BEFORE THE
CALIFORNIA ENERGY COMMISSION

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| In the matter of Staff Workshop on Draft Nonresidential Chiller and Cooling Tower Efficiency, Air Compressor, and Control Systems Revisions for 2013 Building Energy Efficiency Standards | Docket No. 10-BSD-01 |

CALIFORNIA ENERGY COMMISSION
HEARING ROOM A
1516 NINTH STREET
SACRAMENTO, CALIFORNIA

WEDNESDAY, APRIL 27, 2011
10:00 A.M.

Reported by:
Kent Odell

Present: (* Via WebEx)

Staff Present:

Patrick Saxton
Ron Yasny

Presenters

Mazier Shirakh, CEC
Mark Hydeman, Taylor Engineering
Erika Walther, Energy Solutions
Dave Watson, LBNL
Jeff Stein, Taylor Engineering
Reid Hart, PECI
Matt Tyler, PECI
Russell Torres, Energy Solutions

**Also Present**

Attendees

Eric Bessey*
Larry (Lee) Burdick, Trane Co.
Ransom Byers, Energy Solutions
Mark Cherniak, New Buildings Institute
Bill Dietrich, Baltimore Aircoil Company (BAC)
John D. Douglass (ph.)
Reid Hart, PECI
Trevor Hegg, Evapco, Inc.
Paul Lindahl, SBX
Richard Lord
Mike McGaraghan, Energy Solutions
Jon McHugh, California Statewide Codes & Standards Program, McHugh Energy
Frank Morrison, Baltimore Aircoil Company (BAC)
Kirk Oatman
Josh Rosa, California Association of Sheet Metal and Air-Conditioning Contractors
Mick Schwedler, Trane Company

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P R O C E E D I N G S

APRIL 27, 2011 10:01 A.M.

MR. SHIRAKH: Good morning. I’m Mazi Shirakh. I’m the Project Manager for the 2013 Building Energy Efficiency Standards. This is the fourth staff workshop that we’ve had this month on Nonresidential Buildings, and we have a long agenda today. We’re going to start with going over the agenda and then some brief introductions.

So the topics for today are going to be Chiller Efficiency and k Factors, Air Cooled Chillers, and Cooling Tower Efficiency, and all of those topics are going to be presented by Mark Hydeman of Taylor Engineering; then we will break for lunch around 12:00, then coming back we will talk about Cooling Tower Water Efficiency and Erika Walther will present that topic; and then we’ll briefly talk about Automated Demand Control for Nonresidential HVAC, and Dave Watson of LBNL will present that; and at 1:50 is going to be Single Zone VAV Fan Control and Integrated Economizers, and Jeff Stein of Taylor Engineering will represent that; Reducing Reheat, and again by Jeff Stein; HVAC Controls and Economizing is going to be Matt Tyler of PECI; about 4:00 p.m. is going to be Air Compressors by Russell Torres of Energy Solutions; and then there will be public comment and we'll adjourn around 5:00.  These times are tentative. If you're interested in a topic, you need to be paying attention throughout the day because, you know, we may go faster or slower than what is indicated on the agenda and we'll just have to deal with it.

I have a set of slides that I want to present. Most of you have seen these before, so I'm not going to spend much time on most of them.

Again, I'm Mazi Shirakh and Martha Brook is not here now, we are the Project Managers for this effort. These are the Policy Goals that we're trying to pursue, which is the articles and objectives for the 2013 Standards, that most important one being zero net energy goals for residential and non-residential buildings.  And the goal is zero net energy for residential buildings by 2020 and nonresidential by 2030, and these are our collaborators, which includes the California Investor-Owned Utilities, PG&E, SCE, SDG&E, and Southern California Gas, with PIER support, and we also seek input from the public.  These are the famous Rosenfeld Graphs that show the effect of Buildings and Appliance Standards on California's per capita consumption and, basically, what it shows, the green graph here, the per capita income energy consumption in California has been relatively flat, while the U.S. as a whole has been increasing. The next graph basically shows the same thing, that California is probably the most efficient state in the Union when it comes to per capita income, partly or largely due to Buildings and Appliance Standards.

Again, these are the policy goals for this round of Standards. We're seeking anywhere from 15-25 percent energy savings relative to 2008 Standards. And the 15 is more indicative of nonresidential buildings and 20-25 is more indicative of the residential buildings.

Another goal of the standards this time around is to align our timelines with the tri-annual cycle of the California Building Standards Commission. And we're for the first time also publishing Reach Standards, Part 11, as part of these proceedings. We're trying to address several compliance and enforcement issues with this round of Standards, a simplification of Standards is one of our goals and, as part of that, we're migrating many mandatory measures into prescriptive measures, and mandatory measures are typically more easily understood and enforced. We're reviewing and reducing the number of exceptions in the standards, exceptions to the complexity of standards and making it not clear what the actual requirements of the standards are, so we're looking at the whole list of exceptions to be eliminated. We're trying to create user-friendly compliance forms and a form generator. The idea here that users could answer a series of questions related to their project, they need to know nothing, next to nothing, about the forms, and the form generator will fill out the forms for them, this is not unlike some of the tax software that people use to file their State and Federal taxes, where you don't need to know much about the forms, you just need to answer the questions and the software will generate the forms for you.

We're also trying to simplify the performance software interfaces to make it easier for alteration projects, the idea here is that you can actually indicate the type of building systems that you're interested in, like just envelope measures, maybe cool roofs and insulation and do trade-offs against those two, and the program will neutralize everything else that's not part of the project, like HVAC, water heating, and so forth. Improving third-party verification and acceptance requirements, we're looking at all those and trying to clarify or improve them. Improving electronic record-keeping, the CEC Central Document Repository, we're building on the 2008 Standards requirement for HERS registries and we're expanding that to create a central repository where all those forms can be found and can be used for program evaluation and enforcement actions. Integrating energy efficiency and demand controls, things like the controllable ballasts, are also a part of this cycle of Standards. We’re trying to capture some of the measures that are not directly energy-related, like global greenhouse gas emissions that may not have a directly benefit or impact on the building itself, but it has an impact on the amount of carbon or equivalent. So, we’re trying to capture those. For the first time, we’re looking for direct water savings as part of these standards. We’re going to be considering roof deck insulation for residential buildings, in addition to ceiling insulation, which is one of our probably biggest energy savers for this time around; encouraging proper building orientation to take advantage of the sun for proper placement of the PV systems, and solar hot water heating.

This is the schedule for the 2013 standards and we’re right in the middle here, where we’re holding the staff workshops to present the result of the case efforts that have been going on for a while. And later this year in September to March, we’re going to be moving into the rulemaking phase of the Standards, where we’ll present it for the 45-day and a 15-day language, and adoption is set for March 1, 2012.

The Building Standards Commission will adopt these and publish all parts of Title 24 sometime in the – I can't read the date here, it's blocked by this, but I think it's in July of 2013. And the effective date of the Standards is January 2014.

As with all the other cycles of Standards, we do lifecycle costing as the basis for each measure per climate zone bases, and to that, we have to update our Weather Files, we have to update our Time Dependent Valuation, TDV values, for both Base and Reach Standards, and update our lifecycle cost methodology, which was presented in a workshop back in November of 2011. The documents are all online.

And this time, we're working with the IOUs through their Case or Stakeholder Meetings, and the IOUs have been holding these meetings over the past year and a half or so throughout the state and most of the topics you'll be hearing today have been presented at least three times to the stakeholders, so if you've been participating in those meetings, this should not come as new material to you.

Again, we're holding seven or eight days of workshops this spring and this is the fourth one, and these are the dates. The previous ones were April 4th, that was the lighting and res lighting, April 11th was the ventilation issues, the 18th was Acceptance Testing, Design Phase Commissioning Refrigerated Warehouses, Supermarket Refrigeration, Solar Rated buildings, and Solar Hot Water Heating, and that was presented last week.

Today's topics, you know, we went over the agenda. Next May 5th will be probably the last non-residential topics presented and the only exception is next week we'll also be talking about the residential domestic hot water. May 24th, May 31st, and June 9th are three dates that we have set aside to preset our residential topics. The agendas will be released for those dates. Later on in June, we will have one day to present our Reach Standards for both residential and non-residential buildings, and also late in June we'll have a workshop to present the modification to the ACM Manuals. We're trying to work on the software that will be used for compliance for 2013 standards, and Martha Brook is leading that effort, and the goal is to have this software in place in time, way ahead of the actual effective date of the Standard.

If you have any questions or comments related to today's material that is presented, please send them to me by May 4th, which is a week from next Wednesday. So, with that, I'm going to close my presentation and if there are no questions, I'm going to turn it over to Mark Hydeman to talk about Chiller Efficiency and Chiller k Factor.

MR. HYDEMAN: Thank you. Okay, I'm going to cover the Chiller Efficiency Measure and this is, again, a continuation of a number of workshops that we've had, we had one last week and I made some changes based on input that we received at that meeting last week. I also have rolled in, if you've looked at earlier versions of this, we just completed the analysis last night and so what you're going to see up here is relatively fresh information. We had to re-do all of the analysis due to errors that were found in the curves in the initial analysis, which was presented to us by HRI.

Overview – I'm going to talk about just briefly kind of the history of this, and then we'll get into the actual measures. Chiller Efficiency has been unchanged since 2001. Title 24, Chillers are not federally preempted, but Title 24 has up to now always followed 90.1. 90.1 2010 recently published – had a number of changes, very significant, in the area of Chiller Efficiencies, Addenda M provided higher efficiencies for chillers, and two paths for compliance, Path A being a fixed speed machine, Path B typically being a variable speed machine, and one could comply either with Path A or

Path B in the 90.1 method of compliance. And then, Addendum BL and BT, which also were adopted in the 2010 Standard, dealt with this “k” equation for non-standard – basically centrifugal chillers that are not designed to operate at the standard ARI 55590 conditions of 9585 and 44˚ chill water temperature. And we used to have a whole bunch of tables in the Standard to deal with those non-standard conditions; it turned out the range of that k equation was quite limited and we’ve gone from, I think, about 28 percent of the market being covered at present to the vast majority of the market being covered now with the extended range equation.

Addenda M also deleted the category of Air cooled Chillers without condensers and consolidated all of the positive displacement chillers to one set of requirements, so the same requirements apply to screws, scrolls, and reciprocating.

So, what we’re proposing for Title 24 2013 is under mandatory following exactly what’s in 90.1, will adopt 90.1’s chiller efficiencies, both Paths A and B, will delete the air-cooled category without condenser, will consolidate all the positive displacement chillers, will adopt a new “k” equation, and now that we have the new “k” equation, and it’s much broader, we can delete the non-standard chiller tables 112H, I, J, K, L, and M.

The prescriptive path, we're proposing to require Path B and this is based on lifecycle cost analysis, which I'll be showing you here this morning. And we'll provide exceptions as noted in some of the following slides that are responses to industry comments that we've received in previous workshops. And then, the performance path will have a budget system that follows the prescriptive requirements. So, although we're going to be more restrictive than 90.1 is, and prescriptive by requiring Path B as opposed to alternately Path A or Path B, one could get by with a Path A Chiller minimum requirement and trade off that energy elsewhere in the building.

So the first thing I'm going to change is the definitions, we're going to update the references as was done in 90.1 from ARI 55590 1998 to ARI 5590 2003, you can see this with underline and strikeouts. This table is straight out of 90.1 and the section I've got blotted out there will not be relevant by the time Title 24 2013 takes place, so we'll only be looking at the Path A and Path B requirements for various classes of equipment. And, again, we're suggesting that this table, the contents of it will replace Table 112(d) Mandatory Table in Title 24. This is exactly what was adopted in 90.1.

There's a number of notes under the table, these are not the exact words in the notes, but it gives you the intent. There are no requirements for centrifugal chillers that are operating at very low evaporator temperatures less than 36˚ Fahrenheit. Positive Displacement Chillers, again operating at low temperatures, less than 32˚, and Absorption Chillers operating at less than 40˚.

When you comply, you comply either to Path A or with Path B, and you must meet both the COP requirement and the IPLV requirement to comply. And that is from one path, either Path A or Path B. Note C refers to the definitions, which goes back to the ARI 5590 Standard. If it says NA, it means Not Applicable, and if it says NR, it means there is no minimum requirement for that field. Exception to Section 112A, it used to say “Water Cooled Chillers, blah blah blah,” this is all the non-Standard stuff. We’re getting rid of that because the new “k” equation will take over, and here is what the new “k” equation says -- I’m not going to read this, but this is straight out of 90.1, nice bit of mathematics down there. And again, just straight out of 90.1, and we would be putting this in as an exception to Section 112(a), which refers to the chiller efficiencies.

Proposed Code change prescriptive – it’s whatever is the lowest lifecycle cost. I can tell you, as of last night, Path B came up again as being cost-effective. We did this previously with the AHRI's chiller curves, the same ones they use to evaluate the Addenda M impact, for 90.1, and found some errors in the curves after we did our analysis, so we then went and re-did the analysis with real machines that we had data from, from performance-based bids, and that fit. As I'll mention later, we took these curves that represent real performance off of real machines, and fit them to the COP in each category, the baseline Path A and Path B, and then checked to see which curve gave us the closest fit to the IPLV when you calibrate the curve to the COP. And that's what we'd use.

So, we're providing exceptions, I mentioned earlier there were some industry comments that these variable speed-driven chillers, Path B, aren't always the most effective and one issue that was brought up was that the cost curves that we used from HRI did not take into account the increased cost for high voltage chillers, so 12 kva or the 2130, or 4160, I think, are the two voltages, high voltages, that people use. To put variable speed drives on chillers at that higher voltage has a very high increased cost, and so we're going to give an exception to this prescriptive requirement for high voltage service.

Chillers attached to heat recovery systems often need to have a little bit more lift, a fixed speed chiller is often used for this, and so we’re saying if the heat recovery capacity is greater than 40 percent, the design cooling capacity will allow them to use a Path A chiller, or Path B, either one. It just won’t require Path B. Chillers used to charge thermal energy storage systems, again, it’s a high lift condition where the charging temperature is less than 40˚. We would then be able to use either path under the prescriptive and chillers installed in plants with no more than three chillers, and there the issue was if you have four, five, or six chillers, then the unloading characteristics of any individual chiller would become less important because you have now multiple stages by the fact that you have many many machines. So, all those were in response to industry comments.

We received a letter from Trane Co. and this is summarized here, the actual letter is a part of the case report that’s now up on, I think, the Energy Commission website. There was an issue that they believe we misused IPLB in the analysis and the correction is that we didn’t actually use IPLB, we used full DOE2 curves, so we didn’t just use a point of data that represented four points of operation, but we actually modeled these things in an

8760 model, using eQuest and useful curves. And we actually had a stakeholder workshop last week and Trane admitted that they didn't fully understand the analysis and so they feel that this is resolved.

We failed to factor in cost for VFD's [ph.] was brought up, but we used ARI's cost, which I'll show you, for the analysis, and that did include in Path B the cost of the variable speed drives, and I also point out that, when we go out and do performance-based chiller bids, and our company does a lot of these for facilities we're building, variable speed drives, when the lifecycle cost analysis for, you know, real lifecycle cost for customers using their discount rates in their lives for equipment, so they have been, in fact, included and, again, that was a point that was conceded at our meeting last week with the exception of the high voltage cost premiums for high voltage variable speed drives, but we provide an exception to get around that, or to address that.

We failed to factor in electric demand and, in fact, the TDVs employed in the analysis do include electrical demand, electrical energy, and some T&D costs. And Mazi had mentioned earlier that the report on the TDVs is up on the CEC's website and available for everyone for review, and again, this point was discussed at our workshop last week and seemed to, again, the authors of the letter were fine with the response.

And then, these are other issues that were brought up and all of them, now, we've provided exceptions in the proposal to address them.

So, as I mentioned earlier, we did a DOE2 model, it's a five zone per floor, 15-zone model, so north, south, east, west, kind of the classic Title 24 nonresidential model, 10 floors, 100,000 square feet, you can see the occupancy lighting and equipment assumptions in there. The plant was two equally-sized chillers. We've received actual data from some manufacturers that we unfortunately cannot share because it was sent to us as a proprietary data, but they got it from their service organization, that showed that, in fact, two equally sized chillers is probably the most – is the most prevalent distribution of chillers. And certainly in our experience as a design firm, that's what we see most. So that's what we modeled, two equally sized chillers, chiller size based on the load. We have one two-cell cooling tower, so these are water cooled chillers and then we have air cooled chillers as just two equally sized chillers, and then the other conditions are down there below.

The Climate Zone 3, I'm going to show you actually eight climate zones, the analysis, but here's the load profile for Climate Zone 3 in a building that meets the minimum requirements of Title 24, which includes economizers, either air or water that, in this case, add air size Economizers, lots of hours at very low load. The blue line is the chiller, the red line is the lag chiller. So you can see how they're loaded.

Okay, so I'm now going to go through eight climate zones, they're all going to look very similar. The scale changes on the left, this goes from zero to $6,000, and that is a 15-year lifecycle cost per ton using the TDVs, so it's got the first cost premium, the chiller, and the energy cost using TDVs. The blue lines are the baseline efficiency, which would be Title 24 2008 base, and the red line is Path A, and the green line is Path B. There is no green line for air cooled chillers, there's really only a Path A, but you can see there's a green line on these charts, I apologize, again, I received the data very late yesterday and didn't have a chance to clean them up. But the other categories starting with the water cooled positive displacement, that's a WCPD and what are called Centrifugal, different size ranges, they each have a separate baseline Path A and Path B. The thing to note here is this is present value of the first cost of the machines and the energy costs, and in every case, Path B – this is for everything but air cooled – is the lowest in each category. Same thing in Climate Zone 6, Climate Zone 7, Climate Zone 8, Climate Zone 9, Climate Zone 10, Climate Zone 12, Climate Zone 13. These eight climate zones represent 85 percent of the new construction starting in 2013, according to the Dodge [ph.] database.

So, the incremental costs we got, we do not go out and get costs for these chillers, HARI already did that as part of their work with 90.1, to show that these requirements for Path A and Path B were cost-effective, and these are incremental costs for Path A and Path B over the minimum based standard, and these are in dollars per ton, so we used exactly the same data that they had. So, next up, so we have to complete the analysis for the remaining climate zones and then we have to recalculate the statewide savings. As I mentioned earlier, we did this for all the climate zones with the curves we receive from HARI and, then, to our chagrin discovered that the curves had some errors in them. So, that's why we're redoing it.

