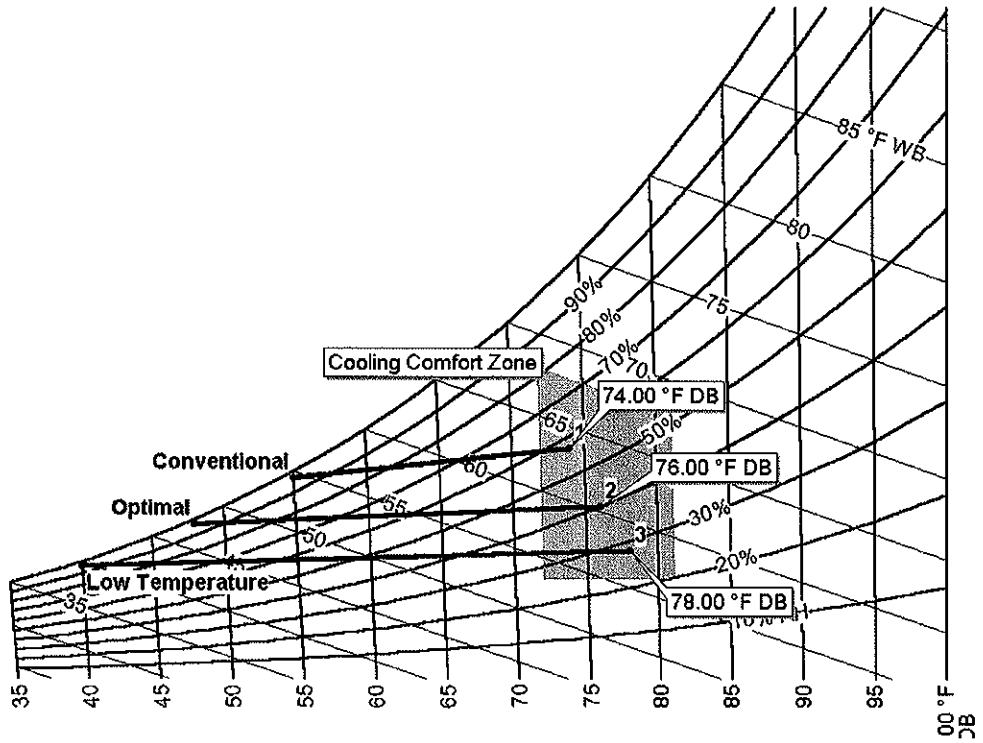
SAT	TSP	Supply Air Volume	Design Cooling	CWST	Perform-ance	Room Setpoint	Chiller Work	Fan Work	Total Work
F	" w.c.	cfm	Tons	F	kW/ton	F	kWh/yr	kWh/yr	kWh/yr
55	3	152,686	473	44	0.55	75	219,605	202,736	422,341
54	3.02	145,415	475	43.6	0.554	75	226,594	194,061	420,655
53	3.04	138,805	477	43.2	0.558	75	236,352	186,175	422,527
52	3.06	132,770	479	42.8	0.562	75	241,228	178,975	420,203
51	3.08	127,238	481	42.4	0.566	75	248,661	172,375	421,036
50	3.1	122,149	483	42	0.57	75	255,777	166,303	422,080
49	3.12	117,451	485	41.6	0.574	75	261,285	160,698	421,983
48	3.14	113,101	486	41.2	0.578	75	269,044	155,508	424,552
47	3.16	109,061	488	40.8	0.582	75	276,512	150,689	427,201
46	3.18	105,301	490	40.4	0.586	75	286,605	146,202	432,807
45	3.2	101,791	492	40	0.59	75	292,832	142,014	434,846

ARI is 44F. It Works! Optimal Air Changes The CWST. Analyze Your Design!

Optimal Air

- **Balance Point Is Lowest Supply Air Temperature That Will Not Increase Annual Operating Cost**
- **At Balance Point**
  - **Trade Off Fan Savings Against Refrigeration Work**
  - **Probably Won't Save Annual Energy**
  - **Will Save Significant Capital**
    - **Air Side System Is 23% Smaller!**
  - **Overall Design Energy Consumption May Be Up But Annual Energy Will Break Even**
    - **Larger Refrigeration Compressors Will Offset Smaller Fan Motors**

Space Conditions



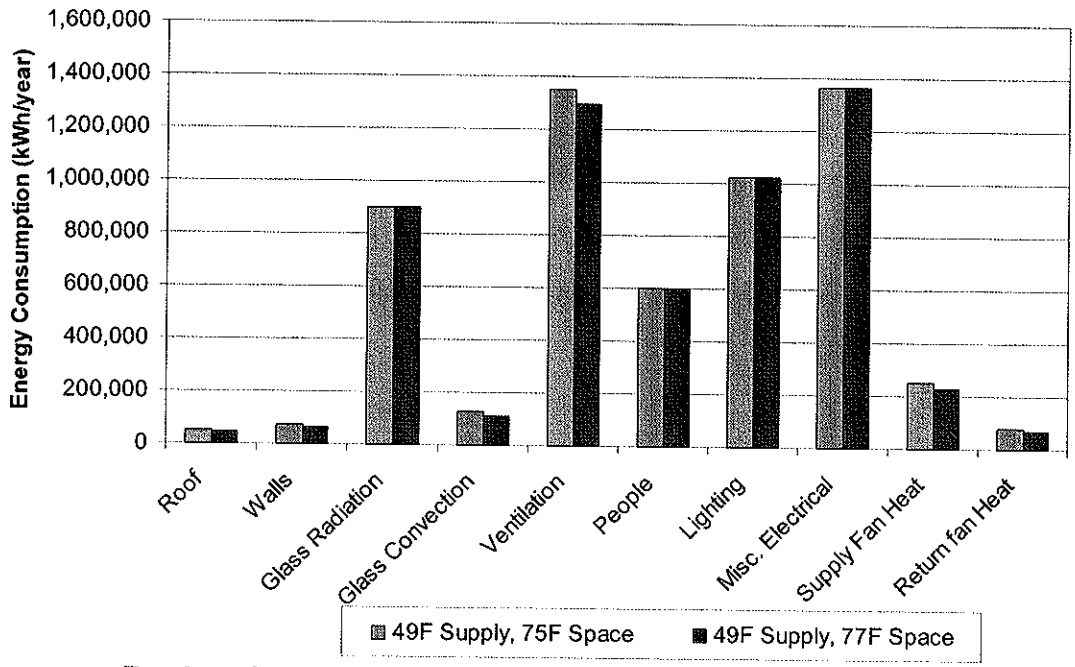
ASHRAE Std 55: Comfort Zone-80% Of People Are Satisfied

Space Conditions

- **Based On 49F Supply Air**
- **Raise Space Temperature From 75F to 77F**
  - **Now Airside System Is 29% Smaller!**
  - **Plus 6% Annual Energy Savings!**
  - **(Improved Refrigeration, Reduced Fan Work)**

SAT	TSP	Supply Air Volume	Design Cooling	CWST	Perform-ance	Room Setpoint	Chiller Work	Fan Work	Total Work
°F	" w.c.	cfm	Tons	°F	kW/ton	°F	kWh/yr	kWh/yr	kWh/yr
49	3.12	117,451	485	41.6	0.574	75	261,285	160,698	421,983
49	3.12	112,549	480	41.6	0.574	76	255,231	153,695	408,926
49	3.12	107,997	475	41.6	0.574	77	249,350	147,438	396,788

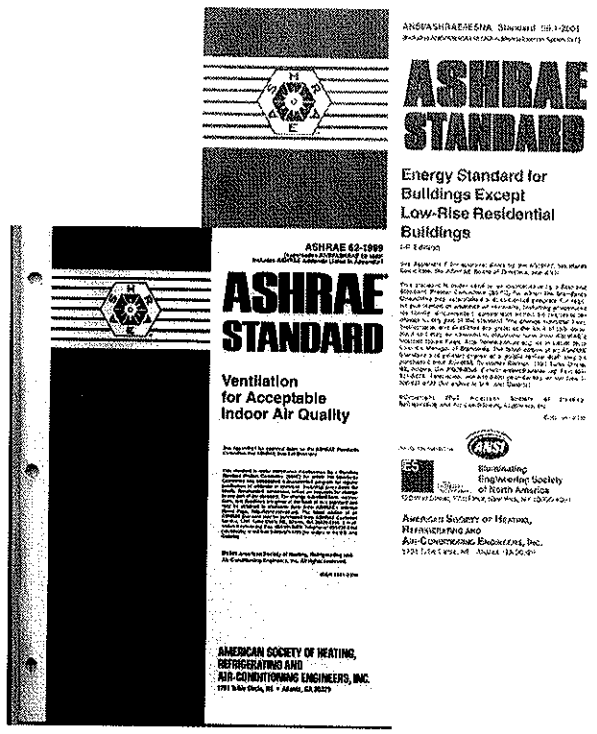
Space Conditions - 77F, 40% RH



Design Cooling Loads Are Reduced

Complying With STD 90.1 & 62.1

- **STD 90.1 Gives Fan Power Credits For Optimal Air**
- **Provides Credits For Non ARI Operating Conditions**
- **Optimal Air Is Does Not Affect Std 62**
- **Lower RH Probably Improves IAQ**



Design Considerations

- Optimal Air Design Is Basically The Same As Conventional Design
  - Conventional Diffuser Will Work
  - Slightly Different Controls
  - HVAC Equipment Must Be Capable Of Optimal Air Temperatures
- Best Suited For High Sensible Heat Ratio Buildings
  - Office Buildings
- Building Envelope Must Be Good

Design Considerations

- Load Calculations Are Same AS Conventional System Except For Increased Delta T
- Need To Evaluate Balance Point
  - Energy Analyzer™
- Evaluate Space Temperature Setpoint
  - 75°F is Safe
  - 78°F Will Reduce Supply air Volume Further

Blowthrough Vs. Drawthrough Discussion

- Fan heat Will Raise Supply Air Temperature About 3°F
- Blowthrough Has The Fan Upstream Of Cooling Coil So Fan Heat Is Added Prior To Being Cooled
  - Supply Air Temperature Is Coil Leaving Air Temperature
- Drawthrough Has the Fan Downstream Of Cooling Coil So Fan Heat Is Added After Being Cooled
  - Supply Air Temperature Is Coil Leaving Temperature Plus Fan Temperature Rise (3°F)

Blowthrough Vs. Drawthrough

- Blowthrough Makes Sense With High Sensible Heat Ratio Applications
- Can Provide Same Supply Air Temperature For 10% Less Refrigeration
- Can Provide 10% Colder Air For Same Refrigeration

Air Distribution

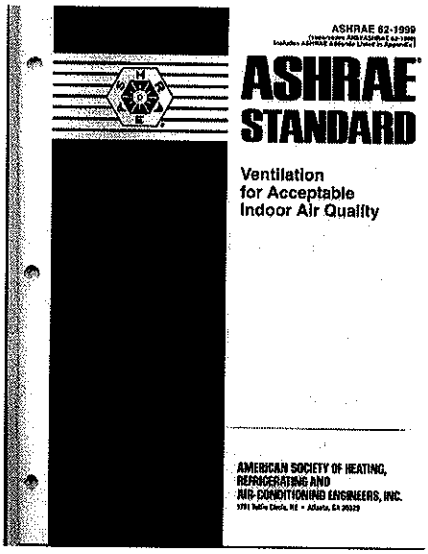
- As Supply Air Temperature Is Lowered, Air Distribution Becomes More Challenging
- Five Main Types Of Diffusers
  - Slot
  - Radial
  - Jet

  } Best Choice For Optimal Air
  - Swirl
  - Perforated

Life Cycle Analysis

- May Not Be Required If Optimal Air System Uses Same Amount Of Energy And Has A lower Capital Cost
- Duct Sizing Will Vary Proportionally With Air Volume Reduction
  - Price Will Not Drop Quite As Quickly
- HVAC Equipment Will Be A Wash
  - AHU Goes Down - Refrigeration goes Up
- Possible Space Savings
- Envelope Changes on Retrofit Applications Can Be An Issue

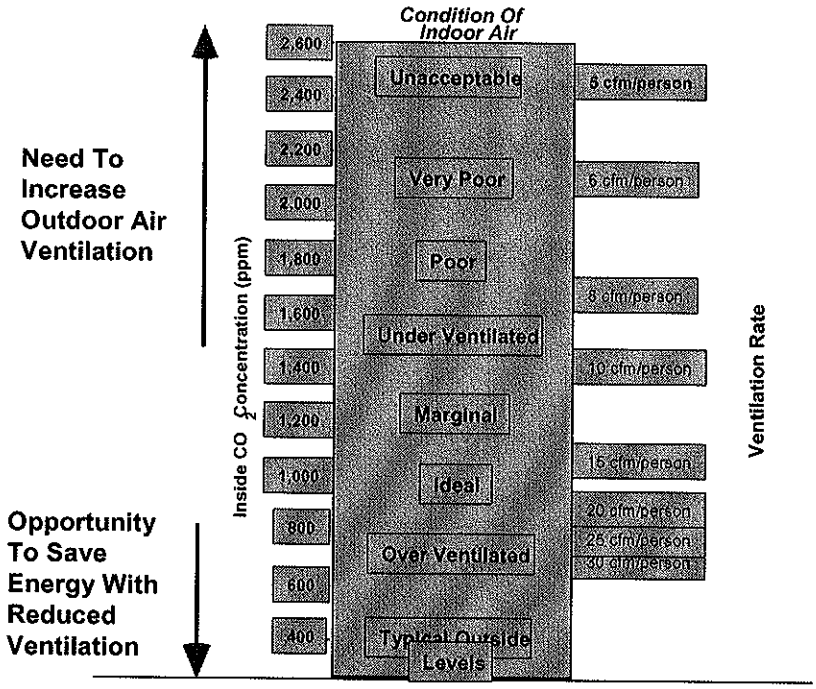
ASHRAE Std 62.1-1999



- Revised In 1999
- High Profile Standard
- Indoor Air Quality Procedure
- Ventilation Rate Procedure

CO_2 & Ventilation Rates

- CO_2 Good Indication Of Ventilation Rates



ARI Standard 550/590-98

Know your Standards!