Mazi, should we wait and take questions at the end of all of this?

MR. SHIRAKH: I say let's take questions for this topic now before we move on, we have got plenty of time. Any questions in the room related to the Chiller report of efficiencies? How about online? Our driver is not here. Can people online, can somebody speak so I can -- can somebody say hello?

MR. HYDEMAN: Yeah, I'm going to go through these next three slides. We cannot hear anybody online and I suspect that there are some comments, so we'll wait until the technician comes back and we'll address your questions, but for right now, let me go ahead and go through the air cooled chiller issue.

There's been some concern by people who are involved on the CEC side with compliance issues with the air cooled chiller limitation in 144(i), this is an existing requirement that we put in, I think, in 2005. And the issue that was brought up is that there's a lot of confusions over how to apply this requirement and it appears that people are sometimes actually gaming systems to get compliance with air cooled chillers, and we've seen data, by the way, from HARI that shows that shipments have gone way up in air cooled and way down in water cooled and, in fact, we know from the analysis we did in 2005 that air cooled chillers are not as efficient as water cooled chillers in our climates.

So, one of the issues brought up is, if somebody puts in a 299-ton chiller, could it be all air cooled because the limit currently is at 300 tons. But however above 300 tons, you can only have 100 tons worth of air cooled, so people are sometimes providing multiple permits, so they put in chillers one at a time at 299 tons and can build up a 900 ton plant.

People put in two very large chillers whose combined capacity would exceed the 300 ton limit and they put an interlock in it and then they subsequently come and get rid of the interlock and now they can run both chillers, so same thing. People put in smaller than 300 ton air cooled chillers in a series of permits, so I think I already addressed that issue. And then, finally, there’s kind of an interesting lack of coordination between 144(i) which is this prohibition on air cooled chillers, or limitation on air cooled chillers, and the sizing requirements we have for equipment in 144(a). So, the proposal to clean this up is to change 144(i) and 149, which is the renovation and retrofits section, so I’m showing 144(i) here and then I’ll go to the renovation section in a moment. 144(i), we’re saying that chilled water plants shall not have more than 300 tons provided by air cooled chillers. This makes the intent very clear, whether it’s a new plant, or an existing plant, that once you reach your limit, then you can’t add anymore air cooled chillers, the next steps have to be water cooled. And so it gets rid of all that confusion.

And then, I've cleaned up some of the language under Exception 2, to match the proposed language that we have for an exception for thermal energy storage systems under the other proposal we talked about this morning, the prescriptive proposal.

And under Proposed Changes to 149(c), Additions and Alterations, we're recommending that we just simply strike this because now the language in 144(i) applies to the total tonnage of air cooled chillers in a plant. We don't need to redefine under 149 what happens in an existing plant when you're doing an expansion, and this simply just renumbers the previous exceptions. And we just need to get feedback on this proposal, so that's about it.

Lifecycle Cost Analysis, by the way, for this proposal, back in 2005 and still to this date, it is posted on the CEC website if you go to the 2005 Standard and look under Workshop Reports, you can find it. It's hidden under the Cooling Tower Measure Report. With that, can we open the floor to questions from the Web?

MR. SHIRAKH: We have Ron back in the room. Ron, can people online be heard? Before, we couldn't hear.

MR. HYDEMAN: Let's try that and if it becomes a problem, then we can have them raise their hands and we'll get them.

MR. SHIRAKH: Richard, I think we can hear you if you want to make a comment.

MR. LORD: Can you guys –

MR. HYDEMAN: Yeah, we can hear you, Dick.

MR. LORD: Okay, just a couple quick comments. Look, the HARI analysis was not wrong, it used a different method than you're using – so the stuff we did for ASHRAE is correct. The preliminary curves I developed, as you know, do have an error in them, but those are not HARI curves, they are my curves.

MR. SHIRAKH: We have a Court Reporter here and he needs to get your name and your affiliation.

MR. LORD: Yes, Dick Lord with [inaudible].

MR. HYDEMAN: Okay, Dick, I didn't mean to –

MR. LORD: Not a big deal, Mark, I understand where you're coming from.

MR. HYDEMAN: I just wanted to say that there was an issue with the curves and I know that P&L [ph.] decided not to use them and we're trying to match what you did as a member of this working group in the analysis, so we used real chiller curves, we didn't have access to all the data that you had access to, and we matched the COP and IPLD and we're happy to send you those spreadsheets.

MR. LORD: Yeah, what you do is good, I mean, I understand that and, you know, the issue is my fault, but you know, the analysis done for ASHRAE 90.1 is correct because it was done with a different method.

A couple other things, Mark, is – it’s a minor one, you’re changing the reference to HARI 55590 2003, within about a month it’s going to change to 2011. It shouldn’t impact any of the stuff that you’ve done here, it does have a lot more details on testing qualification, instrumentation, and a lot of other just little minor corrections to the standard, so my suggestion would be is you change yours to 2011, anticipating this is going to be released.

MR. HYDEMAN: The one thing I’d ask you, Dick, if you don’t mind, just shoot me an email copy of the draft 2011, just so I can review it.

MR. LORD: Sure.

MR. HYDEMAN: Great.

MR. LORD: We can do that, yeah. I’ll mark it “preliminary” just because it hasn’t gone through a final vote and we’ve got a couple editorial corrections.

MR. HYDEMAN: Great, thanks.

MR. LORD: And the other minor thing I noticed, you noted that positive displacement chillers can run to 32˚ F, that’s not the case, actually water freezes at

32˚, so we don’t let them run at 32˚. What we did in 90.1, and go back and look at the words, is we said any positive displacement chiller that has a fluid for freeze protection and a set point above 32˚ F, has to be certified and show compliance with the standard by operating at the standard ARI rating point. And that’s the language you really ought to put in Title 24.

MR. HYDEMAN: Well, again, what you’re looking at here is a synopsis of the actual footnotes because it was too hard to get the actual footnotes into the slide, but I plan – if you look at the workshop report, I’m taking the exact language that’s in the footnotes from 90.1, so I think, you know, the intent here is to have exactly the same language that 90.1 has.

MR. LORD: Okay, yeah, we just did that and that was kind of a big deal, it’s not as big a deal in California, but you know, a lot of chillers were getting around the qualification criteria because it’s charged with Glycol, so it’s outside the scope of the standard. And that’s what that note was all about.

MR. HYDEMAN: Okay, so I’ll make sure you get a chance to review those footnotes and that we’ve got them right, but I literally pulled the text – if you look at the case report, which should be posted hopefully by the end of the day today, the draft case report, the footnotes that we have are exactly the same verbiage that's in 90.1 2010.

MR. LORD: Okay, good. I just wanted to double-check on that, just to make sure. All right, and those are my questions.

MR. HYDEMAN: Thank you, Dick.

MR. BURDICK: Mark, this is Lee Burdick from Trane. Can you hear me now?

MR. HYDEMAN: We can, Lee. And thank you for giving us your affiliation, as well.

MR. BURDICK: Okay. A question for you with respect to the analysis that you re-ran. Looking at Path A, Path B, did you use any combinations of Path A and Path B?

MR. HYDEMAN: We did not, Lee. We said that we would go ahead and run one or two test cases with a mixed plant, and I'm happy to share that analysis with you when I get it, but we were scrambling to get eight climate zones with the base analysis, so, I will send that to you as soon as we do it.

MR. BURDICK: Okay. The only thing that I've got any problem with at this point is in the prescriptive path where you're saying use only Path B, if you said use Path A, Path B, or a combination, based on lifecycle cost, then I think that leaves it free for designers to explore combinations of A and B, and it still is based on lifecycle cost. So, my suggestion would be to add in there A or B, or a combination of A and B.

MR. HYDEMAN: Well, that would be the same as striking the prescriptive requirement because, under mandatory, they could do either A or B and, so, what we have in here is similar, again, in that we're setting a bar that says the bar is based on a plant that has Path B minimum requirement chillers, and if someone wants to go and come up with a better mousetrap, they found a way to control Path A chillers, or a combination of Path A and B chillers, they can still do that. Being a prescriptive requirement, this merely sets the basis for the performance method and, in the performance method, you could take a plant with nothing but Path A chillers and show that that plant, as operated in the performance method, used equal or less energy, using the TDV values in California, than the Path B chiller that's mandated under the prescriptive, so, in a sense, we're doing that by putting this in the prescriptive, not in the mandatory measures.

MR. BURDICK: But as long as the prescriptive path is based on lifecycle cost, and I think that's proper, then why do we need to specify it be A, or B, or a combination, simply leave it open?

MR. HYDEMAN: Well, it seems to me that if you leave it open, you might as well – what you're saying is we should not have a proposal here that says –

MR. BURDICK: What I'm saying is, you know, in a practical way, if it's based on lifecycle cost, and what you're telling us, based on your experience, is you see B as the best way, and at least in most cases, fine, that's the way it turns out. But it leaves at least the opportunity for combinations of A and B to be the lowest lifecycle cost, and that that be, then, the basis.

MR. SHIRAKH: I think what Mark is saying – this is Mazi at the Energy Commission – is that, by picking Path B, basically what we're doing is we're establishing our performance budget based on Path B, so that's the framework for what the performance budget should be. Now, if you go to performance, you can do either/or. It doesn't really restrict you from doing Path A or a combination, or using the performance, or anything else, it just – you're going to be compared against a building that has Path B equipment in it, but you can do anything you want.

MR. BURDICK: But I think, if I understand it correctly, that the intent here is that the budget system on a performance path be the lowest lifecycle cost. And if a combination of A and B provides the lowest lifecycle cost, then you’re not getting that into the budget system.

MR. HYDEMAN: Well, Lee, if I could speak on this, again, these standards are not correct for every case, they have to be correct for the majority of the cases, and that’s why we do all the analysis that we do on them. And albeit the analysis is somewhat simplified, it does cover a wide range of the applications, and so it’s largely correct to say Path B, given the cost that the HARI developed for the 90.1 work and the curves that we have in the climates of California, with the TDV values we can now say that Path B in a two chiller plant, with both chillers, is the lowest lifecycle cost – as we showed in the eight climates that we’ve done. Now, that doesn’t mean that we’ve done every single case, right? We’ve got equally sized chillers, not unequally sized. We’ve done it for a protypical office building, you know, there are all sorts of other facilities out there. But this covers a large percentage of the applications, and so, again, it’s consistent with what we’ve done in other – people have done in developing requirements for Title 24.

MR. BURDICK: Okay, I follow what you’re saying. I guess, just to emphasize my point, is that you needn’t constrain the use of A or B, or combinations, to achieve your objective.

MR. SHIRAKH: Again, if I understand correctly, we’re not constraining, we’re just setting the budget based on Path B, which Mark has demonstrated it is the lowest in most cases and climate zones in California; there could be exceptions to it, but, you know, we have to set our budget based on something and there seems to be pretty good rationale for using Path B. But, again, that doesn’t constrain any sort of alternative to that.

MR. BURDICK: Uh huh, well, I follow what you’re saying, you know, it would be interesting to see what Mark comes up with if he gets the opportunity to look at combinations based on the surveys and analysis here.

MR. SHIRAKH: Okay, I think he has offered to do that.

MR. BURDICK: Yes.

MR. HYDEMAN: Okay, is there any other questions or comments? Okay, if you’re trying to speak out there, the best thing to do is to type in a message and then we’ll know to unmute you, or raise your hand. Okay, so anymore comments on this, either from people here or out in the Webland? Okay, well, thank you very much and, Mazi, are you ready to switch to the other one, then?

MR. SHIRAKH: We also presented the Air Cooled Chiller topic. Any comments on the Air Cooled Chiller?

Clarifications in the room or online? I hear there are no questions, so we’re going to move to the Cooling Tower Efficiency topic. Again, we’re a little bit ahead of schedule, so we’re just going to talk about this topic and, if we’re done early, we’ll just have a nice long lunch.

MR. HYDEMAN: Are you concerned about people calling in for this later?

MR. SHIRAKH: No, but we can talk about the Cooling Tower, but I think we need to break and then regroup for the afternoon topics because, yeah, then I’ll be concerned about people not being dialed in.

MR. HYDEMAN: Okay. Okay, so I’m going to go ahead and move to the Cooling Tower Energy Efficiency stakeholder topics and, again, just like we did with the chillers, this is probably the fourth or fifth time we’ve presented material on this, and it’s changed again in response to comments. We received on our original proposal a letter from members of TC 8.6, that’s the Cooling Tower TC, actually, I think it’s evaporative cooling, or something like that, the name of the TC, but essentially the manufacturers of the Cooling Towers and people – there are also engineers and others that are members of 8.6, but, anyway, the letter that was written had a number of issues that were raised and we tried to address those in this revised proposal.

So, cooling tower efficiencies in both 90.1 and Title 24 have been largely unchanged, in fact, with one exception; in the case of California, we actually put a limitation on centrifugal cooling towers and that now got adopted in 90.1. But the actual efficiencies, the gpm per horsepower at 95/85/75 had not changed since we developed them for the 90.1 1999 standard. At the time that we did that, we did not do thorough analysis, it was the first time the cooling towers had any regulations whatsoever, and it was just an agreement between the manufacturers, members of TC 8.6 at that time, and the 90.1 Committee, to cut out the bottom approximately five percent of the market. And that's what we did with the efficiencies that we have. And there was no analysis at the time to determine the efficiency levels. Like chillers, cooling towers are not federally preempted, but, again, Title 24 followed 90.1, we added a requirement for variable speed drives, both the 90.1 and Title 24 for 7.5 horsepower and above, fans on cooling towers. And in 2005, we added a requirement for minimum flow at Section 144(h)(3), it's a prescriptive requirement, and the idea there is that it is always more efficient to run as many cells of cooling towers as possible. Every time you double the cells, you drop three-quarters of the energy for the same heat rejection. In 2005, we also put in a restriction on centrifugal fan cooling towers, they use twice the energy for the same service, generally, as propeller fan towers. And a similar provision is now in 90.1. ASHRAE 90.1 added requirements for Closed Circuit Cooling Towers in the 2010 Standard, and so we’re going to pick those up, as well.

So the measure scope is – the prescriptive scope, what we’re going to recommend, is for new construction only and we’re not going to propose it for replacement or expansion, as space is often limited, so if you’ve got an existing roof, you’ve got an existing structure, also there is the issue that you can’t mix towers together, the basins, atmosphere is atmosphere, the top of the water basins have to be at exactly the same level and if you don’t do that, bad things happen like water going down, out of the tower basin, into the building, and people generally don’t like that. It covers commercial industrial and institutional, and we’re talking about evaporative cooling towers.

Proposed Code change – in mandatory, add requirements, straight out of 90.1 2010 for closed circuit cooling towers, and we’re not going to touch the open towers. I didn’t put it here, but I did put it into the case study, so the report that we’ve got up on the website, we’ve added a definition for CTI ATC-105S, which is for closed circuit cooling towers, taken straight out of 90.1.

Under Prescriptive, we’re recommending a minimum efficiency for cooling towers, and this is prescriptive only, it sets a basis for the performance method, so it does not prohibit the use of less efficient towers, but you have to go the performance method, and we’re setting it at 80 GPM per horsepower rated at the CTI ATC conditions of 95/85/75, and we previously were showing in our analysis that a higher efficiency was justified for 24/7 facilities, but we’re pegging it at 80 gpm per horsepower in response to some of the industry feedback that we received at the last workshop.

Again, for new construction only, we’re recommending for 24/7 facilities that we have a maximum approach of 5˚, and that would be for things like data centers, manufacturing facilities, and laboratories, but no requirement for office buildings, and I’ll show you why in a moment.

And then, finally, the minimum flow turndown turns out, even though we went and we surveyed the manufacturers when we put this in in 2005, and everyone said they could live with the 33 percent, in practice that's been very hard for them to achieve. Since the most common plant is two chillers, two tower cell plants, we're going to increase this up to 50 percent. The intention is not, you know, it's to make it high enough that manufacturers are not jeopardizing the tower performance by meeting the requirement, and this has been based on several years of feedback.

Analysis – we did an office building, a nominal 900-ton load, but we sized the chillers to be in excess of that, which is quite difficult, so it is a two chiller plant, equally sized, each at 500 tons, meeting a 900 ton peak load with two cell cooling tower. The cooling towers were designed for 50 percent flow turndown, and we used eQuest to develop the load profile, but then to actually look at the cooling tower performance and the plant performance, we used what we're calling the top model, which is a way of optimizing the control sequences, when you have things like variable condenser water flow and variable speed drives on the cooling tower cells, and lots and lots of choices. What it basically does is it discovers out of all the modes of operation, the one that operates most efficiently, so it basically tells us what the potential is for the set of equipment that's on the building. The input is, again, data we got out of an eQuest model, outside air dry bulb and wet bulb, so the weather data, the chill water load, the flow, the tons, chill water supply temperature, and return temperature from the eQuest model, and then we get data specifically on individual pieces of equipment which we calibrate to the chillers, the towers, the pumps, the heat exchangers if we have water side economizers. And then we step everything through the modes of control, so we look at every hour, the load, and we vary the number of chillers, one or two. With the cooling towers, we always run the maximum number that are allowable within the flow limits because we know that's always the most efficient. The condenser water flow, we actually go from 10 percent to 150 percent, in 10 percent increments, and the cooling tower fans from zero to 100 percent in 10 percent increments, and every hour we look at the load and we say, "Has the load been satisfied?" If it has been satisfied with that condition, we store the data, if it hasn't, then we throw the data out and it's not considered, and then we take every hour and we look at the plant energy for each hour, take the minimum plant energy, add up the sum of the minimums and that is the score for that plant, or the kilowatt hours. In the case of using Title 24 TDVs, it's the lowest TDV value for each hour summed up, over the year.

Okay, the original simulations were run in

Oakland, Albuquerque and Chicago, we just ran the same eight climates I showed you for cooling towards, and the towers that we used, we had literally 12 different towers, and I'll show you the models in a moment. They represented different approaches and three levels of efficiency. So, the approaches vary, depending on the application, so the approach changes with the wet bulb, but the Tower A was roughly three to five degree approach, Tower B was five to seven degree approach, again, depending on climate, Tower C is seven to 10˚, and Tower D is nine to 12˚. We looked at it both with a Path A and Path B chiller initially when we did the runs for 90.1, and you'll see in our analysis, we initially did it with both A and B, but, again, we had the same problems with the curves that we did with the chillers, and we've now summed up the analysis for the Path B, which is consistent with our chiller proposal for prescriptive.

So 12 towers, low, medium and high efficiency, a range of 45 to 100 gpm per horsepower, and four approaches going from roughly five to 12 degrees. And then we got contractors cost, freight on board to job site, from a vendor, and added 28, almost 29 percent contractor's mark-up, that's for the general and for the sub, and 50 percent installation on top of that for installation cost premium to account for structure, drayage, all those other things. Here are the models that we had, these were all from one manufacturer, VAC, but towers are fairly similar, in fact, there's one tower model in all the programs, one set of curves that fit all of the towers, and we did a research project as part of the Cool Tools project that actually took as much data as we could get for manufacturers and showed that all towers really do collapse down to one set of curves if you look at it right. So, you can see the range of pumping horsepower, so that was taking into account each of the models does, in fact, include the specific penalty for head off of the cooling tower, the motor sizes that are off there, and the nominal gpm per horsepower at the rating condition of 95/85/75. The tower names, L, M, and H are Low, Medium, and High efficiency, and 01 to 04 goes from a High range to a Low range at 01, the closest range to a high range approach. Let me try it again. 01 to 04 are the approaches, bigger horsepower means that you could drive closer to the web bulb, so 1 is a close approach, and 4 is a high approach.

So I'm going to show you the same eight climates in four slides, so these are groups of two climates. You can see Climate Zone 3 and Climate Zone 6, all 12 towers for each climate, and the cell that is red is the lowest lifecycle cost, including the tower cost, with the markups I mentioned earlier, and the energy costs, the TDV over a 15-year life. So, those are the last three columns. And if you're interested in how the actual plant ran, here is the individual kilowatt hours per year for each of the components – chiller, tower, chill water pump, condenser water pump, total, and then this is a converted 15 years' worth of present value TDV energy cost. Tower installed cost, including the markups that I mentioned earlier, and these two together are the net present value of the lifecycle cost. So, the lowest lifecycle cost per climate in each case is HO4, which is the high approach, high efficiency tower. So, this is 3 and 6, same results, this is 7 and 8, exact same results, this is 9 and 10, exact same results, this is 12 and 13, exact same results, so, again, these eight climates represent 85 percent of the construction activity in the Dodge [ph.] database.

These are runs that we did earlier as part of that 90.1 study that we were doing, but for the Oakland climate, Climate Zone 3, and we found that when we expanded the data and the hours of operation, basically taking the energy here and multiplying it by four, that although office buildings appeared not to want to have a close approach, a data center, or other 24/7 facility, because of the increased energy use for the same fixed costs of the tower, it is justifiable to have a lower approach. And the current requirement is written as based on this preliminary analysis and we need to re-do this analysis specifically for data centers.

Okay, later today there is a presentation on cooling tower water usage and there will be information on the cooling tower market in that presentation. Here is a proposed co change – changes. I was unable to do a nice underline and strikeout for the table 112g here on the slides, but basically we're adding the closed circuit fluid coolers and you can see the efficiencies there, they're about half that of the open towers and, again, I mentioned earlier, there is a new standard that we have to reference, the CTI ATC-105S, and that's in the report.

And then prescriptively, this is all the stuff that's under 144(h) for cooling towers, and we're not changing anything on the fan speed control. We are recommending, again, an increase of the minimum flow per cell in response to issues the industry has had and raised. We're striking out all of this stuff about the limitation of centrifugal fan cooling towers and just making it an efficiency requirement based on gpm per horsepower, so if somebody comes up with a super efficient centrifugal fan cooling tower, that's fine, as long as they meet the 80 gpm for horsepower. And then there's an exception for new towers added to an existing condenser water system, so this is to address the fact that, if you've got an existing footprint, you've got an existing structure, we're not requiring you in a retrofit to put in this really big tower that may or may not fit where the other towers are.

For 24/7 facilities, we're requiring a maximum approach of 5˚, a design condition, and again, an exception for new towers added to existing condenser water systems. The letter from TC 8.6 is in its entirety attached to the case report, but here is again a synopsis, just like we did with the chiller letter. There is a negative impact on product offering, I think specifically they said 100 gpm for horsepower was represented only by 10 percent of the products; and in recognition of that concern, we dropped the maximum efficiency from 100 down to 80; may drive the market to less efficiency systems, and unlike 90.1, we have a prescriptive limit on air cooled chillers. That's important because, again, as I said, most of the country, the rest of the country, at least from the sales data we've seen from the HARI, is that the sale of air cooled chillers, which are the less efficient, are on the rise, and water cooled chillers are going down. So, we don't want to drive people to that market.

There was some comment from 8.6, as well, that they're worried about competing with Package DX, but in our experience, in the facilities we've designed, people pretty well know based on the footprint of the facility, and how they want to operate the facilitate, whether they're going to go with a bunch of package DX units, or a chill water system. Some buildings, you can't get by with anything but a chill water system, and we do lots of data centers, high rise buildings, very hard to do with package DX, although water source heat pumps, or water source air-conditioning units can be used.

Increased customer cost was raised, but, again, that was accounted for in the lifecycle cost analysis, so we're following the TDVs and lifecycle cost criteria that all the other standards are.

Increased footprint – so, again, we're putting this increased efficiency into the prescriptive setting the bar for the performance of the building, and if you've got a limited footprint in your facility and it's a new facility, then you can go with the performance approach and get by with a smaller tower, but make up the energy elsewhere.

They mention that they thought that these larger towers would require more sophisticated controls, but, really, a tower is a tower, it's a fan, it's either got stage fans or, 99.9 percent of the time, it’s got a variable speed drive. And it doesn’t matter what the gpm per horsepower is, it’s exactly the same in terms of controlling the fan.

And then they were concerned about water loading, in particular, getting dry spots in the fill when you get a dry spot in the fill, the air rushes through, flash evaporates the water, you end up with a lot of crud on the fill, the fill doesn’t work very well, and you have a deep degradation in the heat rejection. And, in part, to offset this, we’re proposing to increase the turndown to 50 percent, so you don’t want to end up with that ragged edge of where the fill – of where you can go with the tower, but make sure that you keep all the media wedded.

And one of their comments was, if you’re going to have an efficiency, you should also have a maximum approach, and as we saw in the analysis, it really was not borne out by the analysis for office buildings, however, it does appear to be borne out for 24/7 facilities.

So, we have to complete this for the rest of the climates and we’re going to repeat at least a couple of cases for single-phase chillers just to make sure that it’s not completely different than what we got for the variable speed chillers. And then, there’s a whole bunch of background material through the models that we used, the cooling tower models, the chiller model, and the overall top model procedure. So, with that, I'll open it up to questions.

MR. SHIRAKH: Any questions on Cooling Tower Efficiency from the audience in the room? How about for those online?

MR. HYDEMAN: Yeah, for those of you online, if you didn't hear earlier, please either raise your hand on the system, or type in the chat window and we'll unmute you. Go ahead.

MR. SHIRAKH: And please identify yourself and your affiliation for our Court Reporter. Thanks.

MR. YASNY: Jamy Bacchus is not on the phone, but he typed in, and I don't know his affiliation –

MR. HYDEMAN: NRDC.

MR. YASNY: He typed in, "Mark, what is the percent of CT products which meet or exceed the 80 gpm hp and has percentage of available products three years before Title 24 became effective ever been used to rule out or include a measure in Title 24?"

MR. HYDEMAN: I don't have the data; there are a number of cooling tower manufacturers on the line, does anybody want to take swag at that? I will tell you that cooling towers, again, are fairly simple devices, they have basins typically or spray nozzles, they have media that get wedded, and then they’ve got fans. And the way you increase the gpm per hp, is you drop the hp on a box, so, for a given box size, you just drop the hp until you meet the gpm per hp rating, meaning you have more media for the hp and the fan, so I think that almost any box could meet the 80 gpm per hp, at least propeller fan towers, with a small enough motor, but I’m not going to say that categorically.

MR. HEGG: Gentlemen, this is Trevor Hegg, Evapco, chiming in. Can you guys hear me?

MR. HYDEMAN: Yes.

MR. HEGG: All right, just want to make sure because our notification, it doesn’t look like we’re on the line, but one of the things, we’ve done an analysis of our product line and the 80 gpm level, or the original 30.2 allowed all the 80 product line, by the requirement, going to 80 gpm lowered it only to 28 percent of the line to satisfy the requirement. So there is going to be a significant limitation on what products are available to meet this efficiency standard. We agree that the efficiency standard should be raised, but we kind of question whether 80 gpm is still maybe pushing a little too much in tying the hands of the designers and engineers who are coming up with these facilities.

MR. HYDEMAN: Okay, thank you, Trevor. I'm just curious, do you have a rough sense of where things would land if it was 70 or 60 gpm, or – even if you don't –

MR. HEGG: Yes. At 60 gpm, 60 percent of that product line would meet the requirement, would satisfy the requirement. That's to say 60 percent of our product line meets 60 gpm per hp. And 42 percent meets 70 gpm per hp.

MR. HYDEMAN: Okay, thank you very much. Any other comments?

MR. YASNY: And anyone that is not commenting, can you mute your phone so, if we unmute, we won't hear your papers rustling?

MR. HYDEMAN: Go ahead, John.

MR. MCHUGH: Thanks, Mark. This is Jon McHugh with McHugh Energy. And, Mark, I have a question about, you know, we've seen over time for the various requirements for variable speed of equipment, you know, that those thresholds have come down over time. I guess it's been a number of years now for cooling towers. I was wondering, have we hit that time where basically the thresholds aren't needed anymore for variable speed?

MR. HYDEMAN: Jon, you're specifically referring, I assume, to the 7.5 horsepower --

MR. MCHUGH: Yeah.

MR. HYDEMAN: -- threshold. I think we could probably certainly drop it to 5 horsepower and possibly just get rid of it altogether, and we’ve got ECM and brushless DC motors, I don’t know what the manufacturers are offering now in those areas, but we’re seeing this in other areas like Kraw units, air handling units, you know, like Hunt Air with their direct drive fans that are variable speed, Stolz, others, so it’s quite possible. We’re looking at this elsewhere in the standard to require variable speed on everything, so it’s quite possible that those will drop out. Jeff, do you want to comment on – where is that variable speed requirement? Is that – aren’t we doing a separate measure on variable speed? Okay, so that will be covered this afternoon and, certainly, if it gets covered, it will cover all products, including this.

MR. MCHUGH: Then, another question related to product lines, I understand that there might be product lines with, you know, different matching of fans to amount of media in a tower, but is there anything that indicates that, you know, this is the current mix, you know, if there is a new rule, that you would expect that the mix would change. I mean, is there any technical obstruction to going to something like, you know, 80 gpm per hp? And, I guess, this I guess goes to probably some of the commenters on the phone, so what sort of limitations are we running into if people basically redesign their product line basically for the same size fill, for using smaller fans, and for larger capacity, if they increase the size of their fill?

MR. HYDEMAN: I think that was an open question for those that are online that are actually manufacturing these products.

MR. YASNY: Anybody online want to answer that question?

MR. HEGG: This is Trevor Hegg again from Evapco. I'm not sure that I had it fully – there are limitations with respect to the design. In our current product line, the new [inaudible] product line, adding more fill, you can look at it in a couple different ways, which I know one of them was to look at increased plant area, and the concern was already proposed that, as you increase the plant area, bottom loading, which has the potential for scaling, spinning, just inefficiency within the unit, which is there and I know there can be ways to address that, but in terms of going vertically, there's sometimes the limitation that going vertically doesn't really add much more anyway, that, as manufacturers, we've stopped our designs because the added capacity [inaudible] not cost effective in terms of the increased capacity for doing so.

MR. YASNY: Can everybody mute their phones if you’re not talking? Everybody online, please mute your phone? Thank you.

MR. HYDEMAN: Okay, thank you, Trevor.

MR. MCHUGH: And then, one last comment, which is, in your analysis, are you finding that basically 100 gpm was essentially the minimum lifecycle cost, or basically you ran it to what you felt was feasible within the market, and that even if you had gone even further, lifecycle cost would drop even more?

MR. HYDEMAN: We’re kind of running out on the edge of what we’ve seen as being feasible on real structures, you know, we went out, for instance, on a data center recently, up in Napa, on an open bid, and everything was based on lifecycle cost, and we ended up with a tower that was in the range of 90-100 gpm per hp, but we didn’t get many bids that were more efficient than that, so I think we’re kind of running to the edge of the product.

MR. MCHUGH: Thank you.

MR. SHIRAKH: Any other questions related to cooling tower –

MR. MORRISON: It’s Frank Morrison.

MR. HYDEMAN: Hi, Frank.

MR. MORRISON: How you doing, Mark? We certainly have similar numbers to Trevor, and basically the answer is, you know, you're going to go up, increase in plan area for the tower, which is you're going to occupy more space, and I know from your slides here, you include extra costs for that extra space and grillage and the like. Will you be releasing your study here, so we can have a little closer look at it?

MR. HYDEMAN: Yes, I've got a draft case report that I will give to the Commission and they will be posted in the case reports. And, Frank, I'd be glad to just shoot you a copy directly, as well, since I know how to find you.

MR. MORRISON: Yeah, you're kind of breaking up. I mean, apparently some folks aren't on mute, so it's hard to hear you.

MR. HYDEMAN: Yeah, what I said is, yes, the Commission will have a copy of both case reports for the chillers and cooling towers, in fact, it's already there, it's online. I'm getting a nod from Mike McGaraghan because I sent them out last night and apparently they got posted. So, you can get to it, I think, either through the Commission website and, Mike, is it also on the – is there a link on the case website? Okay, so if you have trouble finding it, Frank, just shoot me an email and we'll get you a copy, and you guys are welcome to look at the models, as well. Everything is transparent.

MR. MORRISON: I have another question, Mark.

MR. HYDEMAN: Yes.

MR. MORRISON: The air cooled vs. water cooled chiller data, do you have that specifically for California from HARI?

MR. HYDEMAN: No, I don't. The air cooled vs. water cooled is the same stuff that was presented to us; this has to do with kind of a number of units, I think, I remember Dick Lord was the one that presented the data, but it was presented to 90.1 and it's a couple years old now. You know, Frank, in all those seminars we did on air cooled vs. water cooled, there's been two of them, at least, that were sponsored by 8.6, that graph was in those seminars, so if you need me to pick up a copy of it, I can send it to you.

MR. MORRISON: [Inaudible] for California, to evaluate the air cooled vs. water cooled limitation that you have in your Title 24.

MR. HYDEMAN: I don't have any data that is California specific.

MR. MORRISON: Could we ask for that from HARI?

MR. HYDEMAN: I would love to have that data, so knock yourself out. I asked for all sorts of things from HARI, and I can't say they've been extremely cooperative, but I think, coming from a TC or, you know, an industry member, you're more likely to get that data than I would be.

MR. MORRISON: Okay, because our concern, you know, as Trevor had said earlier, is these towers are going to get larger, obviously, and when you lower your horsepower, you're also dropping your horsepower – gpm per hp, or raising your gpm per hp, but you're also lowering the absolute tonnage of the bottom, but your costs are going to go up and that's our concern, vs. other systems that are not having similar efficiency requirements mandated to them.

MR. HYDEMAN: Yeah, I share your concern, Frank, but, again, it's cost justified and if people are really looking at lifecycle costs when they're designing a chilled water plant that is water cooled, they would be putting in larger towers. And so I guess the real question is, are we going to shift the market, and I don't know if there's any evidence that we have shifted the market.

MR. HEGG: This is Trevor again. And then the other part of it is, in addition to air cooled, what about forced draft? If the numbers of forced draft become – I guess the concern becomes the goal of energy efficiency, which everyone agrees with doing this and, again, increasing the level, but if nothing is being done on the forced draft or air cooled side, because it potentially – and, again, our gut feeling would be that this opens up the door to other technologies like the forced draft or the air cooled if the market for the induced draft, which are more efficient lines, becomes more limited.

MR. HYDEMAN: But, by forced draft, are you talking about closed circuit fluid coolers?

MR. HEGG: Forced draft towers.

MR. HYDEMAN: Are the forced draft towers excluded from the CTI ATC test standards?

MR. HEGG: No, I mean, induced draft and forced draft cooling towers are all part of the CTI certification program, and as I recall there is – yeah, a centrifugal fan open cooling tower has a performance requirement, 20 gpm per hp.

MR. HYDEMAN: Right, so my point is, so you're talking about the difference between centrifugal and propeller or axial?

MR. HEGG: Well, yeah, exactly, in that if we're increasing efficiency on the propeller or axial fan type, which limits the number – or reduces the availability of the induced draft product like that, for the propeller product, does it open the door for more forced draft, which is actually a higher energy consumer?

MR. HYDEMAN: Yeah, so we actually have a limitation on centrifugal fan cooling towers existing, you can see it up there on the screen right now, and if we go to the minimum efficiency of 80 gpm per hp, they'd have to meet that, as well. So, you know, again, I mentioned this earlier, I think we're protected about moving people to forced draft, I just kind of had this like little epiphany here is, do we have a loophole for closed circuit fluid coolers? Which would be horrible, and I think we need to address that, somehow. We're covered on air cooled chillers because we have a prescriptive limit on air cooled chillers, we talked about it this morning. We are covered on the forced draft towers because we have this efficiency target that is not specific to any technology, therefore, you know, if they had forced draft met, that's fine, but very unlikely that they would be able to. But, it seems to me we need to do something to prevent people using closed circuit fluid coolers.

MR. MORRISON: Mark, this is Frank Morrison, again. Closed circuit cooling towers are generally much much more expensive than open circuit because, really, you're combining a heat exchanger and a tower together. So what you're looking at there is comparing it, say, to an open circuit tower and a plate and frame heat exchanger. So, I think there are more apples than oranges. The biggest market for the closed circuit is water source heat pump loops.