1998
STANDARD for

WATER
CHILLING
PACKAGES
USING THE
VAPOR
COMPRESSION
CYCLE



Standard 550/590

Date _____ Page _____

4301 NORTH FAIRFAX DRIVE • ARLINGTON, VIRGINIA 22203

99% Of All Operating Hours Are At Part Load

The New Industry ARI Standard -1998

Part Load Analysis (IPLV)

% Load	Old % Hrs	New % Hrs
100	17	1
75	39	42
50	33	45
25	11	12

Systems Solution

Equipment - Properties

- Different Chillers Operate Differently
  - VFD Chillers Need Condenser Relief
  - Duals Are Most Efficient At 50% Load
  - Absorption And Gas Driven Chillers Operate On a different Fuel
- System Must Take Advantage OF Chiller Properties To Get Best Results
- Part Load Performance Is Usually More Important Than Full Load Performance

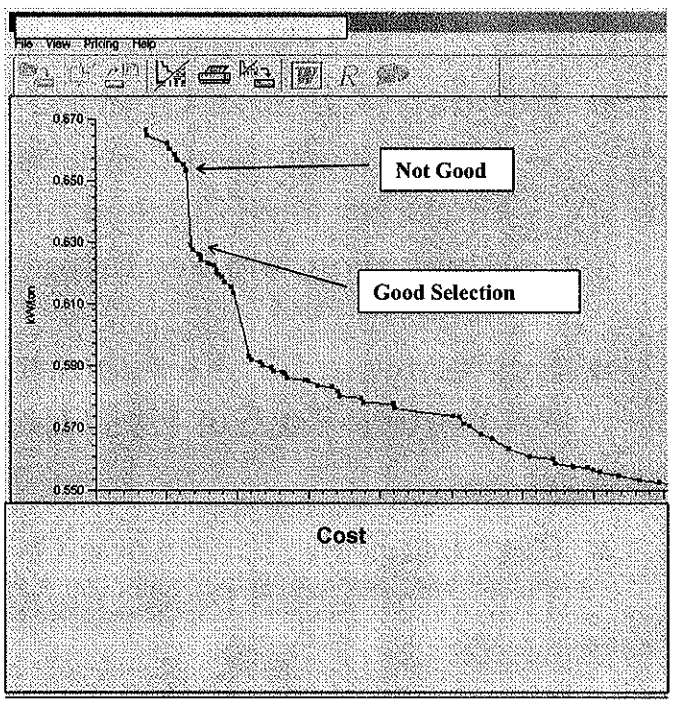
Equipment

Summary

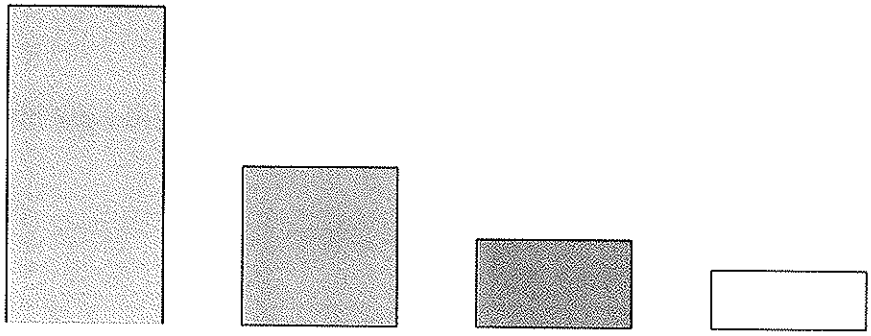
- **Be Careful That High Performance Equipment Can Pay For Itself**
  - **Ask For A Couple Of Selections And Some Budget Pricing**
- **Understand And Take Advantage Of Chillers Operating Properties**

Water Cooled - Optimizing Selections

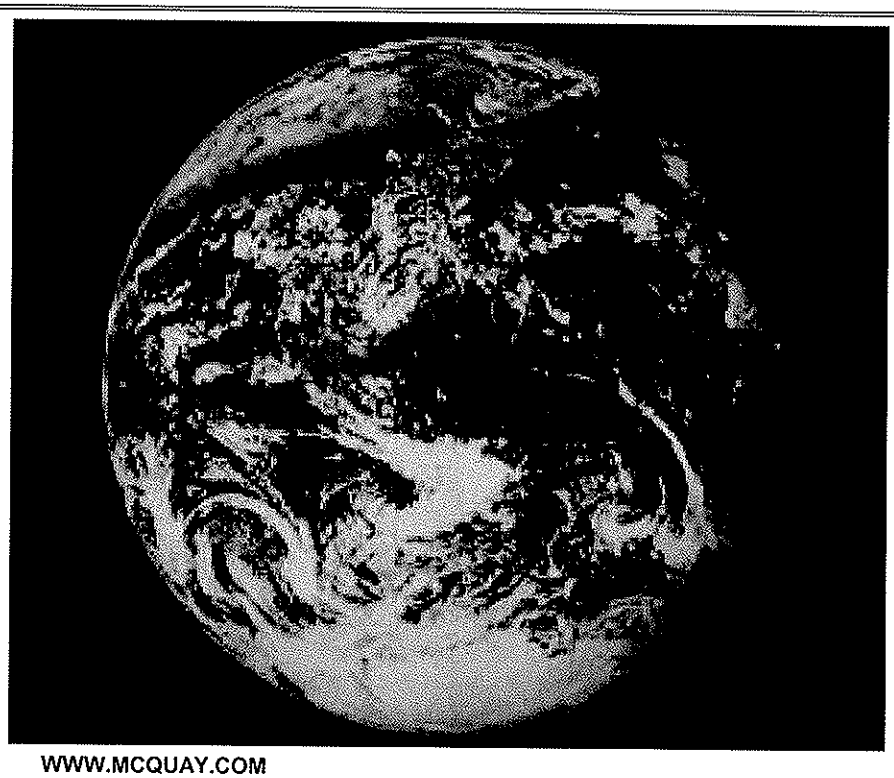
- **Enter Design Data Into The Program**
  - **Start The Process**
    - **Choose The Chiller After A Large Vertical Drop**
    - **Look At The Number Of Selections And Prioritize**
    - **Analyze Options**



Various 500 Ton Chillers



THANK YOU



WWW.MCQUAY.COM

DAIKIN

McQuay International

REFRIGERANTS UPDATE

Please mute cell phones!

Refrigerants Update

Julian de Bullet
Director of Industry Relations
McQuay International 703-395-5054

ASHRAE Distinguished Lecturer

ASHRAE Vice President
(2001-2003)

What is a Refrigerant?

- 1. In a refrigerating system, the medium of heat transfer which picks up heat by evaporating at a low temperature and pressure, and gives up heat on condensing at a higher temperature & pressure.

- 2. (Refrigerating fluid) fluid used for heat transfer in a refrigerating system which absorbs heat at a low temperature and low pressure of the fluid and transfers heat at a higher temperature and higher pressure of the fluid, usually involving changes of state of the fluid.

Ref: ASHRAE Terminology of Heating, Ventilation, Air Conditioning, & Refrigeration

Refrigerants:

Where We Were

History of Refrigerants

- 1830s - Jacob Perkins - Vapor Compression (ether)
- 1851 - John Gorie - Patent for Vapor Compression Cycle
- 1859 R-717 / R-718 (Ammonia / Water)
- 1866 CO_2 - Naval Applications
- 1873 - R-717 (Ammonia) Commercial Refrigeration - Karl Linde
- 1875 - R-764 (Sulfur dioxide)
- 1920s -R-600a (Isobutane) & R-290 (Propane)
- 1922 - Willis Carrier - R-1130 (Dielene)
- 1926 - R-30 (Methylene Chloride)

Toxicity - Best Minds Tried to Solve

Solving Problem of Toxicity was a Large Problem to Development of Refrigeration

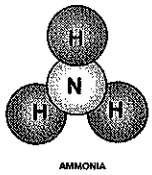
1927 - Leo Szilard & Albert Einstein

- **Improved on von Platen / Munters Absorption Design**
- **Electromagnetic Pumping Patent**

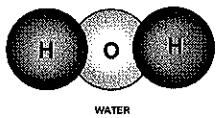


Natural Refrigerants

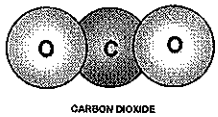
- Ammonia (NH_3) R-717
  - Efficient
  - B2 Classification
  - Industrial Applications
- Water R-718
  - Absorption Chillers
  - Large Volumetric Flow
- Carbon Dioxide (CO_2) R-744
  - Low Critical Point



AMMONIA



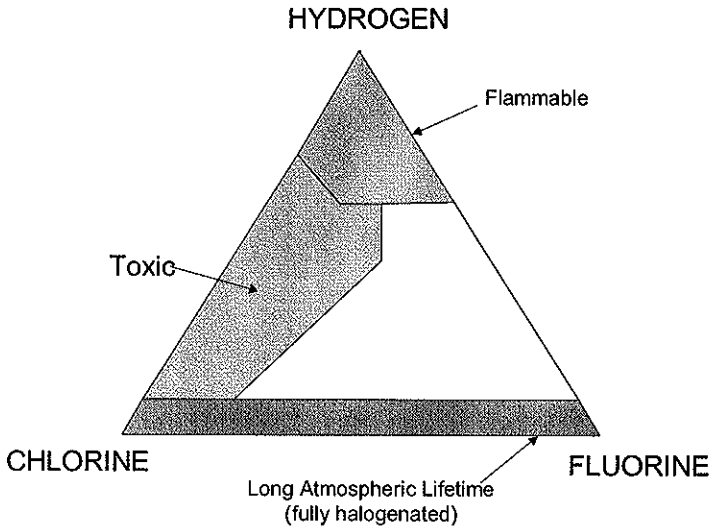
WATER



CARBON DIOXIDE

Fluorocarbons

- **CFCs, HCFCs and HFCs**
- **Limited Combinations**
  - **Adding Chlorine Or Bromine Increases ODP**
  - **Adding Fluorine Increases GWP**
  - **Adding Hydrogen Increases Flammability And Lowers Atmospheric Lifetime**





Mario Molina and F. Sherwood Rowland

Montreal Protocol

- Rowland And Molina Theory Suggests Ozone Depletion In Upper Atmosphere
- 1985 Antarctic Ozone Depletion Observed
- 1987 Montreal Protocol Signed
- Included Phase Outs For Specific Chemicals In Developed Countries 1974

Refrigerants Circa 1987

- Common HVAC Refrigerants Include CFC-11, CFC-12 (ODP=1.0), HCFC-22 (ODP=0.05)
- Chiller Efficiency 0.70-0.80 kW/ton
- Low Pressure Chillers Lose 25% Of Charge Annually

Corrective Measures

- HFC-134a (ODP=0) Replaces CFC-12
- HCFC-123 (ODP=0.02) Replaces CFC-11
- Chillers Tightened Up (High Eff. Purge, PRVs, Blankets Etc.)

Phase Outs

- CFCs (11,12) Phased Out By 1996
- Toxicity Issues With HCFC-123 Arise
- Initial AEL Is 10 PPM (Later Raised To 50 PPM As Per ASHRAE STD 34)
- HCFC-123 Classed As A B1 Refrigerant

Phase Outs-Developed Countries

Montreal Protocol

- HCFC Schedule;
- 1996 Cap (1989 HCFC Production Plus 2.8% Of 1989 CFC Production)
- 2004 65%
- 2010 35%
- 2015 10%
- 2020 0.5% No New Equip.
- 2030 0%

USA - EPA

- 2004 End Production R-141B
  - Lower Cap For R-22 to 65%
- 2010 Service Only For R-22
- 2015 10% For R-22
- 2020 End Production For R-22
  - Service Only For R-123
- 2030 End Production For R-123

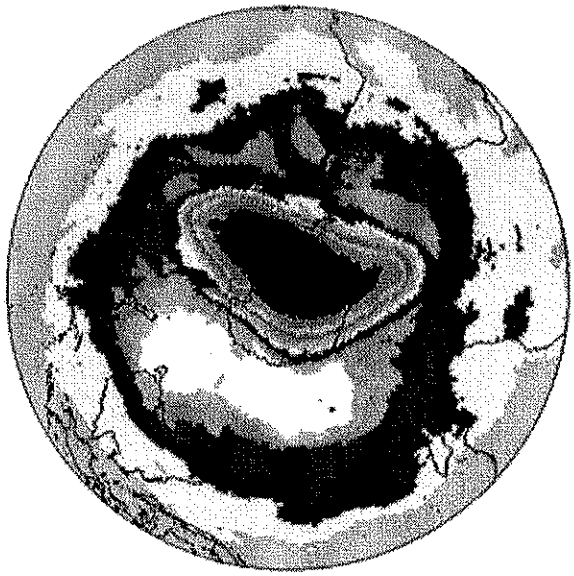
HCFC Phase Outs

- USA At 82% Of Cap In 1996,7
- USA At 92% Of Cap In 1998
- EPA Has Given Advanced Notice Of An Allocation Procedure To Ensure Cap Is Not Exceeded

Where We Are

Stratospheric Ozone Depletion

- 1994 ODS Concentrations Peaked In Lower Atmosphere
- ODS Concentrations Peaking Now In Stratosphere
- Concentrations Of CFC Substitutes Increasing
- ODS Levels Back To Norm By 2050 ?



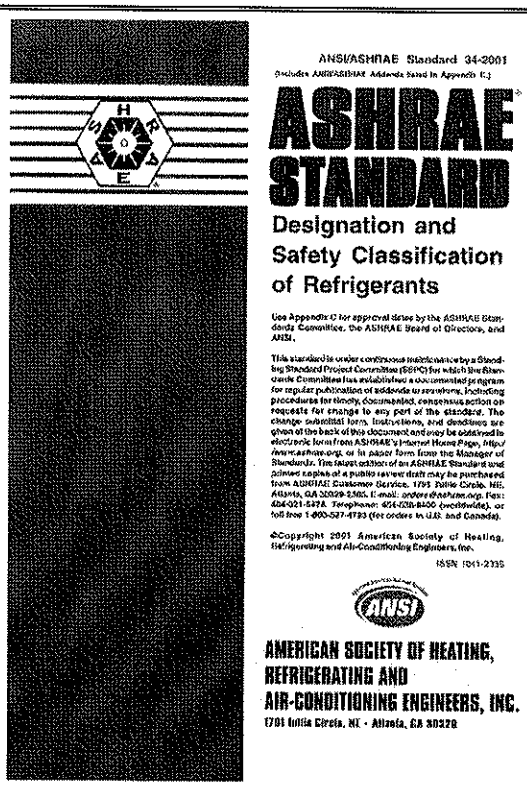
Record loss of ozone over Antarctica

5 November 2003

Recent analysis of satellite-based measurements by NIWA scientists shows that record amounts of ozone were destroyed over Antarctica in September this year. On each day over the period 23 to 26 September the amount of ozone lost over Antarctica exceeded the previous daily record of 45.5 million tons recorded in 2000.In 25 September this year a record loss of 47.3 million tons was reached.

'This is a significant amount of ozone,' said NIWA scientist Greg Bodeker. 'It's equivalent to about 8 kg of ozone for every person on Earth. The amount lost this year (47 million tons) is the amount that would be required to restore Antarctic ozone levels to the minimum levels that were measured before the onset of the Antarctic ozone hole in the early 1980s.'

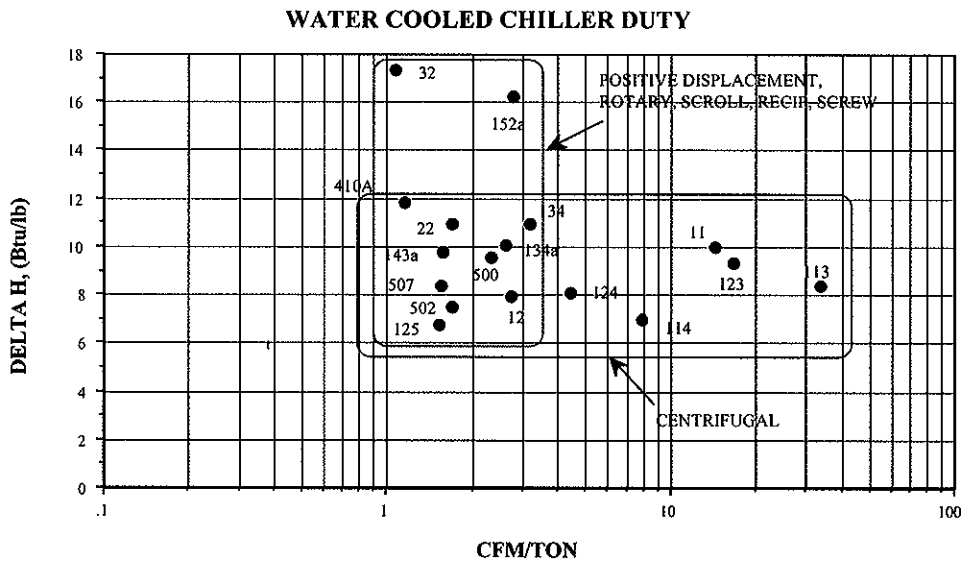
ASHRAE Standard 34



ASHRAE 34 SAFETY GROUPS

High Flammability	A3 Propane Butane	B3
Low Flammability	A2 R-142b, 152a	B2 Ammonia
No Flame Propagation	A1 R-11, 12, 22, 114, 500, 134a	B1 R-123, SO_2
	Lower Toxicity	Higher Toxicity

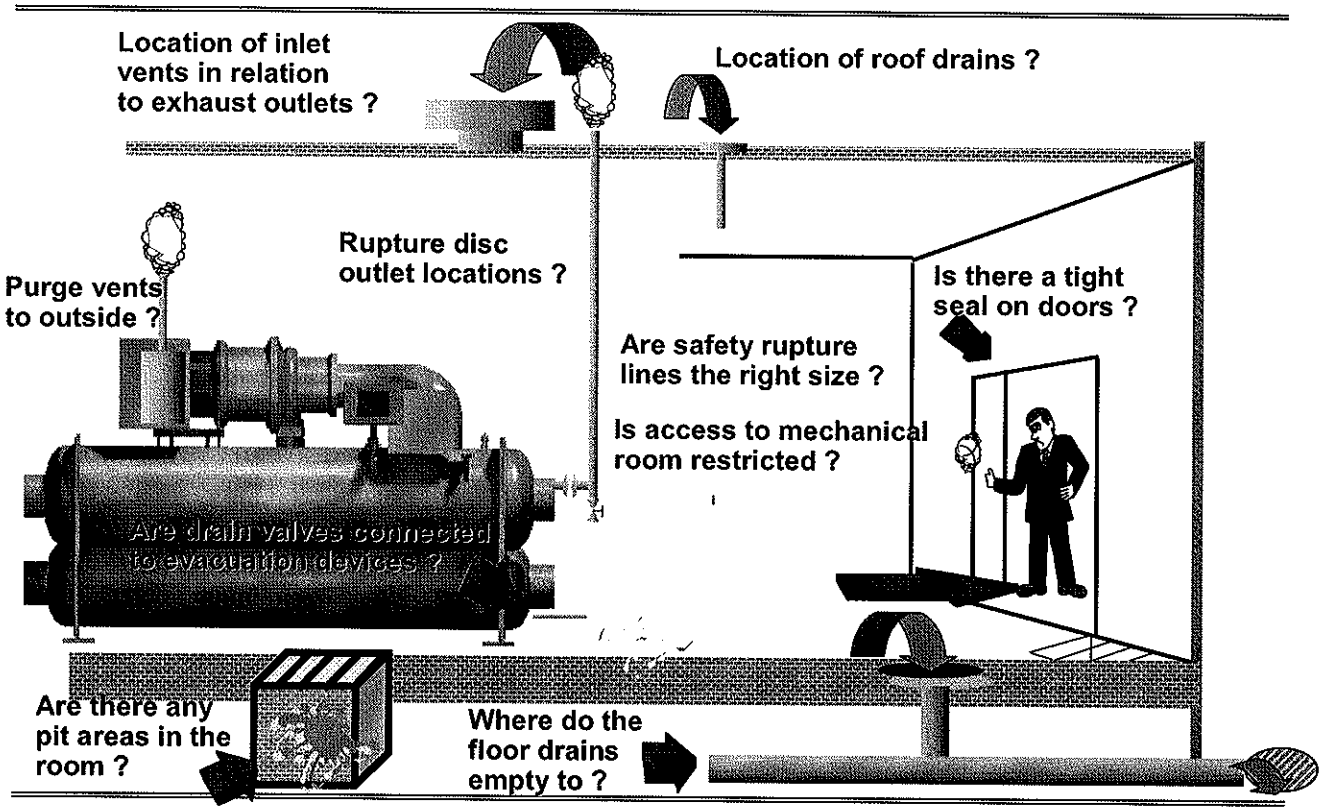
Application Considerations
Refrigerants & Compressors



ASHRAE Standard 15



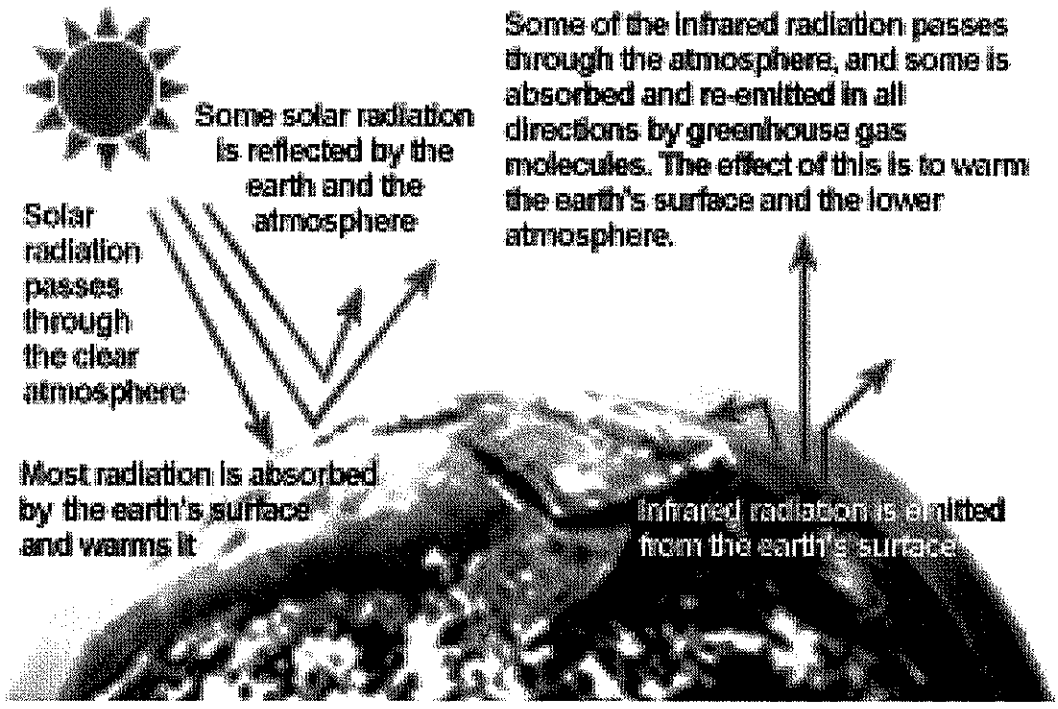
Mechanical Room Safety Check:



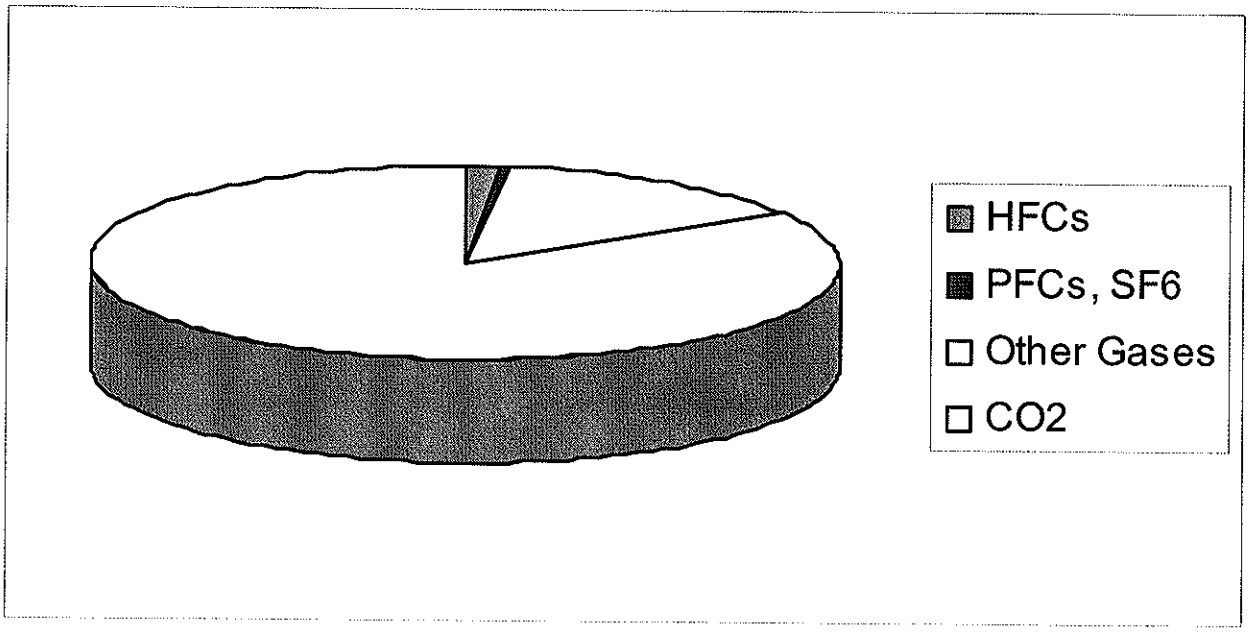
Global Warming

- CO_2 Concentrations 30% Above Pre-Industrial Levels (275 PPM)
- CH_4 Concentrations Doubled
- Average Global Temperature Up ½ To 1°F
- Average Sea Level Rise 4 To 10"
- Almost Guaranteed To Double CO_2 Levels
- Business As Usual Will Mean Quadruple CO_2 Levels

The Greenhouse Effect



Emissions, USA 1995



Kyoto Summit 1997

- Annex 1 (Developed) Countries Agreed To Limit Greenhouse Gas Emissions By 2008-2012
- Canada - 6% Of 1990 Levels
- USA - 7% Of 1990 Levels
- Note Projected Growth Will Increase Emissions From 1990 Levels By 20 To 30%

Kyoto Details

- Each Government Controls Internal Policy
- Only Overall Target Must Be Met
- 6 Gases Included (CO_2, HFCs, CH_4, PFCs, SF_6, NO_2)
- No Specific Gas Phase Outs
- "Sinks" Allowed
- Quota Trading Allowed

Kyoto Issues

- Developing Countries Not Included
- Currently USA Is Largest Emitter (4,881,000 Metric Tons Equiv. CO_2)
- China Is Second (2,667,000)
- Canada 9th (409,862)
- In 15 yrs Developing Countries Will Exceed Annex 1 Countries

Kyoto Status

- Ratified By Enough Countries (Needed 55 Parties And 55% Of Annex 1 Emitters)-Russia Was Last Vote Needed
- U.S. Against It
- Montreal Protocol Was A Warm Up For The Kyoto Protocol!

Energy Efficiency

- Kyoto Protocol Will Drive Energy Efficient Buildings
- 1/3 Energy Buildings
- 1/3 Energy Industry
- 1/3 Energy Transportation

TEWI

- **Total Equivalent Warming Impact**
- **Includes Direct (Leakage) And Indirect (Energy Consumed) Contributions**
- **TEWIs Change Depending On Power Grid**
- **ASHRAE Has Adopted TEWIs**

(Climate Change PP/PS -www.ashrae.org)

The New Industry ARI Standard -2001

Part Load Analysis (IPLV)

% Load	Old % Hrs	New % Hrs
100	17	1
75	39	42
50	33	45
25	11	12

Systems Solution

Current Issues

- Kyoto Approval
- Conflict Between Montreal And Kyoto Protocols On Refrigerants
- New ASHRAE 90.1 2004
- New Std Will Produce 16% Site Savings And 20% Source Savings
- 4% Of Savings From Mechanical Systems

Where We're Going

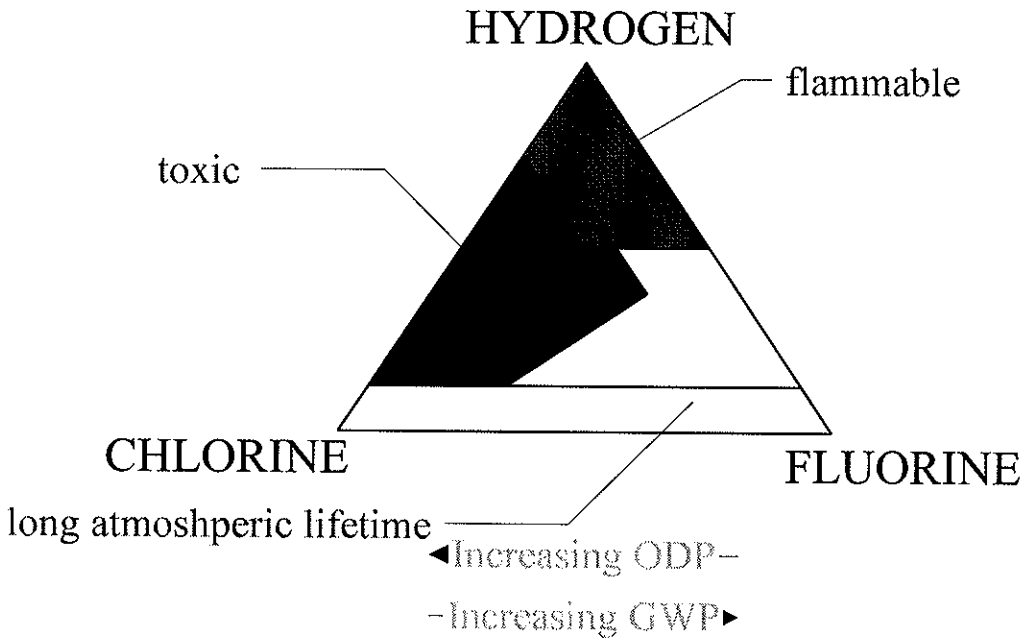
Montreal-Kyoto Conflict

- Montreal Protocol Bans HCFC-123,22 While Promoting HFCs (134a)
- Kyoto Protocol Wants Emission Reduction Of HFCs
- Kyoto Protocol Does Not Address Specific Gases, Only Overall Result Hence HFCs *Do Not* Have A Phase Out Date

Montreal-Kyoto Conflict

- Best Guess Is TEWI Will Be Adopted And HFCs Will Live A Long And Useful Life
- HCFCs Will Be Phased Out As Per Montreal Protocol

Other Refrigerants



Other Refrigerants

- **R-718 Water**
- **R-717 Ammonia**
- **R-744 CO_2**
- **R-290,600,600a (Propane, Butane, Isobutane)**
- **R-407c**
- **R-410a**

Water And Ammonia

- Water Can Be Used In Absorption
- Efficiency Is The Issue (COP=1)
- Ammonia Is Very Efficient But Toxic (B2)
- Limited To Large Controlled Mechanical Plants - Industrial, Institutional

CO_2

- One Of The Original Vapor Compression Refrigerants
- Evaporator 450 psi
- Condenser 1240 psi
- ASHRAE Research Looking Into CO_2 Technology

Flammable Refrigerants

- Propane And Butane
- Major Safety Concerns
- Research By UN TOC
- 35% Domestic Market In N. Europe
- 8% Domestic World Market
- Handful Of Large Jobs
- Not Popular In N. America

Blends

- Zeotropes R-400 Series
- Simple Mixtures, Will Segregate
- Azeotropes R-500 Series
- Behave Like Pure Mixtures

Blends Continued

- R-407C (HFC-32/HFC-125/HFC-134a)
- Very Close To HCFC-22
- Glide Issue - Not Good For Flooded Systems
- The Front Runner For R-22 Retrofits
- Can Use Existing Compressor Technology
- Not Very Efficient
- Limited Use In New Designs