MR. HYDEMAN: Yeah.

MR. MORRISON: And then, the occasional chiller plant that's next to the concrete plant where you don't want that concrete dust to get into your chiller barrel, you know, it's very rare you ever sell one on a chiller plant.

MR. HYDEMAN: Thank you.

MR. MORRISON: But to Trevor's point about the centrifugal – you've taken out the exception for sound inducted installations, which happens occasionally, or there's a need for very very little sound on a centrifugal, or someone wants to put the tower inside for either security reasons, or cold weather climate, which may not occur that much in California, but there are areas where that does happen.

MR. HYDEMAN: Okay, so what you're suggesting is, if we do have this efficiency requirement at 80 gpm hp, the exceptions that we had previously for centrifugal fan cooling towers should probably be an exception to this requirement.

MR. MORRISON: I would recommend that. I don’t know how Trevor feels, but you might want to chime in, Trevor.

MR. HEGG: I think that’s what we were kind of getting to, is that there’s – to me, there’s got to be – the verbiage has to be considered – we’re trying to promote energy efficiency and there are obviously situations where installation is going to require forced draft, as you mentioned, Frank, and they have to be exempt. I guess one of the things is, is the gpm per hp to both propeller and to centrifugal fan?

MR. HYDEMAN: As I wrote it here, it is any open cooling tower. So, basically it says, to meet the prescriptive requirements, you have to have a tower that is rated at 80 gpm per hp, or higher, or you go the performance method. So, it covers both technologies. Any other questions?

MR. LINDAHL: Can you hear me?

MR. HYDEMAN: Yes.

MR. LINDAHL: This is Paul Lindahl from SBX.

MR. HYDEMAN: Hey, Paul.

MR. LINDAHL: How you doing?

MR. HYDEMAN: Good, thanks.

MR. LINDAHL: I’ve been trying to talk here for a while, and I don’t know what was going on, but apparently no one could hear me. I wanted to agree with Frank’s point and Trevor’s about needing to have exceptions for some of the peculiar applications that require extremely low noise and potentially also indoor locations because some facilities are constrained for security reasons from having the towers accessible to where somebody can do something to them. So, I think it’s important to get that in there, somehow. Also, I wanted to point out that I posted a couple of questions on the chat and I see there are some other questions that people have on there, as well. I have a concern about the 5˚ maximum approach for 24/7 facilities, not because I’m worried about 5˚, but I’m worried about anything less than 5 because it’s outside CTI certification. CTI certification only goes to 5˚ and, from a practical standpoint, performance testing is essentially limited in the marketplace to a minimum 5˚ approach. As you get to a lower approach temperature than that, when analysis is done for the test, the impact of any potential errors in measurement is magnified drastically for every degree lower you go in approach temperature, the magnitude of any measurement errors goes up exponentially.

MR. HYDEMAN: It’s like a signal to noise ratio, that you’ve got a tenth of a degree or is bigger over one degree than it is over five degrees.

MR. LINDAHL: The degree of difficulty in a zero approach is just infinity. So that's a difficult thing. I mean, it can be dealt with, but – and what people typically do when they need to have an approach less than five, so that an analysis can be made to validate performance if there is a performance test, is to go to a lower wet bulb where the approach is five, and the equivalent approach at the design wet bulb could be less than five, if you follow me, you're sliding down the performance curve to a lower web bulb, where the approach is five. And that's not a problem for the manufacturer because, again, you're analyzing back to an approach that is doable. You wouldn't accept any measured approaches less than five, though, during test measurement because of the measurement error impact, it could delay doing a test. Do you follow where I'm going there?

MR. HYDEMAN: Yeah, no, that makes sense and, again, as I mentioned earlier, all of our research when we were looking at cooling tower data, which all of you participated in, this has got to be over 10 years ago, you know, we showed that you could take one set of model or one set of curves and fit it to almost all towers. So, obviously, you know, you could get an alternate test point that represented that this tower – or extrapolated that this tower could do the approach of five degrees or less at the design point. So, I guess, do you have a specific recommendation on the approach? I mean, you guys asked me to put in a maximum approach somewhere; we didn’t find that, at least within the range of towers that we looked at for the office building, that one was justified, but –

MR. LINDAHL: Yeah, that’s a separate issue from what I’m talking about. What I’m concerned about is that this will encourage people to ask for design points less than a five degree approach on 24/7 facilities. The issue of a maximum approach is something that would have to be proven out in modeling and it sounds like, for office buildings, at least in the climate zones you’re looking at, it doesn’t make sense. I’m not sure that would be true in office buildings with higher wet bulb – average web bulb – but, Frank, or Trevor, do you have any thoughts on that?

MR. HEGG: Well, the only think I would add is that, in a sense, the maximum five degree in design condition, it really, if the goal is to have a CTI certified product, then that line can’t read maximum, it would say it has to be designed for a five degree approach, because anything less is non-certified.

MR. LINDAHL: Right, and that’s a good point. It basically says you can't go below a five degree approach unless you shift the design wet bulb down.

MR. HEGG: Correct.

MR. MORRISON: Yeah, this is Frank Morrison. The other comment I have is the five degree approach is a different degree of difficulty, depending on what the wet bulb is. Mark, at the last 90.1 meeting, I talked with Steve Taylor about this with Addendum – I think it is C.I., we had come up with a formula for that, so you had the same degree of difficulty and you can establish – let's say you want it 95/85/78, with a seven degree approach, you can have, then, an equivalent approach at other wet bulbs. Does that make sense?

MR. HYDEMAN: Yeah, no, no, so you prefer this at a different test point, I mean, that's exactly what Paul was saying.

MR. MORRISON: Yeah, a fixed approach, I think, is impractical because the five degrees at 66 wet bulb is certainly different than a 74 wet bulb, you're going to get a totally different size tower and maybe not something that you were looking for.

MR. LINDAHL: The five degrees really has to do with certification and with actually running a performance test. It's a different subject from, you know, the equivalency of a duty, what Frank is talking about.

MR. MORRISON: Right.

MR. HYDEMAN: It seems to me, Paul, you know, to answer your question, you said that you could come up with another test condition that had a higher approach at a different wet bulb, so if we decided to move forward with a maximum approach at all, we could do that at a specific test condition, right? Because you could rate your box at CTI rated certified rating, at some other condition, and show that the approach was, in fact, you know, seven degrees or less.

MR. LINDAHL: At an equivalent duty condition.

MR. HYDEMAN: Yeah.

MR. LINDAHL: Yeah, I'm not sure how you'd word that. We need to think about that a little bit because that's where it needs to go, something like that.

MR. HYDEMAN: Jeff.

MR. STEIN: This is Jeff Stein with Taylor Engineering. Maybe one way to do it is just to say – to list the design condenser water temperature for different climate zones or different wet bulbs, just spell it out for them and say, "In Climate Zone 3, your design condenser water temperature shall be," you know, "…72," or something.

MR. HYDEMAN: But the issue with that is that, if we found that the right approach is five degrees, you're right on the hairy edge of the CTI testing and the issues that Paul was talking about, so it may be better to just have a test condition at which the approach has to be X or less, that's equivalent to five degree in any of these given climates.

MR. STEIN: I mean, certainly for enforcement, it would be easier if you just told them what the –

MR. LINDAHL: Yeah, that's what I'm worried about is wording it in a way that doesn't lead to immense confusion in the field.

MR. STEIN: If you told them what condenser water temperature had to be, then there would be no confusion.

MR. LINDAHL: I have to think about that.

MR. HYDEMAN: All right, well, let's take this offline, we all know how to communicate with one another and I think, you know, there's a question in my mind if we want to do this at all, have an approach, I mean, we're pushing the envelope as it is just to put in a gpm per hp, that's more aggressive than what's in the base standard. And at this point, I would be willing to back off on the approach if we can't come up with some wording, but I'm happy to work with 8.6, all of you that are on this call, to work on some wording if we can come up with a consensus, that would be great.

MR. LINDAHL: We'll help on that, Frank, and the rest of us on the working group. The other question I had was related to the analysis and, as I listen to you talk about using BFDs or BSDs, I guess you're calling them, and using one or two cells on a one or two chiller system, it occurs to me that you're getting probably into some pretty low exit air velocities from the cooling tower. As the speed goes down, the air speed leaving the tower goes down and if you're not including an increased recirculation on the tower in your analysis, then your results are optimistic on energy savings because there's going to be more recirculation. What that means is, instead of the wet bulb being 55 degrees, it probably should be analyzed at 56 or 57 in some cases, you know, it's going to vary a lot with the speed. But you could do a kind of prediction of that based on an assumed wind speed.

MR. HYDEMAN: Yeah, that's – well, it's kind of beyond the –

MR. LINDAHL: The thing is it could actually be fairly significant in the energy analysis.

MR. HYDEMAN: Yeah, the question is –

MR. LINDAHL: Because it changes the approach.

MR. HYDEMAN: The question is how many hours are you running it at those very low fan speeds where that becomes –

MR. LINDAHL: No, that's where your judgment is going to have to kick in because I know that that would make the energy analysis optimistic.

MR. HYDEMAN: So, let me put this back to you. Do you guys have a sense of where this becomes a problem in terms of percent rated flow? Because we could easily handle it by putting a floor on the speed of the fan – for modeling purposes.

MR. LINDAHL: I would be happy to share some published recirculation data as a function of the exit speed, and the recirculation goes up pretty fast as the exit speed goes down.

MR. HYDEMAN: Okay, why don't we start there and, again, we can take this one offline, but –

MR. LINDAHL: Okay.

MR. HYDEMAN: Yeah, I doubt it's going to make a whole hill of beans because, of course, you get this kind of cubic relationship of speed and energy, so –

MR. LINDAHL: That's the thing that disturbs me about it –

MR. HYDEMAN: But I'm thinking you would probably end up with the same results, even if we put in a 30 percent minimal speed on the model, but I would be happy to do a test run, but let's figure out what that number is.

MR. HEGG: Well, I don’t know if it’s necessarily speed. I think, to Paul’s point, is that it’s somewhat – it’s velocity out of a stack because I think, with the way this is going, to suggest that we’re going to go with larger footprints that have lower horsepower’s, we’re going to be in situations where the exit velocity off of a tower is still going to be low enough, even at – or potentially low enough at 100 percent fan speed that you could run into recirculation issues.

MR. LINDAHL: Right, that’s what I was talking about, Mark, it’s not just slowing it down from the design, you’re starting with the slower speed at air speed of design, and then you’re slowing it down. So, recirculation needs to be considered in the analysis. It’s pretty different from tower – say you have a tower with a 2,500 foot a minute discharge velocity at a 75 horsepower and you drop the horsepower to 40, or 30, and you’ve got a discharge velocity that’s now, you know, 1,000 or 1,100 feet per minute. And if you start there, that’s about the same speed as the design wind speed for a cooling tower, which is 10 miles an hour, so you’re getting to where the tower is going to be affected by the wind a lot more.

MR. HYDEMAN: Okay, well, I hear you, understand the issue, and obviously we've got to chew on this a little bit more.

MR. LINDAHL: Okay.

MR. YASNY Good, Mark, it looks like there are about three people online that have questions. The first one is Jamy Bacchus. Are you on the line?

MR. HYDEMAN: Jamy talked earlier – or, no – yeah, he did.

MR. YASNY: Okay. Then also Mick Schwedler.

MR. SCHWEDLER: Hi, this is Mick. Can you hear me now?

MR. HYDEMAN: Yeah, we can hear you.

MR. YASNY: A little louder, please.

MR. SCHWEDLER: Okay, and I apologize, I couldn't get on earlier and I don't have the report in front of me. When we're doing the water cooled to air cooled comparison, what costs were added for the building for the water cooled chiller, as well as the requirements to meet standard 15 for that mechanical room?

MR. HYDEMAN: I'll let Jeff answer that. Mick, this goes back to the 2005 standard –

MR. SCHWEDLER: Uh huh, but I'd just like to have a handle on the costs.

MR. STEIN: Yeah, this is Jeff Stein. We actually used a real project, one that we designed and got some pricing from contractors both ways, so we felt like we captured the full cost of the entire system. Again, as Mark said, this is going back, I don’t know, six to eight years now, so I don’t have all the details at my fingertips, but we definitely tried to capture, you know, the true costs. I mean, we had water costs in there to – so we –

MR. HYDEMAN: I recall there was a cost for the structure, actually a cost for the chiller room, there was a cost for the refrigerant system, and exhaust fan. So, you know, again, this got a tremendous amount of scrutiny because of all the manufacturers really pushing air cooled chillers at the time and everyone that received copies of the report and all of our data, which we handed out freely, nobody could really come back and say, “Oh, you forgot X, Y, or Z.” It was pretty thorough, the costing on it, Mick.

MR. SCHWEDLER: Okay. I just think it would be helpful when practitioners start looking at this and balking against it, for the Commission to have those costs available so they can say, “On this size building, this sized unit, here was the additional costs that were done through contracting,” so people have an idea of what costs were assumed. Thank you.

MR. HYDEMAN: Yeah, and Mick, the report, as I said earlier, is still up on the CEC website, so if you go to Title 24 and you end up on the California Energy Commission website, you can go back to the 2005 standard. You know, easier than that, I'll dig up the report and send it to you.

MR. SCHWEDLER: Thanks, Mark.

MR. HYDEMAN: Sure.

MR. YASNY: And then Richard Lord has a question.

MR. LORD: Yeah, can you guys hear me again?

MR. HYDEMAN: Yes.

MR. LORD: Okay, yeah, just to clarify on the volume data, that was the data I gave Mark, and that volume data is for all of the U.S., it's not for California. The distribution of chiller sales in California is a lot different than the rest of the United States. You know, a lot of the chillers are actually sold in Zone 1A, 2A, 3A, and 4A; in fact, 50 percent of them are sold there. They're a lot less in California. So, I wouldn't take that data that I gave Mark and use it to extrapolate it to California. Unfortunately, you're not going to be able to get any better data on ARI because ARI does not keep data by state. I have some internal methods I use to get to it, but I don't know if I could get to air vs. water cooler, I'll have to take a look at it, but be careful using that data because it is the national data.

MR. HYDEMAN: Well, Dick, I'll just make sure every time somebody asks for the data that we make it clear this is national data, and may not represent California.

MR. LORD: Yeah, that would just be a good thing to caution them with.

MR. HYDEMAN: Good. Any other questions? So we've got some work and stuff we'll do offline between myself and the TC 8.6 Working Group, and appreciate everybody's time.

MR. SHIRAKH: This may warrant another round of stakeholder meetings after this workshop. We'll try to organize that. Any other questions on any of the topics presented this morning in the room or online? So we're a little bit early, about 10 minutes. We'll adjourn for the morning and we'll be back at 1:00, and we'll be talking about Cooling Tower Water Efficiency. Thank you.

(Adjourn at 11:53 a.m.)

(Reconvene at 1:07 p.m.)

MR. SHIRAKH: Let's start with the afternoon session. So, this afternoon, we'll be talking about Cooling Tower Water Efficiency and Erica Walther will represent that, then at about 11:40 [sic], Automated Demand Response for Nonresidential HVAC, followed by

Single Zone VAV Fan Control and Integrated Economizers, and then Reduce Reheat; after that, HVAC Controls and Economizing, and then we'll finish with Air Compressors about 4:00.  So, Erika, are you online?  Can you hear us?

MS. WALTHER:  I'm here.

MR. SHIRAKH:  So, why don't you start?  Thank you.

MS. WALTHER:  Okay.  I'm not seeing my presentation, I'm seeing the Automated Demand Response right now.

MR. YASNY:  What do you need?

MS. WALTHER:  I'm doing Cooling Tower Water Savings.  There you go.  Great, perfect.  All right, well, thanks for those of you who are on the line and at the meeting, I am Erika Walther at Energy Solutions, and I'm going to be presenting on Cooling Tower Water Savings.  Go ahead and advance the slide.

So, I'm just going to go over the measure scope which hasn't changed since the last several meetings that we've had, and the proposed Code language, and the analysis and the lifecycle cost results.  And there have been slight changes to all of those things, based on the last stakeholder meeting and, right up front – and I'll point this out again when it comes up – but there has been a change to the drift eliminator requirement, and also I did add an assumption about the water rate increase projected over the lifetime of the measure, which was not in there before, so that was beneficial, of course, to the lifecycle cost analysis.

So, the measure scope, once again, is due when replacement of evaporative cooling towers in the commercial, industrial, and institutional sectors. And as we talked about last week, this applies to towers that are 150 tons and larger, based on cost-effectiveness.

There are five measures in the proposed Code change, the installation of the conductivity or flow-based controller, documentation of maximum achievable cycles of concentration, based on local water quality, installation of a flow meter on the make-up water line, installation of an overflow alarm, and installation of drift eliminators.

So, I’m beginning here with the Code language for Part 6 and, again, this hasn’t changed since our meeting last week, except in the case of drift eliminators and maybe we’ll just – well, people can make comments on that now, or we’ll tough on it again when we go over the analysis. So, we might want to just leave the comments for the Code language, limit those to Code language comments for this first part, Section 112 Mandatory Requirements for Space Conditioning Equipment, Section E, Evaporative or Open Cooling Towers. For all Evaporative or Open Cooling Towers, they shall be equipped with the following: For conductivity or flow-based controls, towers shall include installation of controls that maximize cycles of concentration based on local water quality conditions. Controls shall automate system bleed and chemical feed based on conductivity and/or in proportion to metered make-up volume, metered bleed volume, or bleed time, and conductivity controllers shall be maintained in accordance with manufacturer specifications to maximize useful life and accuracy.

Regarding Documentation of Maximum Cycles of Concentration. The building owner shall document the maximum cycles of concentration based on local water quality conditions, using the Energy Commission provided calculator. The calculator determines maximum cycles of concentration based on a Langelier Saturation Index of 2.5 or less. The building owner shall document maximum cycles of concentration on Compliance Form MECH 5C, which shall be reviewed and signed by the professional engineer of record.

Flow Meter. Towers shall include installation of a flow meter on the makeup water line overflow alarm. Towers shall include installation of an overflow alarm to alert operator to sump overflow in case of makeup water valve failure. Overflow alarms shall send an audible signal or provide an alert via the building management system to the tower operator in case of sump overflow.