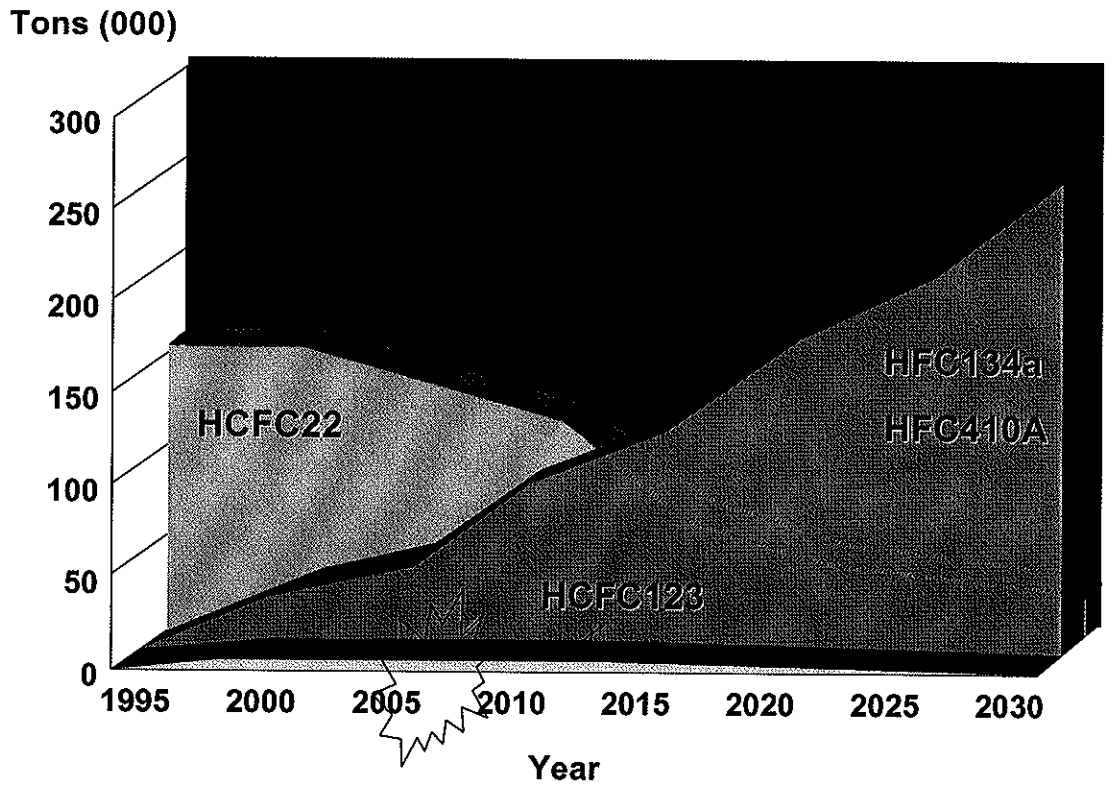
Blends Continued

- R-410a (HFC-32/HFC-125)
- Higher Operating Pressure (350 psi)
- Some Glide But Workable In Flooded Systems
- Front Runner For Redesigned Equipment To Replace Current R-22 Equipment

Best Guess

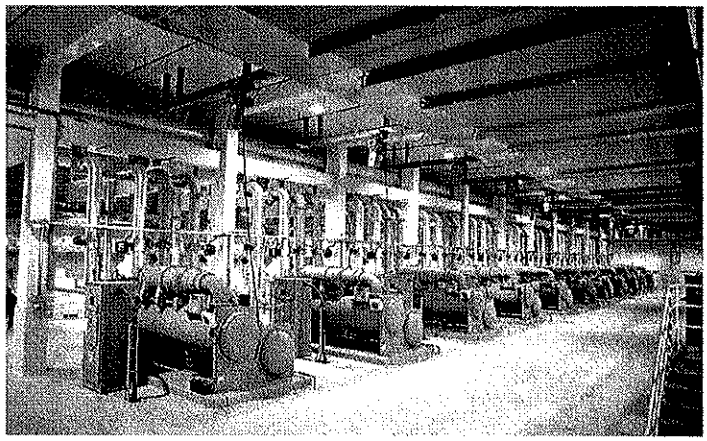
- HFC-134a Here To Stay
- HCFC-22 Will Be Replaced By R-407c And R-410a
- HCFC-123 Will Phase Out With The Montreal Protocol
- Ammonia And Water Niche Uses
- Flammables Limited To Domestic Use

AVAILABILITY OF REFRIGERANT
(Dupont & ICI Projections)



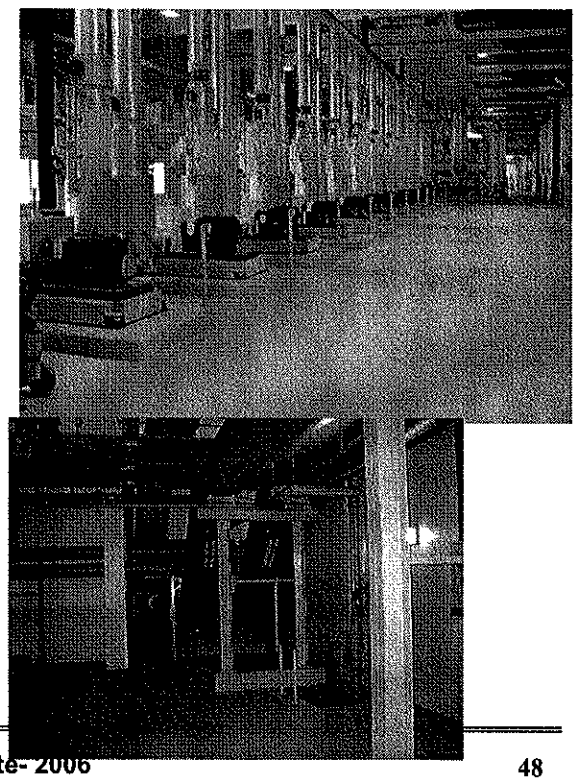
Big Plant Layout

- 60,000 Tons
- 40/58 °F Chilled Water
- 80,000 USgpm Chilled Water
  - 48" Dia. Pipe
- 93/103 °F Cond. Water
- 180,000 USgpm Cond. Water
  - 54" Dia. Pipe



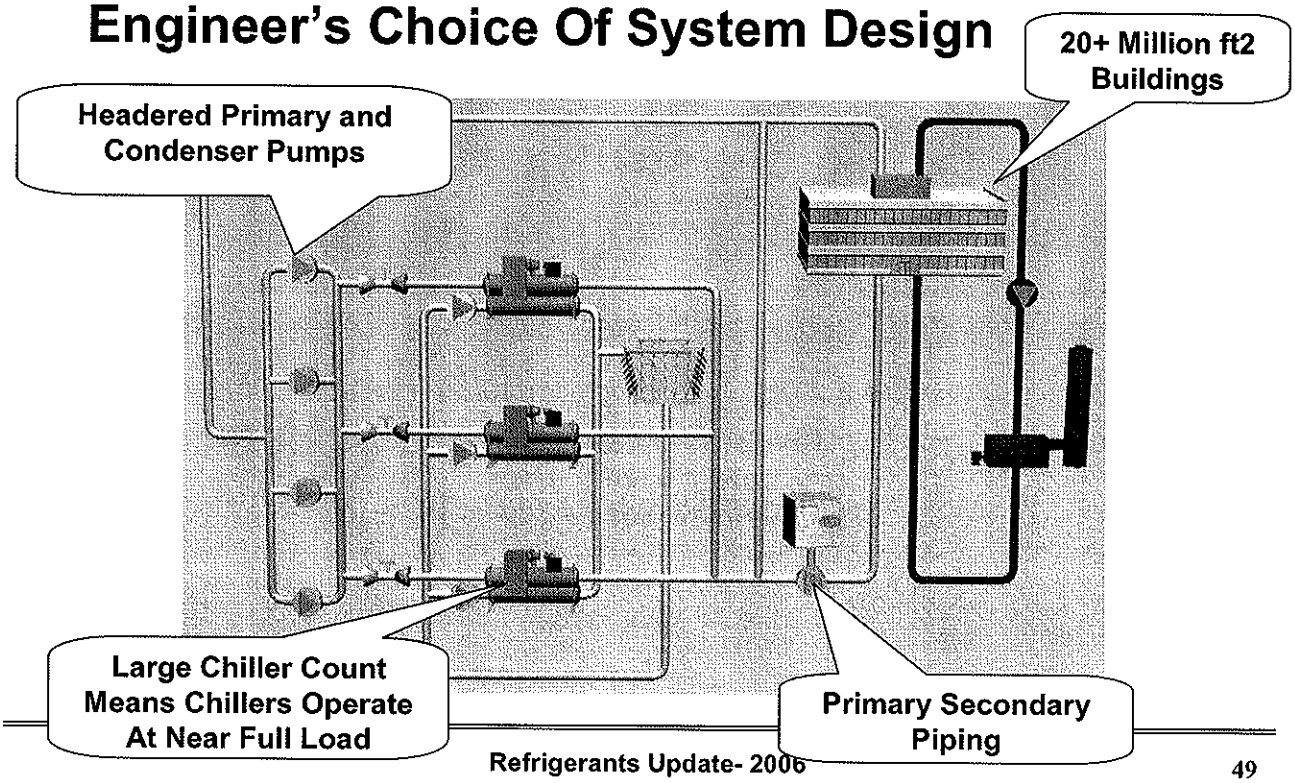
Big Plant Layout

- Large Chilled Water Range Is a Must
- Using 18°F ΔT
  - 48" Pipe - $500/ft
  - 5000 ft = $2.5M
  - 5800 hp @ 200' head
  - (It's Over 10,000 hp @ 10°F ΔT)
- Lower The Supply Water Temperature To Balance LMTD
  - No Pump Or Pipe Savings From Lower SWT
  - Fan Savings

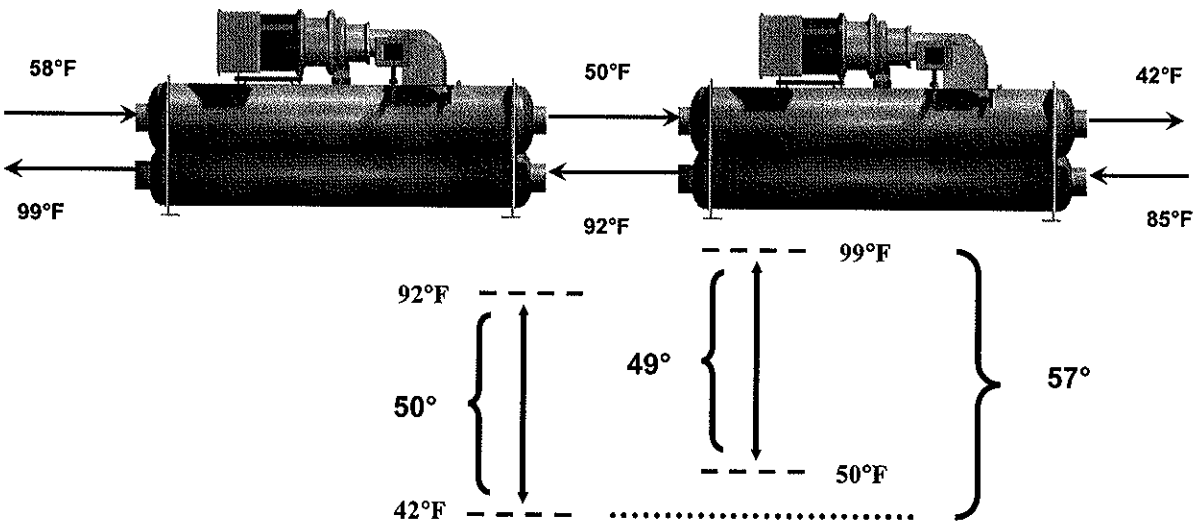


Big Plant Layout

Engineer's Choice Of System Design

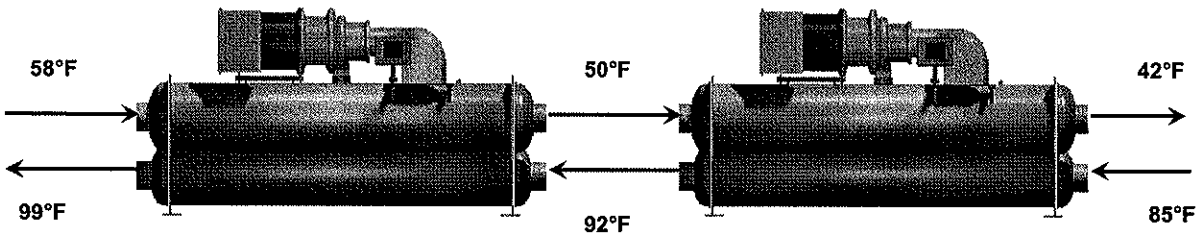


Series Counter-Flow Saves



Series Counter-Flow Saves
- 15% Compressor Work
- 33% Pump Work
- 33% Tower Work

Series Counter-Flow

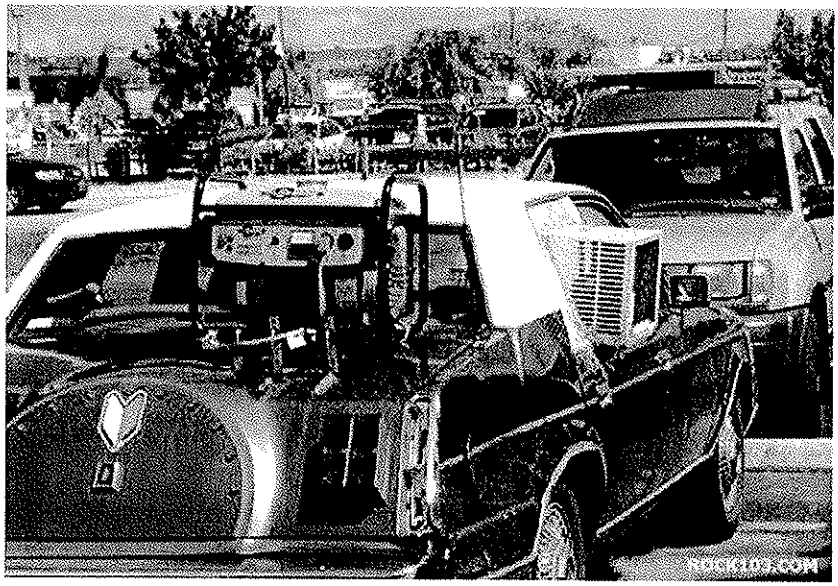


Saves { Chilled H2O Pump HP
Condenser H2O Pump HP
Cooling Tower Energy
Compressor Energy } $\approx \frac{15\%}{yr}$

McQuay Chiller Advantage Summary

- Most Tons In A Small Footprint
- Small Size Of Parts- Low Start-Up Stress
- Lower Sound Levels-Especially Part Load
- +30 Years Of Production/Performance
- Tubing Flexibility-Materials And Thicknesses
- HFC-134a Refrigerant Stability
- Compressor Redundancy On A Single Shell
- Serviceability
- EVERY Chiller Factory Run-Tested
- Long Life –Low Life Cycle Costing
- WCC Chiller Designed For Middle East
- Availability

Innovative Design Award

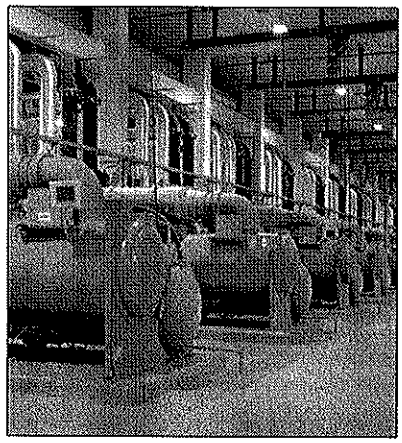


DAIKIN

McQuay International

OPTIMIZING CHILLERS & TOWERS

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Water-cooled centrifugal chiller plant.

Optimizing Chillers & Towers

By Hugh Crowther, P.Eng., Member ASHRAE and **James Furlong, P.E.,** Member ASHRAE

How a system operates at design conditions is not always a good indicator of its overall annual performance. This article emphasizes controls logic and design conditions and how they can be used to optimize chiller, cooling tower and condenser pump system performance throughout the year. A software model of hour-by-hour energy use for a typical office building in three cities demonstrates the effect of different cooling seasons and wet-bulb profiles. (Results are presented in energy use [kWh]. Actual dollar savings can be estimated by assuming a blended energy rate.)

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The model is based on a 160,000 ft² (14 864 m²), eight-story office building with a variable air volume (VAV) HVAC system that includes two centrifugal chillers in a variable primary flow configuration (*Figure 2*). The system uses induced draft-type cooling towers and constant flow condenser pumps that are dedicated to each chiller. Minimum turndown for the chillers is 50% and the bypass line opens below 25% of chiller plant capacity. Condenser pumps and cooling towers operate only when a chiller is on-line. The specific equipment size changes slightly with each location and is listed in the tables. *Figure 1* shows the annual cooling load profile for the three locations—Chicago, Las Vegas and Miami.

Design conditions are based around the Air Conditioning and Refrigeration Institute Standard 550/590 – *Water Chilling Packages Using the Vapor Compression Cycle* (54°F /44°F [12.2°C/6.7°C] chilled water, 85°F/95°F [29.4°C/35°C] condenser water) and the Cooling Tower Institute's (78°F [25.6°C] wet-bulb) test conditions, where possible, with exceptions noted. Changing these design con-

About the Authors

Hugh Crowther, P.Eng., is the vice president of product management & technology at McQuay International in Plymouth, Minn. He is a member of the ASHRAE Standards Committee. **James Furlong, P.E.,** is vice president comfort cooling marketing at Baltimore Aircoil Company in Baltimore.

ditions can have a noticeable effect on annual energy use as well as installed cost. The fixed design conditions allow comparisons of operating logic for each location.

Control Algorithms

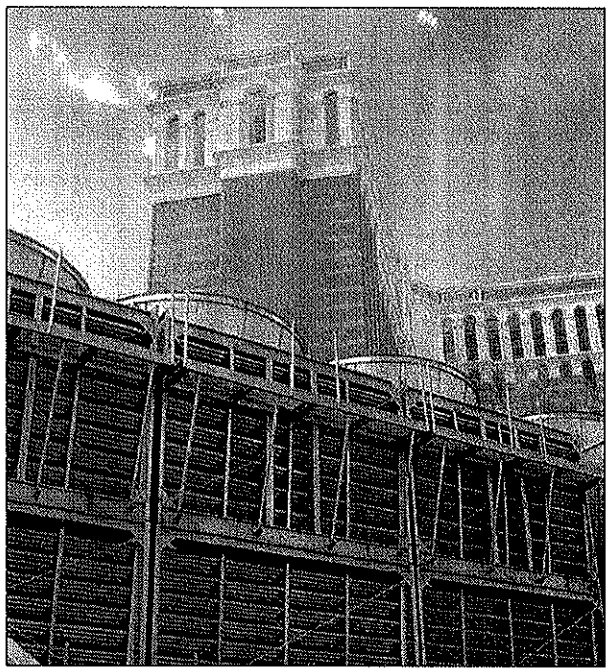
The first analysis is based on optimizing the chiller–condenser pump–tower. *Figure 3* shows the optimal point at which the chiller and tower's combined energy use is at the lowest. It is expected that the chiller plant operates at full load and at design parameters only at design conditions. During other periods (which is 99% of the time), we expect the chiller plant to operate at less than design load, which provides an opportunity for condenser water relief. Condenser water relief means that the condenser water temperature can be lowered because the cooling tower is no longer at 100% capacity and the ambient wet bulb is not at design condition. Lowering the condenser water temperature can provide significant chiller energy savings, as shown in *Figure 4*.

How the system will respond to non-design conditions can be analyzed by looking at the cooling tower fan modulation method and the control logic for condenser water supply temperature.

Figure 5 shows the fan power requirement for three fan modulation methods: on-off fans, two-speed (or pony) fan motors and variable frequency drives (VFD). In addition, three condenser water control methods have been considered: a fixed setpoint of 65°F (18.3°C), a mixed wet-bulb approach of 7°F (4°C) and optimized.

Fixed setpoint is an aquastat in the supply condenser water line that is set at the coldest water temperature you are willing to operate the chillers. Using 65°F (18.3°C) is a good choice because most chillers can operate at light loads with condenser water at this temperature. Colder water temperatures are possible, but the chiller type and load will become more important (e.g., as the chiller load is decreased, the water temperature will need to be raised). Fixed setpoint control logic effectively runs the cooling tower at full airflow until the minimum temperature is reached, and then modulates the fan. This optimizes the chiller, but not the system.

Fixed approach requires monitoring the ambient wet bulb. The cooling tower design approach is added to the current wet-bulb temperature to derive the setpoint. This logic assumes a correlation exists between reduced ambient wet bulb and building load. The condenser water temperature is lowered as the ambient wet bulb drops. As the chiller load drops (heat sent to the tower), the cooling tower fans will modulate and offer fan power savings. The process is complicated by how towers behave in reduced wet-bulb situations (see sidebar on cooling towers following this article).



Crossflow cooling tower installation.

The **optimized method** requires auto-adaptive controls. This control logic constantly adjusts the condenser water supply temperature to the value that uses the least amount of power. The controller measures the power requirement for the chiller and cooling tower. The condenser water temperature setpoint then is altered and the power consumption is checked again. If the total power consumption goes down, a similar adjustment is made and the total power is checked again.

Table 1 shows the specific HVAC system parameters at design conditions.

Table 2 shows the results of each control sequence for the Chicago office example.[2] Using the fixed setpoint and on-off fan control as a baseline, the analysis shows a 6.26% improvement when using VFD-controlled fans and optimized controls. This is substantial because the system components are the same for each example. Only the operation of those components at part load was changed. Referring to *Figure 3*, it is noted that the savings in tower fan work with optimized control far exceeded the increase in chiller work.

The next example is based on the same model building in Las Vegas — a dry hot climate with significantly more chiller operating hours. *Table 3* shows the system parameters. The design condenser water temperature was maintained at 85°F (29.4°C) to allow direct comparisons. The same control sequences were modeled, and the results are shown in *Table 4*.

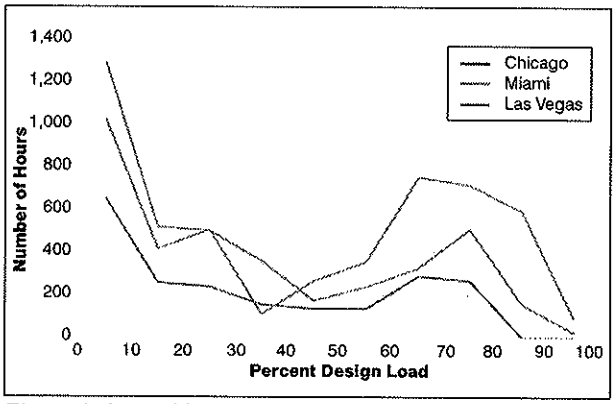


Figure 1: Annual load profiles.

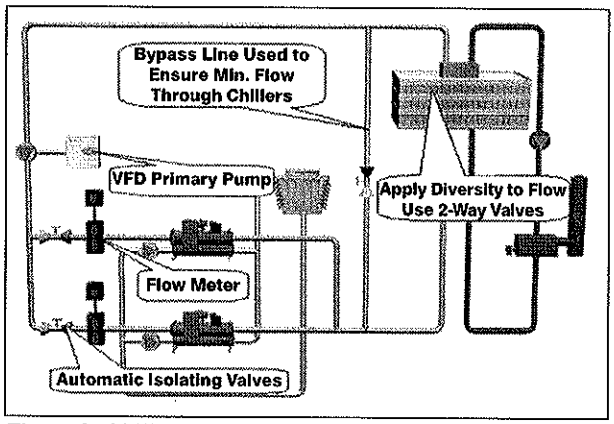


Figure 2: Chiller plant layout.

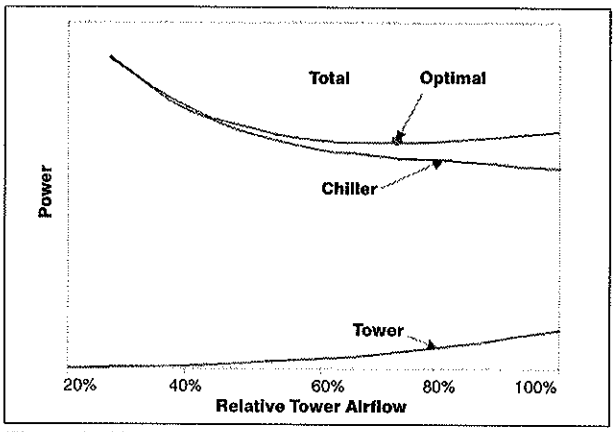


Figure 3: Chiller power vs. tower power.[1]

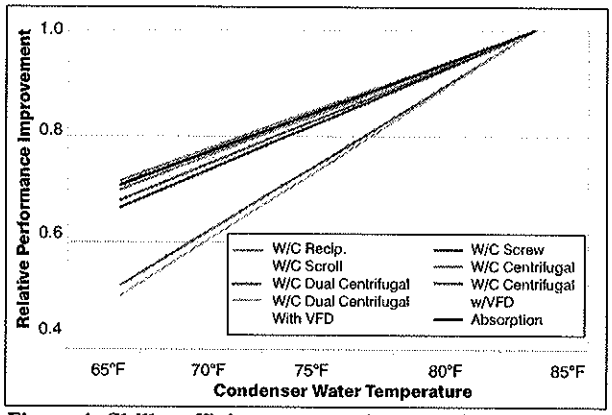


Figure 4: Chiller efficiency vs. condenser relief.

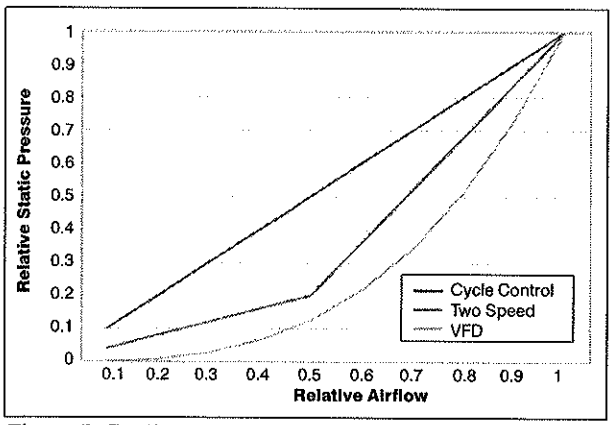


Figure 5: Cooling tower fan modulation.

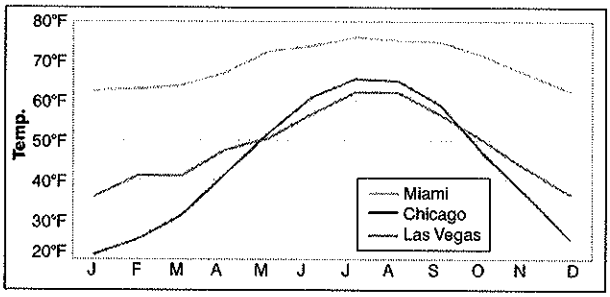


Figure 6: Annual wet-bulb profiles.

For the drier climate, the maximum relative savings from the baseline were only 4.66%. However, the actual energy savings went from 8,819 kWh/year for Chicago to 11,433 kWh/year for Las Vegas. The reasons for this difference are a combination of how cooling towers behave in dry climates (see cooling towers sidebar) and the difference in wet-bulb profiles between Chicago and Las Vegas.

The final example is based on the same building in Miami. Again, the location offers substantial chiller operating hours, but it also has a high wet-bulb with a flatter annual profile (*Figure 6*). In this case, the cooling towers were sized for a 5°F (2.8°C) approach due to the high design wet-bulb. *Table 5* shows the system parameters and *Table 6* shows the performance.

It is interesting that the location with the most humid climate enjoys the best enhancement by optimizing the chiller–tower relationship. From the baseline to the maximum savings, the performance improvement was more than 8%. In addi-

Example 1 – Chicago

Building Type	Office
HVAC System	VAV with Reheat
HVAC System Configuration	Variable Primary Flow Chiller Plant
Floor Area	160,000 ft² (14,864 m²)
Floors	8
Location	Chicago
Summer Design DB/WB	91°F/74°F (32.8°C/23.3°C)
Cooling Tower Rating – WB	78°F (25.6°C)
Design Cooling Load	400 tons (1400 kW)
Supply Fan Power	75.6 kW
Return Fan Power	22.7 kW
Number and Size of Chillers	2 at 200 tons (700 kW)
Chiller Performance	0.55 kW/ton
Chilled Water Temperatures	54°F/44°F (12.2°C/6.7°C)
Condenser Water Temperatures	85°F/95°F (29.4°C/35°C)
Number and Size of Primary Pumps	1 at 23 kW
Number and Size of Condenser Pumps	2 at 8 kW
Number and Size of Tower Fans	2 at 9 kW

Table 1: Chicago system parameters.

	Tower (kWh/yr)	Chiller (kWh/yr)	Total (kWh/yr)	Change From Baseline
65°F (18.3 °C) Setpoint				
On – Off	20,000	129,679	149,679	Baseline
Two Speed	19,271	129,679	148,950	0.49%
VFD	18,754	129,679	148,433	0.84%
Fixed Approach				
On – Off	17,537	130,475	148,012	1.13%
Two Speed	16,326	130,475	146,801	1.96%
VFD	15,317	130,475	145,792	2.67%
Optimized				
On – Off	15,649	132,176	147,825	1.25%
Two Speed	7,936	136,487	144,423	3.64%
VFD	5,892	134,968	140,860	6.26%

Table 2: Chicago chiller-tower performance.

Example 2 – Las Vegas

Building Type	Office
HVAC System	VAV with Reheat
HVAC System Configuration	Variable Primary Flow Chiller Plant
Floor Area	160,000 ft² (14 864 m²)
Floors	8
Location	Las Vegas
Summer Design DB/WB	108°F/66°F (42.2°C/18.9°C)
Cooling Tower Rating – WB	78°F (25.6°C)
Design Cooling Load	394 tons (1386 kW)
Supply Fan Power	79.1 kW
Return Fan Power	23.7 kW
Number and Size of Chillers	2 at 200 tons (700 kW)
Chiller Performance	0.55 kW/ton
Chilled Water Temperatures	54°F/44°F (12.2°C/6.7°C)
Condenser Water Temperatures	85°F/95°F (29.4°C/35°C)
Number and Size of Primary Pumps	1 at 22 kW
Number and Size of Condenser Pumps	2 at 8 kW
Number and Size of Tower Fans	2 at 9 kW

Table 3: Las Vegas system parameters.