Drift Eliminators. Towers shall be equipped with efficient drift eliminators that achieve drift production of .002 percent of the circulated water volume for counter flow towers and .005 percent of the circulated water volume for cross flow towers.

In the Compliance Manual, you would add Section 4.2.4 Cooling Tower Controls under Section 4.2 Equipment Requirements. Section 4.2.4 would reference Section 112 in Part 6 and describe the methodology or tool, which is the calculator that I just mentioned, required to calculate maximum cycles of concentration in cooling towers, based on local water conditions. And it will reference the appropriate compliance form.

Under Section 4.5, HVAC System Control Requirements, Section 4.5.1 Mandatory Measures would require the addition of number 7, Cooling Tower Water Savings Controls. Language would be developed that references Section 112 in Part 6, and describes the methodology or tool required to calculate max cycles based on local water conditions and references the appropriate Compliance Form.

Section 4.10, Glossary Reference. Add Subsection

4.10.11, including Water Balance in Evaporative Cooling Towers, Cycles of Concentration, and Langelier Saturation Index.

For MECH 1C, the Certificate of Compliance, we would add a new Section to HVAC Equipment Efficiencies, Section 112 in the Note Blocks for Mechanical Mandatory Measures. The section will require verification of installation of the following: controls that automate blow down and chemical feed based on conductivity and/or flow rate, and/or bleed time, flow meter on the makeup water line, overflow alarm to alert operator to overflow the sump in case of makeup water valve failure, and drift eliminators. And, again, compliance would be just based on confirmation that the drift eliminators are installed, not that they are achieving a particular efficiency.

MECH 5C, Maximum Cycles of Concentration. This would be a new form on which the responsible party would document maximum achievable cycles of concentration, based on local water quality conditions and where they would record the local water quality data, would be inserted as Subsection 4.11.8, or be added to the end of Section 4.11 as subsection 4.11.10.

Just to review the measure costs in the Useful Lives of the Measures in this proposed Code, the individual measures that are presented here are not discounted, but on the bottom line there, I have the total cost as a present value. So, the methodology for calculating the savings associated with conductivity and flow based meters began with modeling the tower energy load, and we used EnergyPro to do that, we modeled nine building climate zones, which represent 89 percent of the projected new construction for office space. The building had 117,000 square feet of condition space, it had the load profile of an office with cooling operation from 6:00 a.m. to 6:00 p.m. seven days per week, chiller capacity ranged from 239 to 292 tons, depending on the building climate zone. The cooling tower capacity ranged from 280 to 339 tons, depending on the building climate zone, and the condenser water flow rate ranged from 691 to 845 gallons per minute, also depending on the building climate zone. And the outputs that we got from EnergyPro that we were interested in, that we then plugged into a water savings model included the outdoor air dryable temperature of the outdoor air wet bulb, the condenser water load, and the chiller load. Next slide.

Then, we modeled the tower water use, and we did this in an Excel-based model that had been provided by a tower manufacturer to Taylor Engineering, who had used it for their own purposes, and modified it, and had been using it, and then offered it to me to use to model this, and it allows manipulation of cycles of concentration and drift, and the output is the annual lead rate, among other things, but I was interested in the annual bleed rate. We developed a model for each of the nine climate zones, set drift at .005 percent. I did one run for each climate zone at 3.5 cycles, which is our baseline scenario, and one run was 4.9 cycles, which is our maximum statewide average cycles of concentration, based on the implementation of the measures – proposed measures. I wasn’t quite finished with that one, if you could go back, great. I calculated the bleed savings for each building climate zone and then calculated a weighted average according to the new construction projections, and then I scaled the results to represent a 350 ton tower for the lifecycle cost analysis.

In terms of documenting the maximum cycles of concentration, this doesn’t require any installation of anything, but I assume two hours of time to gather the local water quality data, enter it into the calculator, and document it in the required form, and this measure is assumed to work in tandem with the controls, so there are no additional savings attributed in the analysis.

For the flow meter on the makeup water line, again, this measures assumed work in tandem with the controls and I was not able to find data on uncontrolled water losses, so I did not even attempt to attribute additional savings for this measure.

For the overflow alarm, again, I was unable to find data on uncontrolled water losses and, again, there is no additional savings attributed in the analysis for this measure.

For the drift eliminators, we're proposing to require installation of drift eliminators that achieve drift production of .002 percent of the circulated water volume for counter flow towers and .005 percent for cross flow towers. And this is in line with what's in the ASHRAE 189.1 mean code right now, it's also assumed to be comparable to standard practice in cooling towers, and that the vast majority of cooling towers, both new installations and existing installations, are using drift eliminators already that are achieving about .005 percent. So, we're considering this to be a no incremental cost, no incremental savings measure, but we do want people to use them. And, again, they would not be able to enforce the drift reduction levels, but we would just be enforcing the fact that they are installed. Next slide.

So, for the lifecycle cost analysis, I included not only the water savings, but also the chemical savings associated with the reduced bleed, and I assumed a chemical concentration maintained at 100 parts per million, which is about a gallon of scale inhibitor per 12,000 gallons of bleed water. This does not include any bio side savings, just scale inhibitor savings, and assuming about 10 pounds of chemical per gallon and a cost of about $2.00 per pound of chemical. I also calculated embedded energy savings, but they were not included in the lifecycle cost analysis.

Just a couple notes on the results before I present those. The analysis, I think, is conservative in that the water savings may be understated because, again, we used an industrial water rate, which was lower than the commercial water rate, because we don’t know the distribution of cooling towers being used in commercial settings vs. industrial settings. So, I just assumed a commercial water rate for the entire analysis.

The cooling tower energy use was modeled for an office building, and I assume that cooling towers serving the industrial sector are going to have a different load profile of longer operating hours, and so they would likely experience higher savings than are presented in this analysis. And we did not include any water efficiency incentives or evaporation credits from water utilities, although I know they do exist in some areas of the state.

So, here are the results pulled out of the case report. And the net savings of the lifetime of the measure is $7,540.00; again, this is for a 350 ton cooling tower. And I think that’s about it.

MR. SHIRAKH: So, any questions for Erika in the room? Jon. Jon McHugh has a question for you, Erika.

MS. WALTHER: Okay.

MR. MCHUGH: Hi, Erika. This is Jon McHugh from McHugh Energy.

MS. WALTHER: Hi, Jon.

MR. MCHUGH: Hi. So, it looks like this measure is very cost-effective, and I was wondering why not look at, for instance, a smaller tower tonnage, especially since we’re looking at the chiller systems over 300 tons are required to be water cooled? Is there a particular reason why you’re picking 350?

MS. WALTHER: Oh, well, I used the 350 just as a typical sized tower, just to kind of – because the tower sizes were different for every building climate zone, just to kind of normalize the data, I just picked kind of an average cooling tower size and scaled it. But in terms of the cost-effectiveness, the analysis showed that these measures were cost-effective, actually down to 125 ton cooling tower at the statewide population weighted average water rate of $8.12 per thousand gallons. So, 150 tons or larger is what the Code would apply to and it’s based on the lifecycle cost analysis.

MR. MCHUGH: Oh, I see. Thank you very much, that’s very helpful.

MS. WALTHER: Sure. Okay.

MR. SHIRAKH: Thank you, Jon. Any other questions from the audience? What about anybody online? So, Ron, are there any questions or comments online?

MR. YASNY: Let’s see, Jamy Bacchus is asking – he is with NRDC, “Is embedded energy [inaudible] and does it shift the results –

MR. SHIRAKH: You’re not coming through.

MR. YASNY: Is this better? Okay. “Is embedded energy included and does it shift the results of the cost benefit ratio?

MS. WALTHER: I calculated the embedded energy, it is not included in the cost benefit ratio because it won’t be seen by the consumer. I’m not sure who is paying the cost of the energy to move the water around the state, but it’s not the person who is paying the water bill. So, it wasn’t included in the lifecycle cost analysis.

MR. YASNY: Okay, are there any other questions online? You’re unmuted, anyone can speak up.

MR. DIETRICH: This is Bill Dietrich from

Baltimore Aircoil.

MS. WALTHER: Hi, Bill.

MR. DIETRICH: Hi, Erika. We talked about this a little bit last week and I guess I would just restate my thought that we should eliminate any reference to the performance of the drift eliminators in the Code statement and just, if we want to make a provision that they should have drift eliminators, that’s fine, especially since that’s all we’re expecting anybody to verify when they go out and do an inspection. And as we discussed last week, the performance of the drift eliminators does not impact the water use, so it just doesn’t seem to me that a number should be put in to the standard because it implies that it has some effect on the water consumption.

MR. LINDAHL: Can you hear me? This is Paul Lindahl from SBX.

MS. WALTHER: Okay.

MR. LINDAHL: I would also like to ask that the values that are used, the .002 and the .005 are values that manufacturers claim and that not all manufacturers agree that those are reasonable values for the particular type of product, so putting them in there is probably somewhat risky for CEC because they’re not validated numbers.

MR. YASNY: If you're not talking, please mute your phone.

MR. LINDAHL: Sorry?

MR. YASNY: Everyone that is not talking, could you please mute your phones? Thanks.

MR. LINDAHL: So, I would suggest that Bill's comment is very realistic and it should require that eliminators be there and not attempt to put values on the drift rates, unless you're going to measure and validate them, which I don't think you can do.

MR. SHIRAKH: This is Mazi. This is interesting because I remember I specifically asked the question if you could – these were the numbers that were given to us by the manufacturers last week, and at the time they seemed to be okay with including them, as long as we didn't include any sort of field verification, just basically the manufacturer's specifications. But now I'm hearing that that may not be the case and we shouldn't have any numbers.

MR. DIETRICH: Well, this is Bill Dietrich again. I don't think anybody totally agreed that –

MR. LINDAHL: Yeah, I don't think there should be numbers, I think there's some confusion about our discussions the last time because, if you ask the manufacturers what a reasonable value is for an average for all cost flow and an average for all counter flow, you will get different answers from different manufacturers, and there are manufacturers that make those kinds of products, there are manufacturers that will only make one kind of product, and the answers you get are going to vary by which type of manufacturer you are, in all likelihood. So, I think it's something that is sort of playing into marketing positions and it's not appropriate for this venue.

MR. SHIRAKH: So, should we say something like, have the drift eliminators, but you need to follow manufacturer specifications for them? Would that work?

MR. LINDAHL: Well, again, I think the real point is that the product has the eliminators because there are products that are being sold in the United States that don't have eliminators.

MS. WALTHER: This is Erika at Energy Solutions and I'd like to make a couple points; one is, the reason that the number is .005 percent and .002 percent came from – they are a little more conservative than the .005 for both types of towers that we discussed last week, and I modified those because that's what was in ASHRAE 189.1, but I think people are bringing up an interesting point, that, I mean, we could be perpetuating numbers that they may or may not have value just because they're in 189.1.

We don’t know exactly what value they’re going to add to this Code. And the second thing is, I know we want to keep the drift eliminators in the Code because they offer public health and environmental benefits and, you know, to that extent, I don’t know that the percentage is relevant at the level, or to have it. And so, I’m not sure that we lose anything by dropping the percentage requirement. I defer to CEC, the Energy Commission, on that. But that’s kind of what I’m gathering from the conversation.

MR. HEGG: This is Trevor Hegg from Evapco calling. The comment I would make, and I agree with that. From a water savings point of view, Bill is right, the drift rate doesn’t have to do with water savings because of the – you’re going to blow it down, anyway. But the issue would be, you made the comment, Erika, as far as this probably pertains more to environmental aspects and maybe belongs there, rather than in this section.

MS. WALTHER: Well, I think we decided last week, the Energy Commission was interested in having this and I’m interested in having it in the Code, just because it’s the best practice and it’s not so unusual to have a non-energy Code included as part of a – or non-energy measure included as part of the Code. I think the question is more whether it’s important to have a specific efficiency tied to it.

MR. SHIRAKH: There is a precedence in Title 24 for having measures that are strictly for public health and safety and not necessarily energy since we’re encouraging the use of cooling towers, we have an obligation to ensure that the public is safe because of the use, and that’s why we’re putting this in there. As far as whether we should have a specific number in there or not, if the consensus is that it doesn’t really matter, that the public safety is covered, whether we have those numbers or not, I’m not really – and I’m okay with that. I just want to make sure that people just simply don’t put in drift eliminators that do not function as they’re intended.

MR. DIETRICH: Well, this is Bill Dietrich from BAC and I think the market will drive a lot of that. If a manufacturer isn’t doing the right thing, they’re not going to be able to market their products and, I mean, where do you draw the line? This was supposed to be a water savings standard; do we want to prescribe structural things that pertain to the safety and the operability of equipment? You know, I think we have to be careful about what the intent of the Code is, and especially if we’re going to ask inspectors to try to verify something, we shouldn’t prescribe things that, 1) technically have no impact on the water use if we’re following the other rules in the Code, and then putting a burden on somebody to try to verify a number that they can’t verify.

MR. SHIRAKH: So, from the very beginning, it was never a part of this.

MR. DIETRICH: Well, then it shouldn’t be in the Standard if there is no reason to verify the number. I don’t see any point in putting the number in the Standard.

MR. SHIRAKH: Okay, we can reiterate it as we understand it, you know, we’ll discuss it, and we’ll probably drop the numbers.

MR. DIETRICH: Thank you.

MS. WALTHERS: Thanks for the input.

MR. SHIRAKH: Any other questions on the cooling tower water efficiency measure?

MR. YASNY: Mazi, there are a couple of comments that Jamy Bacchus from NRDC mentioned, they may be a little bit – I’m going to mute everyone.

MR. SHIRAKH: Yeah, somebody is coughing really loud in our eardrums.

MR. YASNY: Okay, Jamy Bacchus mentioned that “TDV is more than the consumer cost, we shouldn’t be restricting ourselves to the simple costs to the user." And he's also saying – put in four initials, "IAPM." And the other comment is, "All three of the following used exact same drift rates that have been proposed: ASHRAE Standard 189.1, 209, Section 6.3.2.3.B, and IAPM Green Building – the Green Plumbing and Mechanical Code Supplement, Section 411.0," and IGCC, PV2 Section 703.7.5." And let's see, last but not least, "Why should California be different from these three model codes? Why is there any objection to placing it in Title 24 Part 11?" Okay, so I'm going to unmute.

MR. DIETRICH: It doesn't pertain to water use.

MR. YASNY: Go ahead.

MR. LINDAHL: Yeah, we just talked through that. I don't think that what's in any of those codes is necessarily relevant. I'm not really sure all the manufacturers have input to what's in all those codes, so I know for a fact that we didn't. So, I think there's more than a little bit of marketing involved in this discussion and it's not appropriate for a CEC Standard.

MS. WALTHER: This is Erika at Energy Solutions. I was going to try to respond to the TDV comment and maybe the Energy Commission has some insight into this, too. I think it's a good point. There aren't TDV values developed for water right now. I think there's still a lot of work being done even to quantify embedded energy accurately and I think there may be an effort also to develop TDV numbers for water, or at least get closer to that point, but right now the best that we have is water rates.

MR. SHIRAKH: The whole idea of TDV, this is a commodity like electricity or natural gas that has different values, different times of the day, there's significant difference between the value of electricity, using a summer afternoon vs. midnight. I'm not sure if that's entirely true for water, maybe it is, but we haven't really developed those methodologies, so you just have to go with what you have. Any other questions or comments related to water efficiency measures from cooling towers?

MR. MORRISON: Yeah, this is Frank Morrison from BAC.

MR. SHIRAKH: Go ahead, please.

MR. MORRISON: Yes. From this morning's session, the air cooled limitation was set at 300 tons. Is there any consideration to move in the requirement for this from 150 up to 300 tons, so they would be in sync?

MS. WALTHER: Well, the Code is based on cost-effectiveness, so that's the basis of my analysis. But I'll leave it maybe to someone else to respond to the broader question of what impact it could have on the market or how it fits in with the other measures that are being proposed.

MR. SHIRAKH: We generally go by cost-effectiveness of this measure and it's been determined that these measures are cost-effective for cooling towers, 150 or even less, so that's the rationale for that number. They are actually very cost-effective.

MR. MORRISON: Well, the concern is, you know, that under 300 tons, you're competing against air cooled, which would not have that added cost, and while at least the jobs that I've seen have some sort of water treatment system with a conductivity probe, it does put a burden in that segment of the market for water cooled vs. air cooled. And, of course, you're still going to have energy savings with the water cooled system even under that tonnage. So, I think the original study shows that, down to – it wasn't 300 tons, but actually lower, was energy savings for the water cooled system, taking into account all the costs of water and everything. This is something you may want to consider, you know, to reconcile the 150 vs. 300 ton limit, bring them into sync.

MR. SHIRAKH: Okay, we'll consider it.

MR. MORRISON: Okay, well, thank you.

MR. SHIRAKH: Any other comments? Okay, we're going to move to the next topic which is Automated Demand Response to Nonresidential HVAC. Dave Watson, are you online? Oh, you're here.

MR. WATSON: I'm in your presence.

MR. SHIRAKH: He's here.

MR. WATSON: Hi. I don't know if you guys have the latest file loaded up or, if not, I have a memory stick right here. Is this a good spot, good microphone to speak from?

MR. YASNY: If you stand here, you can even click your own clicker.

Mr. WATSON: Hello everyone, my name is Dave Watson with the Lawrence Berkeley National Lab, Demand Response Research Center, which is a center that's been funded by PIER for about eight years now. We've been doing Demand Response research primarily on nonresidential commercial and industrial facilities, primarily in California, but as many of you may know, our work is having influence nationally, as well as internationally. I'll just go through these very quickly.

In the olden days when there was too much load on the grid, due to either generation or transmission constraints, people would get on telephones, they'd call up large facility operators and say, “Turn stuff off.” Then, not so very long ago, we started automating things and it made it a lot easier, the dollar signs shown there is along the lines of, when it’s financially mutually beneficial, contracts will be written between ratepayers and utilities that make it mutually beneficial to shed loads during certain times, but I do not want to imply that it’s a requirement to join these programs, or that it is only financially-based.  This measure is essentially proposing that new buildings and major retrofits of nonresidential in California are automated demand response ready, it is not implied that anybody will be forced to do anything that is not mutually beneficial.

This diagram shows a little bit more detail.  And one thing that I do want to point out here is that, I know Mark Hydeman and others have said, “Why not just keep it simple?  Why not just have some relay contacts to trigger this sort of thing?”  That’s actually the simplest model right here, this acronym “CLIR” stands for Client software with Logic with Integrated Relays.  So essentially, a signal comes traditionally over the Internet or Intranet, and when the signal comes in, it closes a dry contact, and that means going into Demand Response Mode, and while many building facility managers do not know how to write Java Code, most of them do know how to interpret a dry contact to perform some shed actions in their sites. Over the years, many manufacturers have started integrating those signals so that it is all software-based.

This is, I guess I'll leave it on this architecture diagram just for a moment more, someone might ask, "Why not just leave things like they are in 2008 where demand response automation signals shall be provided by the utility and just leave it at that?" We're proposing that some standardization is used to ensure that these investments persist over time. So, since there are nationally recognized standards for automated demand response, we suggest that they are used, this will help prevent vendor lock-in, for example. It might be very compelling one year for a utility to allow some aggregator to take care of all their demand response business, but what happens the next year when that aggregator is purchased by another company and wants to change the rules? That investment may no longer be valuable, whereas if an open standard is used, even if the company has changed, the name has changed, the equipment still will be there on-site for many years to come, that's the idea. And it's easy to get bogged down on details and complexities of these systems, but I do want to keep something in perspective, that the value of being able to manage demand, both on a day to day basis from facility operator perspective and from a grid perspective, is very valuable to society, and the PUC and others are still trying to figure out who will gain the benefits of those values. But let's keep in mind that whatever we do with regard to sequences, operations, and what sheds actually occur, and so forth, if we pick a standard that changes in three years, software patches are comparatively low cost, you know, we probably do them to our PCs on a weekly or a monthly basis whenever *Microsoft* and *Adobe* and everyone else sends out patches, so it's not the end of the world if minor modifications in software need to occur, compared to the great value of the end-to-end demand response system that goes all the way from CAISO to the utilities to large loads like commercial buildings into their Energy Management and Control Systems, down to the actual electric loads, themselves. And when you consider all that went into making that happen, including program management costs and incentives, and so forth, we should not get bogged down on software versions. That being said, the open standards have been adopted, a few different flavors, but they are embraced by industry and by lots of major companies. We have over 50 various companies that have embraced nationally recognized standards, and as I mentioned before, the simplest method is having a hardware retrofit solution where you put a box similar to that one I showed right here, this yellow box, that it takes the Internet on one side and has relays on the other side.  One relay means medium-shed, two relays means maximum shed, real simple.  Everything from that – those boxes typically cost around $2,000, but the way – everything from that to no incremental cost if the software is already embedded in a given vendor’s product line, like has been done with many major controls and manufacturers.

So, now I’ll get into the actual Code language. This language has already been – what’s shown here is new for 2013, but it’s already been vetted by the Lighting Demand Response Group, so nothing on this page is new to the stakeholders, I think, unless perhaps there are some HVAC Demand Response folks that were not involved in the lighting portion.  I suppose it could be new to them, but I don’t think anything is too controversial here.  It was wordsmithed to death for over a year or so, you know, we’re comfortable with it.  The main points that I just want to make is that some of the things on the other pages refer back to this, which has already been embedded by the stakeholder community.  This is the first page that shows anything new that I'm proposing.  This is under the Section 101(B), the Definitions section.  And this is a style issue, what is shown here in black, without underline, is the exact verbiage in 2008, Title 24.  So, nothing has changed here.  And this is a style issue, to me it was so important to recognize that the Energy Management and Control System response to Demand Response signals, I suggested putting "See Demand Response Signal" here and "Demand Response Signal" is listed up in Definitions, so that's not too earth shattering, again, it's kind of a style issue. What is more substantial, I think, and this was something that was debated and discussed to great degree, is what do we say when we want to recognize the need for national standards and yet several of the national standards are in motion, they are not static.  The current thinking, and I talked this over with my colleagues at Lawrence Berkeley Lab is – and actually, I think it is the CEC that actually recommended this – is just to reference something that is locked down, and that's the NIST framework and roadmap for Smart Grid interoperability, and their version, that release, is still valid and it's what millions and billions of dollars of Smart Grid efforts are following this roadmap, so it's a pretty substantial document with hundreds of people working on it across the country. So, we're suggesting this kind of being the guiding principle and putting it in Code.

What does that really mean in the real world? Well, here we're talking about air-conditioning today. The first item, again, this is Section 122(h), these words are verbatim from 2008, all the black words are completely verbatim. Item 2 here, I think, is just a typo that, to me, Item 1 and Item 2 are saying the exact same thing, "These Energy Management and Control Systems shall have the capability to turn all the set points up, to reduce the cooling load," but when you do that, you better remember to also turn the heating set points down, so that's all that numbers 1 and 2 are saying, so I just made the language more consistent. Again, that's kind of a style thing without too much substance. The new language here on this section is number 5, which was not stated explicitly in 2008, but I believe that all these things were implied in 2008. Remember, the title in the first line of 122(h) in 2008 says the Energy Management Control System for buildings that have DDC to the zone shall be programmed to allow centralized demand shed for non-critical zones. Well, if it has that capability, then all we're saying here is that it also has the ability to turn that feature off, and that might be obvious, but you don't have to have it operating like that all the time, so we believed that was implied. Being able to adjust all the temperature set points throughout a whole facility with hundreds, or sometimes thousands of zones, is an important thing to do for facility managers, for daily manual demand control. And many people are astonished to find out that that typically is not possible, but we've run the numbers and it would take a full shift of changing set points to adjust all the set points on some of these large facilities, so what I have shown here is 5(ii), just as you can do everything that was described in 2008, you can do that manually from one work station. Or, I suppose you could have a big knob on the wall if you wanted to do it completely manually. And then iii is the automatic demand shed control, and this is saying, "What happens when the system is put in automatic?" There has been a contract signed between the ratepayer and the utility, and it's listening for signals from the utility to shed load. What happens then? This describes it. It says, "Upon receipt of a remote demand response signal," remember that demand response signal has already been defined, "…the space conditioning systems shall conduct this centralized demand shed for non-critical zones during that period." So, again, I don't think anything really new is here, it's just making it more clear, at least from the way that I read it.

Here on the page, this is new text that I think is worth discussing.  This is a new Section 135 that has been discussed kind of – I don’t know the exact title of Section 135, does anyone here from the CEC know the title?

MR. SHIRAKH:  Well, the Section 130 is Mandatory Lighting Requirements and I don’t specifically remember 135, but 131 is for Indoor and one is – Jon?

MR. SAXTON:  Patrick Saxton from the Energy Commission.  I think Section 135 is a tentative new section entitled “Building Power” that would include the text here, as well as some issues about controllable task lighting and controllable receptacles that were discussed in the lighting workshop.

MR. SHIRAKH:  Right, I think Gary Flamm created this section, specifically for this purpose.

MR. WATSON:  And part of the idea here is that, this idea of the system level codes are kind of new for Title 24, I think, and the idea is that we don’t want something like a demand response signal to be scattered all throughout every different section of Title 24 in lighting and HVAC and, who knows, it could be in signage and elevators, who knows how far it could go?  So, we want to hopefully just keep it in one spot and point here.

Here, we're referring back to that NIST framework, but we're getting a little more specific and saying a specific table in that NIST framework actually defines national standards that are relevant to this topic. So, again, it's referring to a document that is cast in stone in January 2010 by NIST, but it's not so specific that it would cause this document to become out of date very soon, so how do we deal with that? Compliance manuals is the answer that I think we –

MR. SHIRAKH: Can I ask you a question on that NIST?

MR. WATSON: National Institute of Standards and Testing.

MR. SHIRAKH: No, my question was, is there like a date or something, or a specific version, like when we refer to ASHRAE, you say 2007, or something? So you may want to –

MR. WATSON: It's right here in Definitions.

MR. SHIRAKH: Okay, so –

MR. WATSON: So, between the letter, you see the second line where the letter is, so in this document, we're saying that the NIST framework and roadmap for Smart Grid interoperability, that's kind of shorthand, even though it's kind of long, for – this is the full title of it. And it gives a specific release number and release date, and to describe how stable it is, it still hasn’t changed, it’s a year and a half later, so –

MR. SHIRAKH: Okay, that works.

MR. WATSON: So, in Internet time, that’s an eternity. So, again, here in Section 135 is where we say, “In that document that we already referred to elsewhere, this is the specific table that you look at to find out which standards are we talking about when we say ‘national standards.’” Okay, it still doesn’t tell you what version is relevant this year, or whatever, so how do we deal with that kind of an issue? And for that, we talk about not including it in the Title 24 Code, you know, the main Code book, but to include more specifics in the Compliance Manuals. And here is where we would get into the nitty gritty of we’re talking OpenADR, which is the open Automated Demand Response data model that was developed by Lawrence Berkeley National Lab, and was adopted by NIST, and now is being used all around the world, as well as Zigbee, which is another standard that is being worked on nationally and is also embraced by NIST.

And that’s really the – so, from a big picture perspective, I’d say, for 2013, we’re just clarifying what was done in 2008, we’re saying, “Yes, Demand

Response signals are good and they should be national standards, and we're using NIST as the pointer to say what does that mean to be a national standard, but with regard to getting into the nitty gritty specifics of which protocol, which version, we're kind of punting on that and putting that in the Title 24 Compliance Manuals. That's what we're suggesting. So, that's the end of my prepared slides, and I have some back-up slides, and I'm available for questions.

MR. SHIRAKH: Mark.

MR. HYDEMAN: Yes, Dave, I apologize for being hard to contact this last week, but have been dealing with some deadlines myself. So, I approve of what you guys did in terms of going to the NIST document, my problem before is you had some language about open protocols, and it wasn't well-defined, but I think that, if you point to a document that says, "You shall do all these things," like the NIST document does, that's perfectly fine with me, so it takes care of that issue that I had. Section 135, interestingly enough, is – and this is a problem for the standards, in general, we're running out of reserve sections; 135 is specifically Mandatory Lighting Measures, it's a subchapter 4. I think it would be better to be under something like subchapter 2, but we have no spare sections under subchapter 2, that’s where all the – it’s kind of a catchall for manufactured construction installation of systems equipment and building components, so I don’t know what we do, but it doesn’t seem – you’re talking about Demand Responses beyond lighting, it shouldn’t be in a lighting only mandatory section, so we either need to change what those subchapters are –

MR. SHIRAKH: Yeah, I talked to Pennington about this –

MR. HYDEMAN: Okay.

MR. SHIRAKH: -- because we’re running out of numbers and numbered areas, prescriptive, and also the subchapters, too. So one option is to actually go to decimals, then we have unlimited numbers.

MR. HYDEMAN: Okay.

MR. SHIRAKH: Section 119.1, that sort of stuff.

MR. HYDEMAN: Okay, well, whatever we end up doing, it sounds like staff is working this right now –

MR. SHIRAKH: What I’m saying is, if you identify a better place that it should go, don’t let the lack of numbers deter you, we’ll find a way of getting it in.

MR. HYDEMAN: Okay, so, Dave, I would recommend, since you’re talking about both HVAC and lighting, rather than being in subchapter 3 which is HVAC, or subchapter 4, which is Lighting, which is where 135 happens to fall, I would put it under Subchapter 2, which is Everything Non-Residential. And that way, it's covered. Or, you have to list it in, you know, the mechanical end lighting and cross reference, you know –

MR. WATSON: What about this, looking at this right now, I recognize that, really, the only difference between here on the Definitions, which also applies to everything, and is where I placed it tentatively is 135, the only difference I see is that it lists Table 4-1 here. What about just put 4-1 at the end of here and just be done with it?

MR. HYDEMAN: A requirement can exist in Definitions. Definitions are there to support requirements, so you need the requirement to live somewhere and I think it should go under Subchapter –

MR. SHIRAKH: Mark is correct, you know, Definitions are not requirements, there's generally just one sentence, plain explanation of what – you can't have a standards requirement any better than definitions.

MR. WATSON: Okay. Well, those are very positive feedbacks and I appreciate all the comments that I've got from all the people that have worked on this over the last year. I know you've been busy, Mark, but we did read your emails very carefully and tried to interpret your intent and I think we've got pretty close and I'm glad to hear it sounds like we're in general agreement, but we're going to find a better home for this, what's described here in 135, it sounds like the suggestion is move it to Section 2.

MR. SHIRAKH: Of Chapter 2 –

MR. HYDEMAN: Yeah, to just say Subchapter 2, section to be determined, we can work it out when we find the –

MR. WATSON: Okay. Any other comments about this topic?

MR. SHIRAKH: Anybody online? Jamy, do you have a comment?

MR. YASNY: I think Jamy's comment was addressed. "I believe we would prefer to see the definition describe the term and the Code body provide the requirements and the associated testing standard."

MR. SHIRAKH: Okay, any other comments online? All right, thank you, Dave. We appreciate it.

MR. WATSON: Okay. Thank you, everyone.

MR. SHIRAKH: So, we're going to move to the next topic which is the Single Zone Fan Controls. Jeff Stein is going to present that one.

MR. STEIN: Okay, this measure was originally called "Single Zone VAV" and it was sort of an update to the single zone VAV section of the Standard, but we've expanded it a little bit to go over a little bit more appropriately “Fan Control and Integrated Economizers.” So, the current Code in Title 24 2008 has a section on single zone VAV which basically says, effective starting next January, DX and chill water units over 10 tons shall be VAV with either a variable frequency drive, or a two speed supply fan, and shall be capable of going down to two-thirds of full speed, low load. ASHRAE 90.1 2010 has a similar requirement for DX units over 10 tons and, then, for chill water units, it was phrased in horsepower of 5 hp, and it was effective basically immediately. So, we sort of started from this point and wanted to see where we could take it from there.

Just a little bit of background. It’s pretty standard practice for single zone units, both chill water and DX, to be constant volume, so we’re expecting that the current requirement that goes into effect soon, and then this proposed requirement, will have a significant change in the market. Another sort of piece of background information is that it’s been our experience, and others have corroborated, that economizers on direct expansion units don’t really – wouldn’t really meet a definition of truly integrated economizers. What often happens is the package units don’t fully capture the economizer savings because the minimum compressor run times cause the economizer dampers to cycle, and this phenomenon sort of gets worse, depending on how many stages of compression and what kind of controls the unit has, but here is some data from a couple of our projects. This data on the left is actually from a unit, a pretty large unit, I think it was a 75 ton unit, that had four stages of compression, but still had issues with –- the pink is the economizer dampers that are cycling, and you can see the black is the supplier temperature -- and so what happens is that the economizer can't quite make set point, the compressor comes on, overshoots, you know, drops the supplier temperature quite a bit, the economizer then cycles off, the compressor then cycles off, the temperature goes up, the economizer opens, and you get a lot of cycling.  We've seen that quite a bit on package units.

So here is the start of our proposed change.  The first thing we wanted to do was put in some definitions for multiple zone and single zone systems.  These actually aren't defined in the standard and there's a couple places where there's sort of implied to be referenced and the term that's often used is "VAV," which was often really intended to mean multiple zone systems, and were largely one and the same in the past, but that's sort of changing now with variable volume single zone systems. So, for clarity, we're defining a multiple zone and a single zone system. And the reason it's important to define them differently is because the requirements will be different for things like fan power on a multiple zone system where you have the extra static pressure of a zone damper, for example, that you wouldn't necessarily – that you wouldn't have in a single zone system. So, we're adding definitions for a multiple and a single zone system. Then, we're cleaning up some of the language right now in fan power where it was broken into constant volume and variable volume systems, and we're sort of clarifying that the first requirement really applies to single zone systems and constant volume systems if you had a multiple zone system, but as long as it was constant volume, it wouldn't necessarily have zone control dampers, and so the lower watts per CFM is appropriate for a single zone system that doesn't have control dampers, even if it has variable speed driver variable volume controls. And then, the variable volume system, which had a higher fan power, we're clarifying that that's really only for multiple zone variable volume system, so a single zone variable volume system would still apply to the lower horsepower. These are actually pretty, you know, don't apply very often because they're only for very very large systems, over 25 horsepower, well, relatively large systems over 25 hp, so most systems wouldn’t fall under this area.

And then we’ve taken out some of the performance language and some of the requirements in variable speed drives because we’ve sort of folded that altogether into the Fan Control section that you’ll see in a second. So, we haven’t actually eliminated any of these requirements, we’ve just put them in a more convenient and appropriate place. So, this is basically the fan power requirements that we’re really not changing, just sort of clarifying, and then the fan control requirements we also didn’t really change, we just relocated them.

Going a little bit out of order here, but going in order of the Standard, the way the Standard is organized, the next change is in the Economizer section, and here is where we’re putting in our Integrated Economizer language, so it actually already requires an Integrated Economizer, you know, by saying “…shall be capable of providing partial cooling, even when additional mechanical cooling is required to meet the remainder of the cooling load.” We’ve taken that a step further and defined what that means a little more clearly, and what it – and we’ve put in an effective phase-in date out a few years because some manufacturers have expressed some concern about having to re-tool or, you know, not all their products being able to comply with this, so it could have an effect on other markets and other products, and so we’ve put in time to give them to prepare for it. Anyway, the definition is that “Mechanical cooling shall be capable of staging or modulating capacity in increments of no more than 20 percent of total cooling capacity; controls shall not false load the mechanical cooling system by limiting or disabling the Economizer or any other means such as hot gas bypass, except at the lowest stage of cooling capacity.” So you have to be able to turn down your compressor, or your cooling capacity down to 20 percent, or lower, without having to cycle the economizer dampers or use false hot gas, or anything else like that. You know, certainly there will be a lot of ways to meet this requirement, one way would be to have multiple compressors or stages of compression with the smallest stage being 20 percent or lower. The direction auto manufacturers are going now is with variable capacity compressors, like the Copeland digital scroll compressor, or with variable speed compressors.