	Tower (kWh/yr)	Chiller (kWh/yr)	Total (kWh/yr)	Change From Baseline
65°F (18.3 °C) Setpoint				
On – Off	30,212	226,514	256,726	Baseline
Two Speed	27,207	226,514	253,721	1.18%
VFD	25,021	226,514	251,535	2.06%
Fixed Approach				
On – Off	29,382	226,960	256,342	0.15%
Two Speed	26,090	226,960	253,050	1.45%
VFD	23,634	226,960	250,594	2.45%
Optimized				
On – Off	28,443	222,814	251,257	2.18%
Two Speed	16,831	234,103	250,934	2.31%
VFD	11,384	233,909	245,293	4.66%

Table 4: Las Vegas chiller-tower performance.

tion, the absolute savings (more than 40,000 kWh/yr) were the greatest due to the extended cooling season. The fixed setpoint systems all performed the same, regardless of fan control logic, because the high local wet-bulb never allows the fan to modulate. The control systems that allow fan modulation offer very good savings (remember 80% airflow results in 50% power reduction).

What about Design Conditions?

The previous three examples were based on the typical operating conditions specified in ARI 550/590 and the CTI Standard 201-2002, *Cooling Tower Test.* Moving away from these criteria can result in further system improvement.

One consideration is to "oversize" the cooling tower and lower the design condenser water temperature. This will increase the size and cost of the cooling tower, but it will have an advantageous effect on the chiller. A good place to start is to lower the design cooling tower approach by 2°F (1.1°C). The Cooling Tower Institute (CTI) Standard 201 program will certify cooling towers down to 5°F (2.8°C) approach.

Table 7 shows the Chicago example but with an oversized cooling tower capable of producing 83°F (28°C) water with a 5°F (2.8°C) approach. The tower fan motors increased from 8 to 11 kW. It is important to note that an oversized cooling tower actually uses more power annually with poor tower fan control. However, the same oversized cooling tower can produce about a 3% improvement with good tower fan control.

What About the Condenser Pump?

In the previous examples, we held the condenser pump constant and sized at 3 gpm/ton (0.0538 L/s per kW). Varying the flow of the condenser pump during part-load conditions also can lower the operating cost of the system. However, varying the flow based on load presents challenges. First, the cooling towers must be selected to operate with reduced flow. Also, the control sequence can be difficult.

Optimizing the chiller and tower to use the least amount of power and holding the condenser pump constant is easier to manage because it offers two degrees of freedom (the chiller and tower power) and one parameter (the condenser water sup-

ply temperature). Adding the third parameter (condenser flow) can be a difficult problem to solve. Thomas Hartman, P.E., Member ASHRAE,[3] has done some interesting work in this area. The author has experience using a stepped flow reduction approach, and for these projects, the condenser flow was reduced to 80% (a 50% improvement in condenser pump work) with little adverse affect on the cooling tower. The deciding factor in lowering the flow in each case was based on circumstances where one chiller was off in a series chiller plant, or where one compressor was off in a dual compressor chiller.

Changing the design conditions can also affect the system performance. Increasing the temperature range should be given very careful consideration. While this will reduce the flow and pump power, it will adversely affect the chiller work.[4,5] In many applications, the overall operating cost will rise, even with a 1°F (0.8°C) increase in range, depending on the relative size of the condenser pumps to the chillers. In applications with high pump head, it may make sense to increase the range.

Conclusions

Optimizing the chiller–tower condenser–pump system can be managed in two steps. First, optimize the design conditions while taking into account your local design weather and project specific details. Second, optimize the control logic to take full advantage on the capital equipment in the chiller plant. Changing from very basic to advanced controls can make a 5% to 8% improvement in chiller–tower condenser pump performance without changing the capital equipment.

The payback for integrating more effective controls will depend on the actual plant size and the number of operating hours. The cost of the additional controls remains relatively constant regardless of equipment size, which implies that larger plants will enjoy faster paybacks. Additionally, more operating hours (health care vs. office space) will improve the payback. As a benchmarking tool, designers can use the percent savings times the estimated operating cost of a basic system to estimate the annual savings. The cost of the additional controls can be used to estimate the payback for the project.

This article did not investigate the affect of changing the condenser water temperature range on the savings with more advanced cooling tower controls or different climatic conditions. Changing the temperature range affects many things and this would be a good future article.

Bibliography

Braun, J.E. and G.T. Diderrich. 1990. "Near-Optimal Control of Cooling Towers for Chilled Water Systems." *ASHRAE Transactions* SL-90-13-3.

Notes

1. Braun, J.E., and G.T. Diderrich. 1990. *Near-Optimal Control of Cooling Towers For Chilled Water Systems. ASHRAE Transactions.*
2. Annual energy analysis was performed using McQuay Energy Analyzer 4.0.

Example 3 – Miami

Building Type	Office
HVAC System	VAV with Reheat
HVAC System Configuration	Variable Primary Flow Chiller Plant
Floor Area	160,000 ft² (14,864 m²)
Floors	8
Location	Miami
Summer Design DB/WB	91°F/77°F (32.8°C/25°C)
Cooling Tower Rating – WB	78°F (25.6°C)
Design Cooling Load	426 tons (1498 kW)
Supply Fan Power	76.4 kW
Return Fan Power	23.7 kW
Number and Size of Chillers	2 at 213.7 tons (751.6 kW)
Chiller Performance	0.55 kW/ton
Chilled Water Temperatures	54°F/44°F (12.2°C/6.7°C)
Condenser Water Temperatures	85°F/95°F (29.4°C/35°C)
Number and Size of Primary Pumps	1 at 24 kW
Number and Size of Condenser Pumps	2 at 9 kW
Number and Size of Tower Fans	2 at 10 kW

Table 5: Miami system parameters.

	Tower (kWh/yr)	Chiller (kWh/yr)	Total (kWh/yr)	Change From Baseline
65°F (18.3 °C) Setpoint				
On – Off	75,354	440,145	515,499	Baseline
Two Speed	75,130	440,145	515,275	0.04%
VFD	74,972	440,145	515,117	0.07%
Fixed Approach				
On – Off	63,370	442,873	506,243	1.83%
Two Speed	60,055	442,873	502,928	2.50%
VFD	56,841	442,873	499,714	3.16%
Optimized				
On – Off	43,026	457,956	500,982	2.90%
Two Speed	16,564	466,681	483,245	6.67%
VFD	14,815	460,094	474,909	8.55%

Table 6: Miami chiller-tower performance.

	Tower (kWh/yr)	Chiller (kWh/yr)	Total (kWh/yr)	Change From Baseline
65°F (18.3 °C) Setpoint				
On – Off	25,268	125,990	151,258	Baseline
Two Speed	24,050	125,990	150,040	0.81%
VFD	23,294	125,990	149,284	1.32%
Fixed Approach				
On – Off	22,491	126,408	148,899	1.58%
Two Speed	20,839	126,408	147,247	2.72%
VFD	19,606	126,408	146,014	3.59%
Optimized				
On – Off	14,725	131,590	146,315	3.38%
Two Speed	5,608	133,824	139,432	8.48%
VFD	4,503	132,079	136,582	10.75%

Table 7: Chicago chiller-tower performance with oversized tower.

3. Hartman, T. 2001. "All-variable speed centrifugal chiller plants." *ASHRAE Journal* 43(9).
4. Kirsner, W. 1996. "Three gpm/ton condenser water rate: does it waste energy?" *ASHRAE Journal* 38(2).
5 Crowther, H. 2002. "Why change the chilled water range?" Minneapolis: McQuay Engineering Solutions.

"Cooling Towers: Lower Flow Systems" sidebar follows.

Cooling Towers: Lower Flow Systems

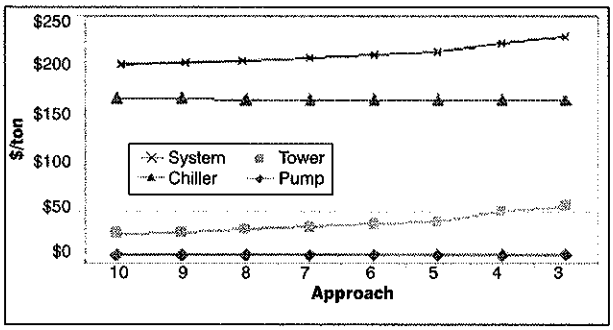


Figure 1: Lowest system first cost 78 wet bulb, 3 gpm/ton.

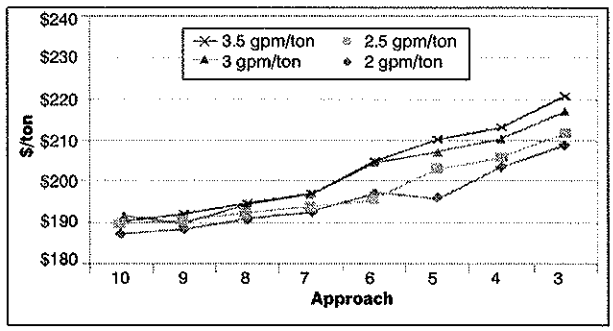


Figure 2: Lowest system first cost, 66°F wet bulb.

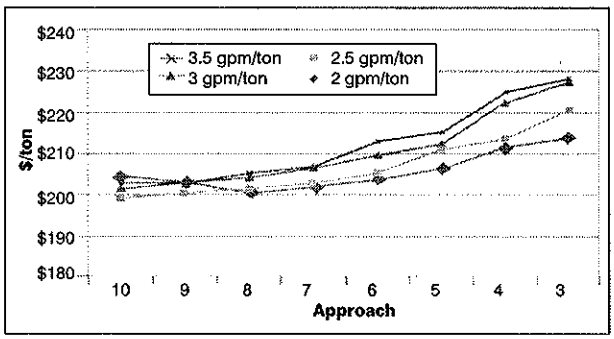


Figure 3: Lowest system first cost, 78°F wet bulb, variable gpm/ton.

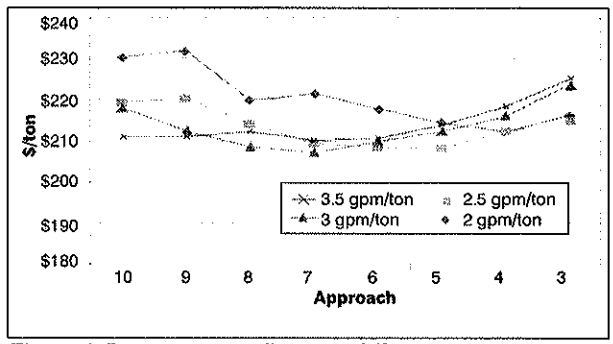


Figure 4: Lowest system first cost, 84°F wet bulb.

For more than 50 years, electric motor-driven, water-cooled chiller systems commonly have been designed around entering condenser water temperatures of 85°F (29.4°C) and a nominal condenser water flow of 3 gpm/ton (0.0538 L/s per kW). For reciprocating-type water-cooled chillers rated at 0.90 kW/ton, the 3 gpm/ton (0.0538 L/s per kW) nominal flow rate yields a 10°F (5.5°C) range, which has long been the ARI standard rating condition for water-cooled chillers.

In recent years, considerable debate has occurred as to the merits of designing around lower nominal condenser water flow rates to improve system life-cycle costs. Those who believe in using 2 gpm/ton (0.0359 L/s per kW) claim that there is a first-cost advantage to the lower flow system. This is derived from the lower costs associated with smaller pumps, smaller pipe, and a smaller cooling tower that more than offset any increases in cost associated with the additional heat transfer surface that may be required in the chiller. They also claim that the lower flow system can deliver improved operating costs because the reduced kW required by smaller pumps and cooling tower fan motors more than offsets the increased power required by the chiller to overcome the greater lift imposed by higher condensing temperatures.

A discussion of condenser water flow-rate optimization with respect to chiller system first cost, operating cost, or life-cycle cost cannot reasonably occur unless two additional parameters are taken into consideration: approach and design wet bulb. Approach, or the temperature difference between the water leaving the cooling tower and the ambient wet bulb, has a more significant influence on cooling tower size and energy consumption than any other parameter affecting the cooling tower. Traditionally, the HVAC industry has designed around approaches of 7°F (3.8°C) or greater for two reasons. First, the majority of geographies are associated with design wet bulbs of 78°F (25.6°C) and below, making the attainment of "standard" 85°F (29.4°C) condenser water possible. Second, an industry mindset prevails that accepts 85°F (29.4°C) condenser water temperatures as "ideal" from an overall system energy standpoint. Whether this mindset is based on reality gives rise to the following question: what nominal condenser water flow rate and approach is best from a first-cost and full-load energy perspective at any given wet bulb?

A study was recently completed in an effort to answer this question, using actual first cost and full load performance data from a variety of chiller, cooling tower, and pump manufacturers for a nominal 500 ton (1759 kW) water-cooled, centrifugal chiller system. The manufacturers involved supplied one set of equipment selections and pricing data aimed at optimizing first cost at the expense of efficiency (low cost but also inefficient) and a separate set aimed at optimizing full load energy consumption at the expense of first cost (efficient but expensive). Chiller evaporator conditions were fixed at 54°F (12.2°C) entering and 44°F (6.7°C) leaving temperatures with a maximum pressure drop of 20 ft w.c. (59.8 kPa). Chiller condensers were all limited to a maximum pressure drop of 20 ft w.c. (59.8 kPa) and condenser pumps were sized at a fixed system head of 60 ft w.c. (179.4 kPa). Piping costs and system pressure drops were not taken into consideration.

The study considered a range of nominal condenser flows from 2.0 to 3.5 gpm/ton (0.036 L/s per kW to 0.063 L/s per kW) in 0.5 gpm/ton (0.009 L/s per kW) increments, cooling tower approaches from 3°F (1.6°C) to 10°F (5.5°C) in 1°F (0.5°C) increments, and wet-bulb conditions from 66°F (18.9°C) to 84°F (28.9°C) in 6°F (3.3°C) increments.

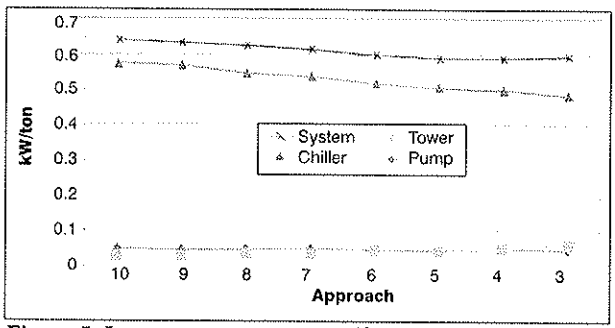


Figure 5: Lowest system energy, 78°F wet bulb, 3 gpm/ton.

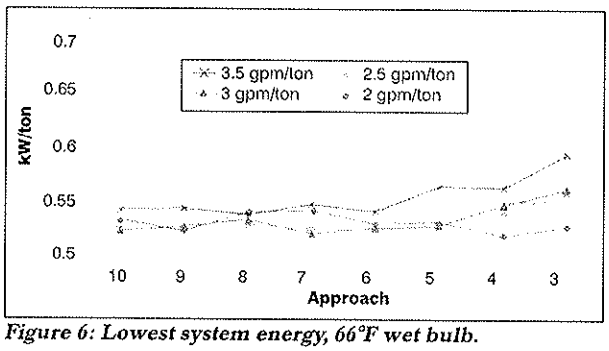


Figure 6: Lowest system energy, 66°F wet bulb.

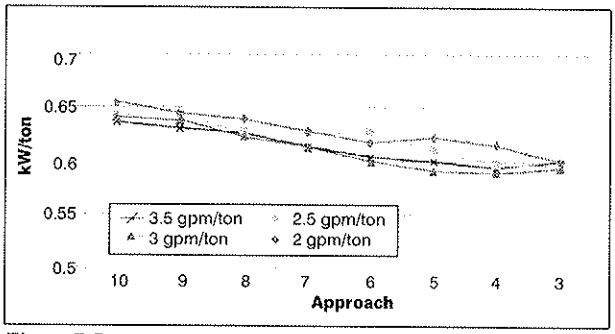


Figure 7: Lowest system energy, 78°F wet bulb, variable gpm/ton.