So, coming back now to the fan control, as I said, we started with the current single zone VAV requirement that’s in the Code and basically sort of re-wrote it and made it a little bit more general, and also extended it a little bit, so you know, as I said, it basically applied only to 10-ton units, and actually only talked about single zone systems, sort of implying variable multiple zone systems were already VAV, which is probably generally the case. Anyway, we sort of combined it just to make it clear that this now applies to all multiple zone and single zone systems listed in this table, and I'll show you on the next slide, "…shall be designed to vary the air flow as a function of actual load. Single zone systems shall have controls or devices such as two speed or variable speed controls that will result in fan motor demand of no more than 50 percent of design wattage at 66 percent of design speed." That's sort of in line with what's there now, it says, "Shall go down to two-thirds of the fan speed," didn't define a fan wattage, but we put in a fan wattage that's pretty conservative, you know, you should be able to do much better than that, and you should also pretty much -- all variable volume single zone systems can go below 66 percent fan speed. And then, multiple zone systems shall include controls that limit the fan demand to no more than 30 percent of total design wattage at 50 percent of design air flow when static pressure set point equals one-third of the total design static. This is exactly what was already in there and so we just relocated it so that all of the fan control language is pretty much all in the same place.

So, really, this actually doesn't show you what we've really changed, it's really just sort of reorganizing what was already there. What has changed is in this table here, so, first of all, this paragraph is the same one on the previous page; it just recopied it here. So, what's changed is this table, you know, whereas before we had 10-ton units for chill water and DX, now we're saying that, for DX systems over 10 tons, those are already effectively required to be VAV or two speed. For DX systems between six and 10 tons, effective January 1, 2015, they'll have to be two speed or variable speed. So, that's really the change there on the DX is the six to 10 tons, so we've basically lowered it from 10 down to six.

And then, in chilled water, we've lowered it, instead of in tons, we've gone to horsepower like ASHRAE has, and we've done an analysis that shows down to quarter hp and probably even below, but the requirement was set at quarter hp, you'd have to be VAV, and we didn't feel that there was any need for a delayed phase-in, you know, obviously the standard won't go into effect until after this, but we can put in whatever date we wanted. But the point is, we don't think it needs to be delayed. The technology is easily available now for chill water and evap cooling systems. So, that's the gist of it. This got cut off a little here. The footnote here is just referring to the table in the mandatory efficiency section where they define sort of capacity of units, so there's no confusion about what constitutes a six-ton or a 10-ton unit. And then we've added an exception here for systems that supply 100 percent outdoor and are required to be constant volume in order to maintain minimum ventilation or make up air rates, so obviously there's no point putting in two speed or variable speed controls on a unit that can't really turn down. So, we've added that exception, which is probably pretty obvious, but just to avoid any confusion down the line.

To justify this change – uh huh?

MR. SHIRAKH: So, on that previous table, is the effective date January 1, 2012 – why do we need to put that in?

MR. STEIN: We don't, I mean, we could change it to whatever you want, and the point is we wanted it to be effective as soon as this version of the standard –

MR. SHIRAKH: The effective date of the standard is January 1, 2013.

MR. STEIN: Okay.

MR. SHIRAKH: It seems like the only date you need for that is for direct expansion of less than –

MR. STEIN: What did it say here, just now?

MR. SHIRAKH: Yeah, just it will go into effect with the next standards.

MR. STEIN: Okay.

MR. SHIRAKH: So, January 1, 2014.

MR. STEIN: Okay. Yeah, I mean, we sort of played with the clearest way to try to convey this and maybe this table isn't even the best way to do it. But –

MR. SHIRAKH: I mean, basically you don't want to put a date that's prior to the effective date.

MR. STEIN: Okay. So I'll change that to January – 1/1/14. Okay.

So, to justify this proposal, we went through a series of analyses for both DX and chill water systems, to justify both the fan control, you know, the VAV and two speed portion of the requirement, and also to justify the integrated economizer portion of the requirement, so this is the analysis that was just for the fan control, and just for DX equipment. And we got some cost data from HARI which did a survey of its members earlier this year, and the piece of data that is relevant here is that it was about $500.00 total incremental first cost for a six-ton unit to go from a single speed, single stage compressor, to a two speed, two stage compressor. The only cost that didn't include was maintenance, and so we worked with some local service contractors. We did an analysis that basically modeled a constant speed unit, compared to a unit that cycled between low and high speeds, so a two speed unit, you know, and when the load allowed it to meet the load at low speed, it went to low speed, and when it needed to go to high speed to meet the load, then it did. Needless to say, the majority of the hours are spent at low speed.

And we feel the analysis was conservative for a few reasons, one is we modeled two speed fan with 30 percent power at low speed, but most existing single zone VAV systems have variable speed fans, so they're not just two position, but they're actually modulating and are able to only go to full speed when they really need to, you know, near design conditions, and also the power at low speed is generally below 30 percent, so it's pretty conservative in that regard. The analysis doesn't account for reduction in energy losses associated with compressor cycling, you know, which would go down if you had a two-stage compressor, rather than a single stage compressor, you know, we assumed that the compressor efficiency was going to be the same in both cases, which is pretty conservative, and then we really didn't account for over-sizing, I mean, I think we sized the unit at 110 percent of the peak capacity, which is pretty tightly sized.  You often see equipment that is much more significantly oversized and, in those cases, the savings would be even greater.

Nevertheless, it was highly cost-effective in all the climate zones we looked at, with what corresponds to less than a two-year simple payback, so it was a pretty compelling argument for this lowering the threshold from 10 down to six tons for DX equipment.

So, lowering the chill water threshold from 10 tons down to a quarter hp, we did a couple analyses, one was for a fan coil that wouldn’t necessarily have any outside air, you know, that might serve something like an electrical room, or a computer closet.  The cost data we got from equipment vendors of fan coil units comparing ECM motors vs. standard motors, ECM motors are now a standard option from a number of manufacturers.  Again, it was pretty highly cost-effective, as well, at a quarter hp.  The one thing this analysis didn’t include was any controls to deal with minimum outside air.  We assumed that, if you’re dealing with a quarter hp, you know, tiny little fan coil, that it’s probably not serving something that needed outside air.

So we then did an analysis of a chill water unit that had outside air and then we had to include the incremental cost to add a modulating actuator to the minimum outside air damper. You can no longer have sort of a fixed position minimum outside air damper, on a constant volume unit, you now need to be able to modulate or at least have a two position actuator on that damper, so we included the cost of that from a damper actuator supplier, again, the ECM motors from some local vendors. Again, we felt that analysis was conservative, doesn’t take credit for reduced fan heat cooling energy. All we took credit for was the fan energy savings. It doesn’t take credit for the increased motor efficiency of an ECM motor vs. a standard PSC motor; ECM motors are typically higher efficiency than a standard motor, so it’s a pretty conservative analysis and it showed it was cost-effective for an air handler, you know, serving a zone as small as 500-square-feet, which is probably, you know, the smallest zone you would typically have for an air handling unit that had outside air on it.

So, that’s the analysis for the fan controls. Now, on to the analyses we did for the integrated economizers. So, we did integrated economizer analysis only for DX equipment, for chill water it’s trivial and there’s really no cost associated with an integrated economizer. On DX, we looked at two kinds of systems, multiple zone DX systems and single zone DX systems, so here is the analysis we did for a multiple zone DX system. And we did a simulation of a typical office building. In the base case, we assumed the economizer worked 75 percent of the time, that you had a 75 percent economizer, and that was based on the field research we had done, showing average economizer damper positions were 75 percent or less for units basically where the economizer was cycling, if you integrated the average damper position over time, 75 percent was actually pretty good, some of them were even worse than that.

And then, in our proposed case, we're assuming that you can do full economizing because you would have a variable capacity compressor and we got cost data, again, from HARI. The data that they gave us was a $700 total incremental cost for a six-ton unit to go from two-stage compressor to a variable capacity compressor. And again, we feel the analysis is conservative, it only takes credit for economizer savings, it does not take credit for compressor efficiency savings, such as reduced cycling, more effective use of the heat exchanger, you know, a larger heat exchangers, lower load. And, again, it was highly cost-effective, even at six tons; in multiple zone systems, it's unlikely that you would even have a system that small, most multiple zone systems are

20, 30, 50, 70, you know, 90-ton kind of systems, and the economics are only better, so even down to six tons, it pays for itself. We did get the cost data from HARI for larger systems and, again, the payback was even shorter than for a six-ton unit.

The next one we did, analysis we did, was fan control and integrated economizer for single zone DX, so we basically didn't really have a way to look at just an integrated economizer, so we looked at both the first cost and energy savings of both the fan control and the integrated economizer for a single zone DX system. Our simulation in the base case was a constant speed fan and a partially integrated economizer, and then the proposed case was a variable speed fan with a fully integrated economizer. The partially integrated economizer, you know, in DOE2 doesn't technically have a way to model that, but we came up with what we felt was a pretty reasonable proxy for that, and it's described in more detail in our report. The cost data, again, was from HARI and the incremental cost for a variable capacity compressor and a variable speed fan over a single stage compressor and single speed fan was about $2,000 for a six-ton unit, so a pretty sizable increase in the cost. The incremental maintenance, in talking with some service contractors, you know, we used a number of one-hour year, some folks argue there was actually negative due to reduced wear and tear on compressor dampers due to reduced cycling, so there may actually be a maintenance cost savings. But, to be safe, we used an incremental maintenance amount of an hour a year. The analysis, again, we felt was conservative, it only takes credit for fan and economizer savings, it does not take credit for compressor efficiency savings. And, again, it was highly cost-effective with a simple paybacks corresponding to two years or less in all climate zones that we looked at, so, you know, pretty compelling again.

In terms of measure availability, you know, we spent time talking to different manufacturer about what they had available now and what they could have available down the road, and it’s pretty clear that there’s quite a few manufacturers that have these kind of systems already available off the shelf, and you know, others that could have them in the near term. One of the things you often see, or that at least a few manufacturers now are offering on larger equipment is not fully variable speed or variable capacity compressors, but having maybe two or three fixed capacity compressors with one variable capacity compressor, which allows it to be continuously variable over the entire range, you know, it starts – the variable capacity starts from zero to 20 percent load, then you stage on a fixed capacity compressor, but the variable one makes up the difference from there, so you could have a fully variable capacity on the compression without having all variable capacity compressors. Anyway, it’s readily available technology now. That isn’t to say that this is dominant, as I said, this is the exception now in the market, but something that could easily be brought in.

And, you know, we wanted also to point out that some of the non-energy benefits improved air quality, a truly integrated economizer is going to be able to keep the economizer enabled and provide more fresh air. Systems without truly integrated economizers often end up failing and either fail in the closed position or put in the closed position because they can’t meet the load if they fail – if the facility operator puts it in the closed position, and then you have no outside air, and I’ve seen this myself on a number of projects with older package units that I’ve visited.

Improved comfort, better capacity turndown results in more stable supply temperature and space temperature, you know, if you recall, that plot I showed before, it showed the supplier temperature varying by 20 degrees and that’s pretty common with DX equipment to have highly variable supplier temperatures. Improved acoustics -- reduced fan speed reduces noise and improved capacity turndown reduces the noise of the compressor cycling. Increased equipment life – you know, better control of humidity and electrical system stability, so there's a number of benefits beyond just the energy savings.

So, in addition to the prescriptive requirement, we're proposing some changes to the ACM Manual, the simulation rules for package single zone systems. Under supply fan power, we're defining the fan power ratio at part load, basically using cube law, or the fan affinity laws, to define the power ratio of part load. And then, for system 1 and 2, we say when the base case would be a constant volume and when it would be a variable volume system, and then there's some language in here, "partially integrated dry bulb economizers for systems less than six tons," actually, this is one maybe Jon knows more about than I do, I don't know if Jon is still in the room, but this was one of the comments that we got back on our proposal and added in, that the base case is going to model a partially integrated economizer for less than six tons; over six tons, we're going to model a fully integrated economizer, I think, is the implication. And then, we've put in a sequence for the base case for the supply temperature and supply fan control and the sequence says that the supplier temperature set point shall be reset linearly from minimum at 50 percent cooling load and above, to maximum at zero percent cooling load, and the fan volume shall be linearly reset from 100 percent air flow at 100 percent cooling load to minimum air flow at 50 percent cooling load and below, minimum fan volume set point shall be 50 percent. So basically it’s sort of what we call an airflow first sequence, so as the load goes down from the design load, the first thing you do is reduce the airflow down to 50 percent airflow, and then the second stage would be to increase the supplier temperature from the design supplier temperature, you know, from the minimum up to the maximum, and maximum supplier temperature would basically be room temperature. And so it’s providing no cooling at all at that point. So, that’s how to control the fan speed for a single zone system.

Similar changes to water baseline system, System 5, so if you’re mapped to System 5, again, we’re not changing the design power, but we are saying the fan power ratio shall use the fan affinity laws less than a quarter hp, it’ll be a constant volume system above a quarter hp, it’ll be a variable speed system, and same control sequence, same air flow for a sequence, so first we lower the air flow and then raise the supplier temperature.

Okay, that's all I wanted to talk about. We have proposed some small changes to the acceptance test and those are in our report, you know, nothing significant, basically just sort of keeping with the proposed prescriptive requirement. So, that's it.

MR. SHIRAKH: Okay, thank you. Any questions for Jeff in the room? Jon?

MR. MCHUGH: This is Jon McHugh, McHugh Energy. First, I'd like to start off that, you know, this has a huge impact on the overall air-conditioning consumption in the state. We're now getting down to the smaller air-conditioners, the variable flow requirements into the smaller air-conditioners, which end up being a substantial fraction of the tonnage of the installed capacity, so you know, I hardly endorse this. Related to the economizer language that you had in there, you had a requirement, I think it was to be able to reduce the cooling capacity to 20 percent of its full load capacity under economizing conditions, and so I guess my first comment there is that there's another case study which I believe is looking at requiring economizing down to, I think it's 5,400 Btu per hour, so I think what you want to do is start this off with something for systems larger than 65,000 Btu's per hour, so it's clear – and I'm assuming that’s what you intended here because I think what you’re doing is, when you’re looking at 2, you’re thinking, “Oh, under the old system, we didn’t require economizers until 75,000 Btu’s per hour.” Then, the other thing is that, even for those systems that are over 65,000 Btu’s per hour, I thought you were only requiring two speed fans. So, are those systems going to be able to modulate down to 20 percent? Or how does this coordinate with the other measures?

MR. STEIN: Yeah, well, the fan speed and the cooling capacity are separate issues, right? What this is saying is that, if the cooling load is less than 20 percent, right, the cooling load can be anywhere from zero to 100 percent, right? And so this is saying if the cooling load is less than 20 percent of the design cooling load, you know, you’d have to be able not to false load the compressor. Whether the fan is running at 100 percent speed or 50 percent speed, you know, doesn’t necessarily matter. The issue here is trying to prevent the compressor from overshooting, right, and resulting in either excessively low supplier temperature, which reduces the efficiency and increases latent cooling, or cycling the economizer. So, there’s no conflict that I see between the two.

MR. MCHUGH: But, this is potentially a higher or greater modulation of cooling capacity than your first proposal, which, I mean, you have a two speed fan and you’re slowing it down to 66 – whatever it was – I think 65 percent of full speed?

MR. STEIN: Yeah, to have a two speed fan, you typically have to have at least two stages of capacity control.

MR. MCHUGH: Right.

MR. STEIN: So that’s all that would effectively be required by the two speed fan requirement. This requirement is requiring you to go further.

MR. MCHUGH: Further, okay.

MR. STEIN: With capacity control.

MR. MCHUGH: And is this compatible with the ASHRAE – I mean, my understanding is your variable speed – or your variable air volume single zone system, that’s designed to be compatible with a similar type proposal going through ASHRAE 90.1. Will this requirement be also compatible with that same –

MR. STEIN: Yeah, I mean, we’re a little out of phase. I mean, the ASHRAE 1 isn’t – we haven’t finished doing the analysis yet, but we’re going to be proposing the exact same thing, yes. There will be the same proposals.

MR. MCHUGH: Okay. Anyway, thank you very much.

MR. SHIRAKH: Thank you, Jon. Any questions for Jeff online?

MR. YASNY: Let's see, there was a comment from Jamy Bacchus, NRDC. "As previously noted in other workshops, 'conservative' implies different things. A conservative response to climate change would imply an aggressive action, whereas, if we apply this to savings, and perhaps we're leaving things on the table, why not include savings due to compressor energy savings in the analysis?"

MR. MCHUGH: A couple reasons, one is it's hard to do, DOE2 doesn't have a good tool or technique for doing that, you know, we'd have to sort of come up with a methodology, and the other is, in my mind, being conservative in analysis like this has value because it leaves you less open to criticism. If we could show that it's cost-effective using conservative assumptions, you know, without taking credit for the compressor savings, then it just makes our case stronger. It doesn't mean that we aren't going to achieve the compressor savings, we'll still get them in reality, it just means to work our way through the Code acceptance process, I think it makes our life easier and the Commission's life easier.

MR. YASNY: Okay, and then Mick Schwedler has a comment or a question.

MR. SCHWEDLER: Yeah, this is Mick. A couple things on the question that was just asked. Digital scroll compressors are primarily capacity modulating devices, not necessarily energy conserving devices. So, be careful about saying that it's conservative because there might be more compressor savings.

MR. STEIN: Right. Yeah, Dick had the same comment and then he and I both looked at the performance data for the digital scroll and it basically shows that the efficiency is constant and that the load goes down, which is not going to be the case for fixed capacity compressors. So, you know, you could argue, then, that it is more efficient, but as we said, we didn't take credit, we assumed that the compressor energy was going to be the same whether it had one stage or multiple stages.

MR. SCHWEDLER: And that was just in response to the question. A couple comments, first of all, on the additional costs, it seemed that the hardware was costed out, but not necessarily the additional controls in order to modulate capacity, modulate the additional mechanical equipment in the space. I'm not sure how much that would be, so I'm not expecting you to answer that right now.

MR. STEIN: Okay, I mean, the cost data we did get from HARI, as I understood it, was for a fully functioning system, you know, including all costs to the owner. And some of these were from manufacturers for manufactured products that would include controls.

MR. SCHWEDLER: Some of the wording in the presentation wasn't exactly clear there. And finally, just a comment about the 20 percent total cooling capacity, if we think about a compressor with a variable speed drive, a lot of times the minimum speed of that drive is around 18 parts or so, which is 30 percent. Now, if this compressor is operating when the unit is in the economizer mode, or partial economizer mode, the outdoor air temperature is lower; if it is an air cooled unit, the compression capability of that 18 Hz just got larger because of the reduced condensing temperature, so I guess a question I would have is, could that 20 percent minimum keep people from installing a single variable speed compressor, which could save a lot of energy, but not meet the 20 percent.