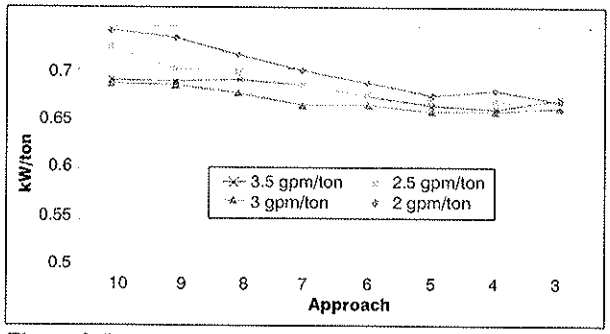


Figure 8: Lowest system energy, 84°F wet bulb.

Optimizing First Costs

Equipment was selected on the basis of meeting full load capacity at lowest possible first cost. Component equipment first costs were plotted as a function of approach for all combinations of nominal flow rate and wet-bulb temperature. Sum of all component costs was taken as the system cost (*Figure 1*).

Removing the component detail allows the system costs to be plotted as a function of nominal flow rate at a given wet bulb. Plots at 66°F (18.9°C), 78°F (25.6°C), and 84°F (28.9°C) wet bulb are shown in *Figures 2* through *4*. A review of these plots will reveal that lower condenser water flows (2.0 to 2.5 gpm/ton [0.0359 to 0.0449 L/s per kW]) and higher approaches (8°F to 10°F [4.4°C to 5.5°C]) are a viable means to optimize system first costs at all but the highest wet-bulb environments (84°F [28.9°C]). The first cost advantage of the lower flow systems would likely have been more pronounced if the material cost savings of smaller diameter condenser water piping had been considered.

Optimizing Full Load Energy Consumption

Equipment was selected on the basis of meeting full load capacity at the lowest possible energy consumption. In the case of the chiller selections and cooling tower selections, this translated to equipment with significantly greater heat transfer surface than the equipment selected based on optimizing first costs. The full load energy consumption of component equipment was plotted as a function of approach for all combinations of nominal flow rates and wet-bulb temperatures. Full-load system energy consumption is taken as the sum of the full load energy consumption of all system components (*Figure 5*). Plots of full-load energy consumption as a function of nominal flow at 66°F (18.9°C), 78°F (25.6°C), and 84°F (28.9°C) are shown in *Figures 6* through *8*.

A summary of the optimized energy cost plots is revealing. The higher the ambient wet bulb, the higher the required nominal condenser water flow rate and the tighter the approach needs to be to optimize system energy cost. Only at the lowest wet-bulb condition considered, 66°F (18.9°C), does a 2.0 or 2.5 gpm/ton (0.0359 or 0.0449 L/s per kW) system show an energy advantage. At 78°F (25.6°C) and 84°F (28.9°C) wet bulb, the optimal energy balance occurs at 3 gpm/ton (0.0538 L/s per kW) with a 4°F (2.2°C) approach.

The advantage of a tighter approach in the higher wet-bulb environments is significant. At 78°F (25.6°C) wet bulb, the optimum system energy is achieved at 3 gpm/ton (0.0538 L/s per kW) with a 4°F (2.2°C) approach. Had a "typical" 3 gpm/ton (0.0538 L/s per kW), 7°F (3.8°C) approach been specified on this system, a system energy penalty of more than 4% would have occurred. It is also interesting to note that approach and condenser flow have little impact on system energy in low wet-bulb environments. At 66°F (18.9°C) wet bulb, system energy is largely unchanged as the approach is tightened from 10°F (5.5°C) to 5°F (2.7°C). The energy saved by the chiller when operating with 71°F (21.7°C) vs. 76°F (24.4°C) entering condenser water is entirely offset by the additional energy consumed by the cooling tower fan to produce the colder condenser water. In this case, it would make no sense to invest in larger size cooling tower required to generate the 5°F (2.7°C) approach.

A number of conclusions can be drawn from the study:

- Low-flow (2 gpm/ton [0.0359 L/s per kW]) condenser water systems generally have first cost advantages over higher flow (3 gpm/ton [0.0538 L/s per kW]) systems in all but the highest wet-bulb environments.
- High-flow (3 gpm/ton [0.0538 L/s per kW]) condenser water systems generally have full load energy advantages over lower flow (2 gpm/ton [0.0359 L/s per kW]) systems in all but the lowest wet-bulb environments
- Approaches in the range of 4°F (2.2°C) to 5°F (2.7°C) offer significant full load system energy advantages over "traditional" 7°F (3.8°C) approaches in environments with higher wet bulbs (78°F to 84°F [25.6°C to 28.9°C]).●

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3 SIMPLE APPROACHES TO ENERGY EFFICIENCY

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3 Optimal Air, Energy Recovery, Geothermal Simple Approaches To Energy Efficiency

By Carol Marriott, P.Eng., Member ASHRAE

Many building owners are choosing sustainable design because the economics make more sense now. For example, rising energy costs are reducing the payback period for capital improvements that improve energy performance. However, in contrast to the wealth of promotional information for why sustainable building practices should be used, surprisingly little application information is available on sustainable design.

This article focuses on one aspect of sustainable design—changing traditional design parameters to improve the energy performance of an HVAC system. It provides several examples of design parameters to improve HVAC system efficiency with solid economic payback, or in some cases, immediate payback.

To investigate the amount of savings that can occur from making changes to an HVAC system, one must start with a benchmark system to provide a performance baseline against which the alternative system can be measured. For the examples provided in this article, ANSI/ASHRAE/IESNA Standard 90.1-2004, *Energy Standard for Buildings Except Low-Rise Residential Buildings*, Appendix G, Performance Rating Method, was used to provide the benchmark HVAC system based on building size. These benchmarks are summarized in *Table 1* for nonresidential spaces.

The energy rates[1] used in all three analyses are summarized in *Table 2*. To perform the energy analysis, a private energy analysis tool using reduced weather data gives a ballpark idea of energy savings. Appendix G details specific energy analysis program requirements including 8,760 hour weather data. Appendix G also is being used as the benchmark for the Energy and Atmosphere Credit 1 in Leadership in Energy and Environmental Design® (LEED)-NC v2.2.

Example 1: Optimal Air Systems

To understand the synergies in the system, *Figure 1* shows average annual energy consumption of a building in Chicago using VAV with reheat with water-cooled chillers. The pie chart will change depending on the location of the building and the HVAC system used. Interestingly, fan energy is the greatest energy consumer in the mechanical system because the fans are used the entire

About the Author

Carol Marriott, P.Eng. is applications manager for McQuay International in Minneapolis.

time the building is occupied. From this you could draw a conclusion that money spent for high-efficiency chillers might be better spent on reducing fan energy expenses to generate greater returns.

Pursuing these airside savings can challenge the most sacred HVAC design principle—supplying 55°F (13°C) air to the space. Cooling supply air to 55°F (13°C) generally provides the required humidity ratio to maintain space conditions at 75°F (24°C) and 50% relative humidity. However, cooling supply air below 55°F (13°C) offers the potential for significant capital or energy cost savings in many applications. Chapter 26 of the *2005 ASHRAE Handbook—Fundamentals* addresses insulation to avoid condensation based on relative humidity levels in the space.

As the supply air temperature is reduced, the supply air volume decreases proportionally. That is, a 10% increase in supply air ΔT (space setpoint minus the supply air temperature) will result in a 10% drop in required supply air volume. Any resulting energy savings from the lower airflow can be taken as a reduction in annual energy consumed or as capital cost savings:

- Sizing the air handler and ductwork as if the system is providing 55°F (13°C) supply air results in energy savings during the life of the equipment. As the ΔT of the system increases, the amount of airflow required to provide the same cooling effect is reduced. This lowers the static pressure and allows smaller fan motors to be installed. The decreased fan size also reduces fan noise, resulting in a quieter system. This approach is ideal for existing buildings where the cost to remove and replace existing ductwork may be prohibitive.
- Reducing the duct, air handler and fan motor sizes to match the lower supply air volume can result in capital cost savings and, in the case of indoor air-handling units, provide more leasable space. This approach is ideal for new buildings where there are no preexisting conditions affecting the design (i.e., ductwork).

The practice of reducing supply air temperature is common in grocery stores to improve humidity control, and it is gaining popularity in comfort cooling applications. The key design parameter is to identify the optimal air or balance point. This is the lowest supply air temperature that can be used without increasing the annual operating cost of the building. While it is typically 48°F to 52°F (9°C to 11°C), every building is different and annual energy analysis is required to determine this point. The amount of time and effort on finding the balance point depends on the complexity of the project.

The penalty for achieving these reductions is that the colder supply air temperature requires more refrigeration work and reduces the number of hours in a year where economizer operation can be used. For example, lowering the supply air setpoint from 55°F to 50°F (13°C to 10°C) removes the opportunity to cool the building with outdoor air when the ambient dry bulb is between 55°F and 50°F (13°C and 10°C). In Chicago, 601 hours of the year are between 50°F and 55°F (10°C and 13°C), or roughly 7% of the hours when a HVAC system may be operating. With integrated economizers, some cooling effect can be gained, but supplemental mechanical cooling is required.

In addition, the primary system must be capable of providing the low supply air temperature. This is not an issue with chiller and air-handling systems, but it does require that packaged rooftop and other unitary systems offer some flexibility in DX coil and refrigeration component selections to avoid oversizing the unit to provide the lower supply air temperature.

Table 3 shows the effect of lowering the supply air temperature to 50°F (10°C) for a one-story, 100,000 ft^2 (9290 m^2) retail building. To illustrate the effects of changing the design supply temperature from the traditional 55°F to 50°F (13°C to 10°C), a VAV packaged rooftop system that meets the minimum requirements of Standard 90.1-2004 was selected as the baseline system from Appendix G (Table 1). In this comparison, the duct size was maintained to keep capital costs flat and investigate energy savings of the lower airflow rate and reduced static pressure. Three locations throughout the U.S. were chosen to illustrate the results in different climates. The 50°F (10°C) supply air temperature was selected for illustration purposes only and may or may not represent the optimal air or balance point for each application.

Note that the building energy cost savings ranged from 4.8% to 6.1%. These are significant energy cost savings with little or no impact on design work or capital costs. With a little more effort, the system could be further optimized to deliver even greater energy cost savings:

However, in contrast to the wealth of promotional information for why sustainable building practices should be used, surprisingly little application information is available on sustainable design.

- The lower supply air temperature also lowers the relative humidity in the space. This should allow the room setpoint to be raised while still maintaining acceptable conditions as detailed in ANSI/ASHRAE Standard 55-2004, *Thermal Environmental Conditions for Human Occupancy.*
- The airside can be specified with a blow-through (fan before coil) vs. a draw-through (fan after coil) configuration to eliminate the additional airflow required to compensate for fan heat in the airstream (typically a 2°F to 3°F [1°C to 1.5°C] temperature rise).
- Specifying more efficient fans and compressors may raise capital costs, but the payback could be very favorable.

Each of these strategies can be justified or cast aside using the same energy analysis shown in this example.

Example 2: Recovering Energy from Condenser Water

A huge amount of heat energy is sent to the atmosphere through the cooling towers. In addition to every Btu collected in the building, approximately 25% more heat energy from compressor and accessory work is sent to the tower. Typically, chillers send 95°F (35°C) water to the cooling tower. By raising this design parameter to between 105°F (40°C) and 140°F (60°C) through one of two condenser heat recovery options, the increased chiller lift penalizes the compressor efficiency of the cooling system but benefits the heating system. Overall, the total HVAC system may perform more economically and show a solid economic payback for the additional capital cost.[2]

Condenser heat recovery can be used for building reheat or to preheat domestic hot water. Standard 90.1-2004 has minimum requirements for heat recovery:

***6.5.6.2 Heat Recovery for Service Water Heating**—Condenser heat recovery systems shall be installed for heating or preheating of service hot water provided all of the following are true:*

a. The facility operates 24 hours day.

b. The total installed heat rejection capacity of the water-cooled system exceeds 6,000,000 Btu/h (1700 kW) of heat of rejection.

c. The design service water heating load exceeds 1,000,000 Btu/h (300 kW).

The required heat recovery system shall have the capacity to provide the smaller of:

a. 60% of the peak heat rejection load at design conditions, or

b. Preheat of the peak service hot water draw to 85°F (29°C).

Exceptions to 6.5.6.2

a. Facilities that employ condenser heat recovery for space heating with a heat recovery design exceeding 30% of the peak water-cooled condenser load at design conditions.

b. Facilities that provide 60% of their service water heating from site solar or site recovered energy, or from other sources.

In addition, Standard 90.1 exempts the requirement for economizers, 6.5.1 (d) if the HVAC system includes a condenser heat recovery system that is required by 6.5.6.2. Standard 90.1 also allows simultaneous heating and cooling for comfort or dehumidification purposes if 75% of the reheat energy is site-recovered, such as condenser heat recovery

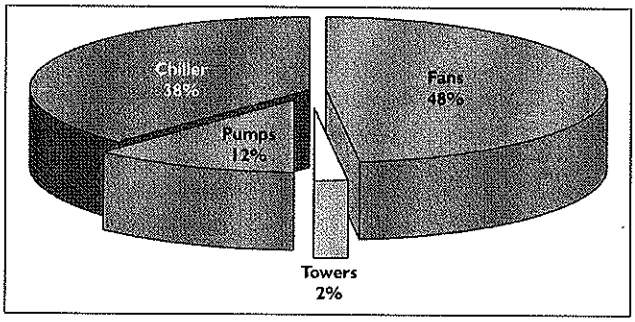


Figure 1: Average office building annual energy consumption.

Building Type*		Fossil Fuel & Purchased Heat	Electric And Other
Number of Floors	Area		
3 Or Less	<75,000 ft²	Packaged Rooftop Constant Volume	Packaged Rooftop Heat Pump Constant Volume
4 or 5	<75,000 ft²	Packaged Rooftop VAV with Reheat	Packaged Rooftop VAV With Parallel Fan Powered Boxes
5 or Less	75,000 ft² to 150,000 ft²		
More Than 5	Any	VAV With Reheat & Chillers	VAV With Parallel Fan Powered Boxes And Chillers

* Nonresidential buildings only are listed here.

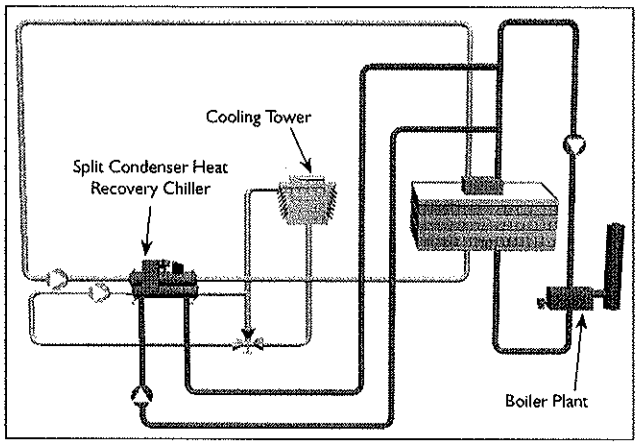
Table 1: Baseline system from Standard 90.1-2004's Appendix G.

Location	Electricity ($/kWh)	Natural Gas $/Therm	Natural Gas $/GJ
Chicago	$0.0754	$1.11	$10.52
Miami	$0.0761	$1.33	$12.61
Philadelphia	$0.0851	$1.20	$11.37
Minneapolis	$0.0631	$1.01	$9.57
Helena	$0.0742	$1.04	$9.86
Denver	$0.0689	$0.92	$8.72

Table 2: Average energy prices used in comparisons.

	Baseline Energy Cost	Performance System Energy Cost	Percent Energy Savings
Chicago	$307,983	$289,259	6.1%
Miami	$228,482	$217,394	4.9%
Philadelphia	$261,108	$248,544	4.8%

Table 3 Energy cost comparison: 50°F and 55°F SAT.



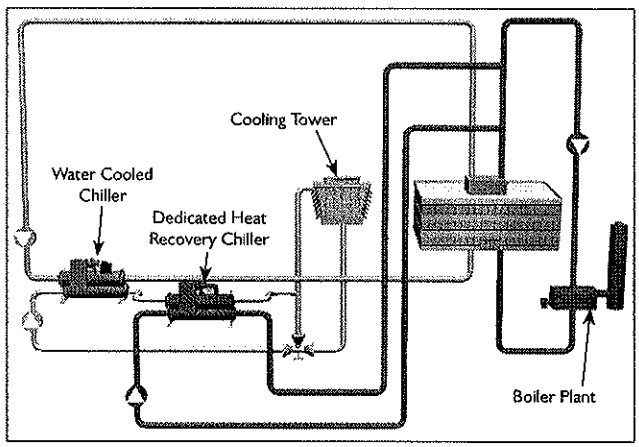


Figure 2 (left): Split condenser heat recovery chiller system. Figure 3 (right): Cascaded system with dedicated heat recovery chiller.

systems. The two main building types likely to meet the condenser recovery requirement are health-care and hospitality facilities. Condenser heat recovery can be performed using a heat recovery chiller, or a cascaded system with a dedicated heat recovery chiller.

A split condenser heat recovery chiller, shown in *Figure 2*, changes the typical design parameters of chiller operation on the condenser side from 85°F/95°F (39°C/35°C) to 95°F/105°F (35°C/40°C). This increases the chiller lift during heat recovery mode and results in a penalty on chiller efficiency. The chiller either operates in cooling tower mode, rejecting condenser heat outdoors, or in heat recovery mode, where the condenser produces up to 105°F (40°C) water for building use. This system is limited by the lift capabilities of the chiller. At 105°F (40°C), reheat coils may have to be resized, increasing capital costs and pressure drop compared to a coil sized for a design reheat temperature of 140°F (60°C).

Cascading the system and having a dedicated heat recovery chiller, as shown in *Figure 3*, allows the source chiller to run at typical comfort cooling conditions. Condenser water is sent to a cooling tower, or through the evaporator of the dedicated heat recovery chiller. The dedicated heat recovery chiller has similar lift requirements as a chiller at standard conditions, so it is able to generate up to 140°F (60°C) water for building use. An advantage of this system is that the dedicated heat recovery chiller can be bypassed in non-heat recovery mode. A disadvantage is the added capital cost of another chiller.

Recovered Energy for Building Heat

Constant volume with reheat and four-pipe fan coil systems are good candidates for condenser energy recovery because they share a common trait of simultaneous heating and cooling. VAV systems may be a good candidate, depending upon the minimum turndown on the VAV boxes. If the minimum turndown is 50% or more, as is the case with many health-care applications, it may make sense.

Figure 4 shows a typical annual heating and cooling load profile for a constant volume with reheat system commonly used in health-care facilities. This figure clearly demonstrates the Golden Rule of energy recovery "You must have an energy source at the same time you have an energy need!"

The ideal size for the energy recovery equipment is the point where the instantaneous heat source meets the instantaneous heating requirement. This is where the heating and cooling lines cross in *Figure 4*. Energy can be recovered in the shaded area under both curves.

The only way to know for certain if condenser energy recovery for heating makes sense is to run an annual energy analysis. For this example, a three-story, 480,000 ft^2 (4400 m^2) acute care hospital with a 1,600 ton (455 kW) chiller plant and a 20,000 kBtu/h (2000 MJ/h) boiler plant is considered. The HVAC system is a combination of VAV and constant volume with reheat and chillers providing 400,000 cfm (190 000 L/s) of supply air and 192,000 cfm (90 600 L/s) of ventilation air.

The baseline system is comprised of two 800 ton (225 kW) centrifugal chill-

Advertisement formerly in this space.

ers. The first energy recovery option includes two 800 ton (225 kW) centrifugal chillers and a 460 ton (130 kW) dedicated heat recovery chiller (4.4 COP). The dedicated heat recovery chiller is designed to provide 140°F (60°C) hot water. Additional design requirements for this system include increasing the source chiller head by 20 ft (6 m) to accommodate the added pressure drop of the energy recovery chiller. A hot water tertiary pump has also been included.

The second heat recovery option includes a 1,200 ton (340 kW) centrifugal chiller and a 375 ton (100 kW) split condenser heat recovery chiller (0.73 kW/ton, 4.80 COP) optimized for the heat recovery load and supplying 105°F (40°C) hot water. A design consideration for this system includes adding deeper heating coils to account for the lower grade heat from the split condenser heat recovery chiller. Static pressure has been increased by 0.20 in. (50 Pa) to account for the deeper coils. In addition, a hot water tertiary pump has been added.

As shown in *Table 4,* both energy recovery options saved between 8% to 10.7% on building energy. Further analysis could be done to achieve more energy savings such as using variable flow, changing condenser water temperature, changing chilled water temperature range, or changing other chiller system design parameters.

Example 3: Geothermal Heat Pump Systems

The final example relies on selecting an HVAC system that is more efficient to begin with and optimizing it to achieve capital cost savings and additional energy savings. Geothermal water source heat pump systems are some of the most energy efficient systems for applications such as offices, schools, medical facilities, dormitories, condominiums, and recreational facilities.

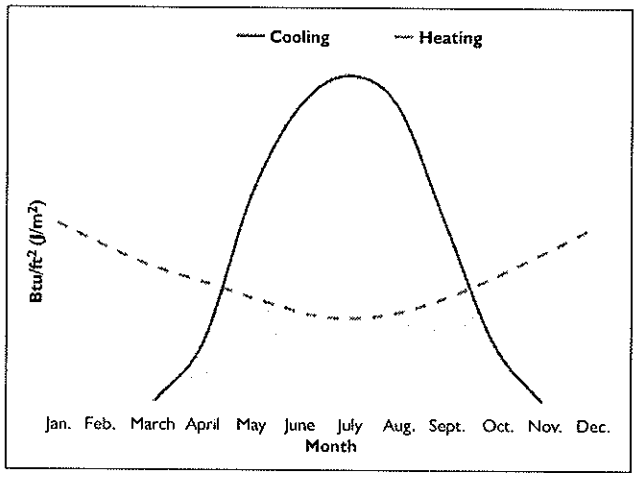


Figure 4: Optimum heating and cooling load profile for condenser heat recovery systems.

Results	Baseline	Dedicated Heat Recovery	Split Condenser Heat Recovery Chiller
Chicago	$1,630,568	$1,494,932	$1,499,952
Miami	$1,853,206	$1,654,514	$1,657,945
Philadelphia	$1,752,934	$1,601,086	$1,603,705

Percent Energy Savings	Baseline	Dedicated Heat Recovery	Split Condenser Heat Recovery Chiller
Chicago	N/A	8.3%	8.0%
Miami	N/A	10.7%	10.5%
Philadelphia	N/A	8.7%	8.5%

Table 4: Building energy comparison: base chiller system vs. energy recovery options.

Location	Baseline Energy Cost	Geothermal Energy Cost	Percent Savings
Philadelphia	$137,503	$118,429	13.9%
Minneapolis	$103,151	$90,260	12.5%
Helena	$112,064	$103,842	7.3%
Denver	$101,320	$95,737	5.5%
Chicago	$117,617	$106,021	9.9%

Table 5: Comparison of baseline VAV rooftop system and geothermal WSHP systems in a five-story 100,000 ft² (9290 m²) office building.

For this example, the baseline building is a five-story, 100,000 ft² (9290 m²) office building with standard office building hours. Referring back to Appendix G (Table 1), the baseline HVAC system is a packaged VAV rooftop system that meets the minimum requirements of Standard 90.1-2004. With the exception of the HVAC system change, all other design parameters such as lighting and building orientation were kept the same to provide an equal comparison of the systems. The results for five different cities are shown in *Table 5.* Energy cost savings range from 5.5% to 13.9%.

What About Costs?

The high efficiency of a geothermal system often is perceived to come at a cost premium, primarily because the costs associated with the ground loop are perceived to be high. As with many new technologies or systems, the cost premium can depend on the experience of the local market.

Table 6 shows the calculated simple payback for the geothermal system, which ranges from 1.3 to 13.3 years depending on the utility and maintenance cost savings, and the capital cost premium. Some assumptions are made in this calculation. First, it is assumed that the geothermal systems cost less to maintain than standard VAV rooftop systems. The maintenance cost savings vs. the rooftop VAV system was given a range from $0.02/ft² to $0.06/ft² ($0.20/m² to $0.60/m²).

Second, the capital cost premium of the geothermal system was given a range from $0.50/ft² ($5/m²) to $1.50/ft² (15/m²). It is often difficult to develop accurate information on the costs of different system types compared to equipment types. For this reason, a range was given for both maintenance savings and capital cost premium. As you can

see from *Table 6*, the cost premium for a geothermal system is easily overcome by cumulative energy savings in some climates and applications. Note that simple payback only looks at first year cost savings. Rising costs for natural gas and electricity in many areas of the United States can reduce the payback.

Optimizing Geothermal Systems

Three design parameters were taken into consideration to optimize the geothermal system: the effect of raising the loop temperature on operating and capital costs, using variable frequency drives (VFDs) on the pumps and using energy recovery ventilators (ERV) for the makeup air.

Loop Temperature vs. Operating Costs

The loop temperature of a geothermal system affects its efficiency and capital cost. A smaller geothermal loop will run at higher temperatures, decreasing the cooling efficiency and increasing the heating efficiency of the water source heat pumps. However, a smaller loop results in significant capital cost savings. *Table 7* compares the energy use and capital cost of geothermal systems with loops designed for water entering the heat pumps at cooling design conditions of 95°F (35°C) and 85°F (29°C). The cost of the loop is assumed to be $10 per linear foot ($30 per linear meter). From this chart you can see that the energy savings for the larger loop (85°F/29°C) is very small and does not justify the additional capital cost. Alternatively, one could say that the insignificant energy penalty associated with the smaller loop (95°F/35°C) justifies the capital cost savings.

Adding Variable Frequency Drives (VFDs) to the Pumps

The pumps serving geothermal systems are small, but they provide constant flow and run continuously. These small pumps can use a significant amount of energy over the course of a year. Adding VFDs to the system to provide variable flow can reduce this energy consumption considerably. In recent years, VFD costs have been reduced, so that the payback (in energy cost savings) is worth the capital cost premium for installing a VFD. Sometimes the cost of a VFD is almost the same as the cost of a pump starter. *Table 8* shows the payback for using a VFD in this geothermal system, considering three different capital cost premiums for VFDs. The payback ranges from less than one year to just over a year depending on the location and energy cost savings.

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Energy Recovery of Ventilation Air

In many parts of the United States, makeup air must be conditioned before it enters the building. Water source heat pumps generally are not suitable for handling ventilation loads because the compressors cycle on and off. In the off cycle, dehumidification does not occur and humid air can enter the space. A standard makeup air unit and an energy recovery ventilation (ERV) unit were modeled to supply ventilation air for the geothermal system. Using a range from $0.50/cfm ($1 per L/s) to $1.50/cfm ($3 per L/s) capital cost premium for an ERV system, *Table 9* shows some of the climates that are most favorable for an ERV system. In general, climates that can recover both heating and cooling energy achieve the greatest energy savings.

Geothermal water source heat pump systems are ideal for achieving high efficiency that pays back year after year in energy cost savings for building owners. While the installed cost of the system can be higher than more conventional systems, the payback is often favorable for achieving lower life-cycle costs.

Conclusion

The examples presented in this article can be implemented today in existing buildings, as in Example 1, or new construction designs, as in Examples 2 and 3, to improve the efficiency of HVAC systems and reduce building energy costs with solid economic payback.

Although whole building energy costs were modeled, only the energy cost differences of changing the HVAC systems were considered. The effects of changing the lighting, envelope, and miscellaneous electric loads in the building were not taken into account in order to focus on the HVAC system. All of these factors should be considered in the scope of high-performance building design.

References

1. Department of Energy Information Administration, www.eia.doe.gov.

2. *2004 ASHRAE Handbook—HVAC Systems and Equipment,* Chapter 8, Applied Heat Pump and Heat Recovery Systems.

Location	Annual Utility Cost Savings	Maintenance Cost Premium ($/ft², $/m²)	Capital Cost Premium ($/ft², $/m²) $0.50 ($5)	$1.00 ($10)	$1.50 ($15)
			Payback (Years)		
Philadelphia	$19,074	$0.02 ($0.20)	1.5	3.0	4.4
		$0.04 ($0.40)	1.4	2.8	4.2
		$0.06 ($0.60)	1.3	2.6	4.0
Minneapolis	$12,891	$0.02 ($0.20)	2.1	4.3	6.4
		$0.04 ($0.40)	2.0	3.9	5.9
		$0.06 ($0.60)	1.8	3.6	5.5
Helena	$8,222	$0.02 ($0.20)	3.2	6.4	9.6
		$0.04 ($0.40)	2.8	5.6	8.5
		$0.06 ($0.60)	2.5	5.1	7.6
Denver	$5,583	$0.02 ($0.20)	4.4	8.8	13.3
		$0.04 ($0.40)	3.8	7.5	11.3
		$0.06 ($0.60)	3.3	6.5	9.8
Chicago	$11,596	$0.02 ($0.20)	2.3	4.7	7.0
		$0.04 ($0.40)	2.1	4.3	6.4
		$0.06 ($0.60)	2.0	3.9	5.9

Table 6: Operating, maintenance and capital cost comparison and payback.

Location	Energy Cost 95°F (35°C) Loop	Energy Cost 85°F (29°C) Loop	Capital Cost Difference	Energy Savings	Payback (Years)
Philadelphia	$118,429	$117,249	$116,800	$1,180	99
Minneapolis	$90,260	$89,492	$75,800	$768	99
Helena	$103,842	$102,978	$66,100	$864	77
Denver	$95,737	$94,754	$92,400	$983	94
Chicago	$106,021	$105,031	$95,400	$990	96

Table 7: Energy and cost comparison of 95°F and 85°F loops.

Location	Utility Cost Savings	Capital Cost Premium $6,000	$8,000	$10,000
		Payback (Years)		
Philadelphia	$12,406	0.48	0.64	0.81
Minneapolis	$8,581	0.70	0.93	1.17
Helena	$9,868	0.61	0.81	1.01
Denver	$9,597	0.63	0.83	1.04
Chicago	$10,612	0.57	0.75	0.94

Table 8: Constant flow vs. variable flow using VFDs.

Location	Utility Cost Savings	Capital Cost Premium ($/cfm, $/L/s) $0.50 ($1.00)	$1.00 ($2.00)	$1.50 ($3.00)
		Payback (Years)		
Philadelphia	$4,869	1.85	3.70	5.55
Minneapolis	$7,274	1.24	2.47	3.71
Helena	$5,615	1.60	3.21	4.81
Denver	$2,768	3.25	6.50	9.75
Chicago	$5,919	1.52	3.04	4.56

Table 9: Payback results using energy recovery ventilation units.