MR. STEIN: Yeah, I mean, typically where you're going to see this is on larger units where you're likely to have multiple compressors, even if they have variable speed. I mean, the variable speed compressors I've seen actually don't go above, I think, three or four tons. So, you're probably going to have multiple compressors anyway, in which case one of them could be variable speed and the other could be fixed speed. There is also, you know, the digital scroll has a 10 percent capacity minimum, so you could have a single digital scroll compressor and, you know, easily get below the 20 percent. So, it can be done, but you're right, there's some variable speed compressors; if you had only one of them, then, strictly speaking, you wouldn't comply.

MR. SCHWEDLER: Even though it could be very energy efficient.

MR. STEIN: Sure. I mean, you know, this is a prescriptive requirement, too, so if it's very energy efficient, you could use the performance approach and show that it does better.

MR. SCHWEDLER: On a six-ton system, that might be tough to do and justify, but thanks.

MR. SHIRAKH: Jeff, just one comment on whether we should capture the compressor savings or not. I understand what you're saying; on the other hand, both the Energy Commission and the IOUs, you know, we use the amount of energy savings to justify what we're doing for the standards.

MR. STEIN: Well, we could certainly make an attempt at that.

MR. SHIRAKH: If it is possible to capture it, I think it benefits us and the IOUs. Any other questions, please identify yourself.

MR. YASNY: It looks like there is a comment from Jon Douglas, let me unmute him. Oh, I’m sorry, Jon will go next.

MR. ROSA: Josh Rosa with California Association of Sheet Metal and Air-Conditioning Contractors. Just two questions, the first being with regard to that 20 percent total cooling capacity. Did you consider how that factors with non-attainment zones designated by the EPA?

MR. STEIN: I can’t say that I have. Can you explain what you mean by that?

MR. ROSA: Just those areas that are designated as unhealthy air, I mean, is there any consideration as to – are the same requirements prescribed for economizers in attainment or non-attainment zones?

MR. STEIN: As far as I know, the economizer requirements don’t distinguish between attainment and non-attainment zones, but, again, this isn’t telling you when to do an economizer, this is basically just saying your capacity controls have to be robust enough that you are actually achieving your economizing when we tell you that you have to do an economizer.

MR. ROSA: Okay.

MR. STEIN: I guess I’m not following where we could be tripped up on that one.

MR. ROSA: Okay. And just with cost, it mentions that the data was gathered from a Bay Area contractor, is that just a single one? Or, I mean, is there –

MR. STEIN: Well, there were a bunch of different proposals here. Typically we worked with two or three. I mean, we usually don't rely on data from one contractor.

MR. ROSA: Okay.

MR. STEIN: So two or three.

MR. ROSA: Okay, thanks.

MR. STEIN: Sure.

MR. SHIRAKH: Thank you. There is a question online?

MR. YASNY: Jon Douglas.

MR. DOUGLASS [ph.]: Am I muted?

MR. YASNY: No.

MR. STEIN: We can hear you.

MR. YASNY: We can hear you.

MR. DOUGLASS [ph.]: Oh, you can hear me, okay. About the 20 percent capacity turndown, I had a couple comments. The first one is on the digital scroll, again, I echo Mick's comments that it is a capacity reduction device, it doesn't really impact your efficiency. The other thing is, it is not exactly a quiet solution, so you might not want to talk about it being a noise reduction solution. And then, also, on variable speed compressors similar to what Mick said, is on a single compressor it's difficult to get a turndown much less than one-third, 33 percent, and it just has to do with some issues with low circulation, the air flow and things like that, so I'm concerned that for capacities where you would want to try to get away with one compressor, that that 20 percent turndown is just a little bit too much.

MR. STEIN: Have you seen variable speed compressors on 7.5, 10-ton units? I mean, I've only seen them on smaller units.

MR. DOUGLASS [ph.]: Yeah, they tend to be in the residential range up to five tons, but it's not out of the range of possibility to put them on and the technology is there, so what you're kind of doing is forcing the hand of somebody that is designing the system. If I want to design a 7.5 ton system, to meet the 20 percent, I have to go to a compressor solution, which means you can add cost. If it's something more like 35 percent or something like that, or maybe even 50 percent, then I have a really good shot at doing it with one variable speed compressor. And so I think – and maybe there needs to be a sliding scale that says, you know, when it gets to a 20-ton unit, it's a lot easier to pull off than when you’re doing a 10-ton unit. So I just think the 20 percent – and there’s a lot of other idiosyncrasies when you get to that low a capacity about, you know, again, we’d like to be able to run that system at 20 percent and reduce the fan speed, so forget about the economizers, we’d like to run it to the variable speed system, and you just can’t run 20 percent air flow and have proper duct distribution, so there are other – I know that’s not part of the analysis, but there’s other things that – other issues that kind of make 20 percent tough.

MR. STEIN: I didn’t follow that. What did you mean, “proper duct distribution?”

MR. DOUGLASS [ph.]: If you just forget about an economizer, if you try to design a variable capacity system, and you vary the air flow rate proportion of the compressor capacity, then you actually can run the air flow at 20 percent of normal, you have issues about whether the air is really distributed uniformly throughout the duct system. All the kind of subtle duct design issues that are not a big deal at full capacity become a big problem and you end up with one room that doesn’t get its fair share of the air flow, which normally does when it’s running at full capacity.

MR. STEIN: Okay, thanks.

MR. DOUGLASS (ph.): So, I'd like to see it more like 35 percent.

MR. STEIN: Yeah, I mean, what we're after is, you know, I'd like to see it at zero percent, what we're after is fully integrated economizing and you can go well below 20 percent, as we said, with something like a digital scroll compressor and, frankly, you know, we could try to require that, but right now I'm only aware of one manufacturer of that product, so we had to make it more generic, and so you've got several options here. You know, you can use a digital scroll, which several of the manufacturers are doing, you can use multiple stages of compression, multiple compressors, you can use variable speed compressors if you, you know, as you said, you're going to have to have multiple compressors with one of them variable speed if you're over five tons, but this requirement only applies with over six tons, anyway. So, you know, I think there are plenty of ways to get what we're trying to do here and, you know, we've shown that you can do it, at least cost-effectively, so –

MR. DOUGLASS [ph.]: I just – I feel like – and I don't know how to say this – I feel like this is a pretty big change to – especially in the five to 10-ton systems – a pretty big change to the system to try to make an economizer work a little better. When you're talking about taking a system that is normally basically an on/off one stage, two stage system, and converting it, requiring it to be a variable speed system. And you know, that's a pretty big change in the design of systems. I understand that there are some systems out in the marketplace that do that, but there aren't that many, and there are a lot of issues, there's reliability, I mean, if you look at even the residential marketplace, there are very few variable speed systems out there. And there's a reason why, it's hard, and it's going to – and I'm concerned that making it –

MR. STEIN: I mean, these units cost twice as much as a standard cost in volume unit, so we're saying that we can justify that cost with the amount of energy you're going to save. So, I agree with you, it is going to be a big change, but we feel it's one that's justified and it's worth pursuing.

MR. DOUGLASS [ph.]: Okay.

MR. SHIRAKH: Thank you. Any other questions online?

MR. YASNY: Jany was just going to weigh in on Josh's questions regarding non-attainment. He says, "I believe the gentleman was suggesting that, in areas which are non-attainment of the NAAQS be relieved of providing fresh air which is outside the Clean Air Act required levels." And he also mentions "the Clean Air Act, which was last amended in 1990, requires EPA to set national ambient air quality standards, 40 CFR Part 50, for pollutants such as NOx, Sox, 03, CO, and PM2.5." And that looks like that's it for online.

MR. SHIRAKH: Okay, well, thanks for that clarification. If there are no other questions on this, I suggest we move to the Reduce Reheat. Since we're about 25 minutes behind time, I was going to suggest only presenting, or spending more time on the actual Code language changes and not so much time on the supporting pages.

MR. STEIN: I could do this one a lot faster, Mazi, I promise.

MR. SHIRAKH: Okay. Thanks. Otherwise, we'll be here until 7:00.

MR. STEIN: Okay, look how fast, it's a blank. So, this one, Reduce Reheat, current requirement for non DDC systems, which there are very few of anymore, is that you have to reduce the minimum airflow down to 30 percent. We call that a single maximum control sequence. For DDC systems, you have to reduce down to 20 percent in the deadband, but you're allowed to go up to 50 percent in peak heating, we call that a dual maximum sequence. And in both cases, the minimum can be increased to meet the zone ventilation requirements. So, here's what we're proposing to change. This is the Code language section 144(d) for DDC systems, which is the vast majority, 99 percent of multiple zone systems now. We aren't changing the heating maximum, you know, the 50 percent, we aren't changing the 20 percent minimum, all we're doing is clarifying what we meant here by air flow between dead band and fully heating or full cooling must be modulated. We've changed that to be more prescriptive and the language is the first stage of heating consists of modulating the zone supplier temperature set point up to a maximum set point while the air flow is maintained at the deadband flow rate. The second stage of heating consists of modulating the airflow rate from the deadband flow rate up to the heating maximum flow rate. So you have to stay at your minimum deadband flow rate as much as possible, even in first stage heating, is basically what we're trying to say. And I'll just show you graphically what I'm talking about here.

So, this is the 30 percent maximum which is now only allowed for pneumatic or non-DDC systems, you know, the air flow can go for the maximum to the minimum in cooling, and then stays at the minimum in deadband and heating, and that minimum can be no higher than 30 percent or the ventilation requirement. And when you're heating, you basically open the hot water valve to maintain the thermostat at set point.

So, this is the dual maximum sequence using what we call a temperature first dual maximum, which I would argue is the most efficient, or at least the ones I'm going to show you. And in this sequence, the air flow goes from cooling maximum down to deadband minimum at zero cooling load, stays at the deadband minimum, even in first-stage heating, and in first-stage heating, the supplier temperature set point is reset from some minimum up to some maximum supplier temperature, and then, in second stage heating, the air flow is reset from deadband minimum up to heating maximum, which can be no higher than 50 percent or ventilation minimum. So, this is what we had intended when we put the requirement in in 2008 for dual maximum, this is what we intended to prohibit, and which is prohibited by the current language; in other words, you cannot step up, we said you had to modulate both the air flow and the heating, or modulate the air flow into heating mode. This doesn't modulate, this is using constant volume heat in heating mode. This is a very inefficient sequence because what happens, not only are you going to be reheating more, but you get stuck in heating mode. In other words, as soon as you set into heating mode, you can't get out of heating mode until the cooling load exceeds 50 percent of the design cooling mode. This is what we ended up with in a lot of cases, which was sort of an unintended consequence of the change, which is that a lot of manufacturers used the sequence that, in heating, as soon as the zone went into heating, it reset both the air flow and the reheat valve position, and this is what we're trying to prohibit with the new revised language. And the reason, of course, is that it has more reheat, right? As soon as you go into heating, you're now reheating more air.

And so we did an analysis and basically the cost is for a discharge air temperature sensor. To do this version here, the simultaneous dual max sequence, you don't necessarily need a supplier temperature sensor. It turns out that you actually probably should have one anyway because the sequence has a number of problems ending up with very high supplier temperature, short circuiting, you know, issues like that, poor ventilation effectiveness, but, anyway, to do the sequence we want to do, which is the temperature first, you really do need a supplier temperature sensor, and many contractors actually put these in even when they're not required because they're so valuable for diagnostics and commissioning. Anyway, nevertheless, we talked to some controls contractors and they said, "Well, really, it's no more than about $75.00 per zone for that sensor and for the controls.” The controls are trivial because, once this becomes code, all the manufacturers will offer this as a standard control sequence, and there really is no incremental controls cost, it’s just this cost to the sensor and the labor to install it.

And we did the analysis in all 16 climate zones and showed that it’s cost-effective down to a 1,000-square-foot zone, so, you know, very few zones are likely to be lower than that, you know, the vast majority were likely to be higher than that, so we felt it was reasonable to simply apply it across the board. And again, this analysis accounts only for the boiler and fan energy savings, not for the pump and cooling energy savings, again, just because that made the analysis easier for us to do, unless open to potential criticism. So, that’s about all I had.

Oh, actually, I did want to talk about one other thing. I don’t know if we want to talk about this, but I just happened to notice that the form that’s used right now for the 2008 version is really wrong, there’s a number of mistakes in it, it was actually – someone started with a 2005 form and tried to tweak it to make it the 2008 form. First of all, there were mistakes in 2005, but they’re really bad now. There’s at least five mistakes on this form, you know, pretty serious mistakes. It doesn’t account for demand control ventilation, so even if you had demand control ventilation, it’s going to allow you to put in a higher zone minimum than you should. This is the worst mistake on the form. This used to say 30 percent. The only change they made from the 2005 form is they changed the 30 to 50, so they’re implying that you can have 50 percent minimums now, instead of 20 percent, which is what we really are after with DDC systems, and 30 percent with pneumatic systems. They left the .4 CFM per square foot, which was eliminated in 2008. They misapplied the 300 CFM exception, this exception is for if the design supply flow is less than 300 CFM, then you’re exempt from the whole table, it’s not that your minimum flow can be 300 CFM is your zone flow is 350 CFM. So, they really booted that one. And then, of course, this doesn’t account for the heating maximum, which can’t exceed 50 percent of your design. So, anyway, this form needs a lot of work and I told Mazi we would do it.

MR. SHIRAKH: Fortunately, we can still change the form.

MR. STEIN: Right, and this needs to be changed immediately, by the way, because I happened to see this on a project we were peer reviewing of somebody else’s design, which was using this form, and we said, “Well, they must have made that up themselves,” and then I just went online and downloaded it from the website and that’s what is on the Commission website, so….

MR. SHIRAKH: So, while you’re at it, can you look at our other forms?

MR. STEIN: Question, go ahead.

MR. WATSON: Dave Watson, Lawrence Berkeley Lab. It’s not the reason I’m here today, but just because I have some experience working and commissioning buildings with minimums on the heating side of VAV systems, I know that, in morning warm-up, sometimes the building will fill up with warm air from the top, and as soon as it reaches the sensors, it will stop and the building will still be very uncomfortable in the morning because, below that, the stratified air will still be at everybody’s ankles. And the solution to that was to, in heating mode, to increase the velocity or the volume, to mix up the air.

MR. STEIN: Right, which is one of the reasons –

MR. WATSON: So what is your comment on that? I guess I’m just trying to wave the flag for comfort and energy efficiency, both, so I just wanted to hear your comments on that.

MR. STEIN: Okay, well, that’s one of the reasons why we increased the minimum from 30 percent to 50 percent in heating, because allowing you to supply more air means you could supply it at a lower air temperature, which means you have less stratification. And so, the 30 percent that has been in Code forever and ever, right, was there because that was the same air flow that was used both in deadband and in heating, and you wanted it low in deadband because you don't want to overcool the space and force yourself into heating, but you needed it high enough in heating to get your airflow so you could do heating without trying to heat with 130 degree air that's just going to stratify, so we really solved that one already back in 2008 by putting in the dual maximum sequence. And just to take it a step further, you know, the proposed change here is only going to make it better because what we found, you know, if you did the simultaneous sequence, is you had pretty much the same problems because, you know, it looks nice and linear here and you say, "Oh, well, as I increase the airflow and I open the hot water valve, I'm going to keep a nice reasonable supplier temperature." But hot water valves aren't linear, as soon as you crack it open you get most of the heat out of it. And so, what happens is you end up, like you probably have seen, with 130 degree air and now you've got it at, you know, 20 percent of airflow, so it’s even more likely to stratify. So, this sequence, you know, not only is it less efficient, but it doesn’t really work. Surprisingly though, this is what comes with most canned zone control sequences for most manufacturers, and so we actually – even more surprisingly is that a lot of them still do this, even though this doesn’t even meet code. And maybe even more surprising is that very often you see systems like this, except you see it with a 40 percent or a 50 percent minimum because of the kind of issues you’ve described, and so, you know, we feel like we’ve really solved a lot of issues at once by going with the new proposal.

MR. WATSON: Good, I’m glad you took comfort and the efficiency into account. Thank you very much.

MR. SHIRAKH: What about UFAD? Would that – are we doing anything related to that on the floor air distribution system?

MR. STEIN: Well, you know, a lot of the issues, and I’ve forgotten the gentleman’s name from LBL, but those are largely issues that occur with overhead suppliers, so when you’re doing under floor supply, you don’t necessarily have the stratification issues. But we’ve used this sequence on UFAD systems -- where is my sequence –- this one, you know, and it works well, so I don’t see any conflict or any issue there and, as Mark said, UFAD is kind of on the way out.

MR. SHIRAKH: Why is that? Is that because of changes to building profiles or –

MR. HYDEMAN: Well, this is a little beyond the text of this particular requirement, but Center for Building Environment and Alan Daly from our office worked very hard to diagnose what's happening with UFAD systems and why a lot of them are having problems at perimeter zones, and it turns out, if you take a floor and you put cold air underneath it, and it faces a slab that's really hot because you've got stratification, you have a huge amount of radiant exchange, and so a lot of the Delta T that we were expecting to see in UFAD, you know, the part of the load comes back much warmer, is going away, and so the only way to fix them is to run the under floor system with the same kind of supplier temperatures you would have run an overhead system with, and so now you've paid for a very expensive overhead, under floor system.

MR. SHIRAKH: With no benefits.

MR. HYDEMAN: With no benefits.

MR. STEIN: They didn't work as well as advertised, they cost more, you know, they weren't as efficient as we expected, so –

MR. SHIRAKH: Okay, you learn something new every day. Any other questions from online?

MR. YASNY: Yeah, one second. Jamy Bacchus was asking if we can fix the MECH 3C form prior to 2013.

MR. SHIRAKH: Well, actually we can fix it now.

MR. YASNY: Okay. And then Steve Taylor, I think.

MR. TAYLOR: Yeah, my comment – 90.1 has a limitation on what that maximum set point is that is mentioned here, but not limited. Did you consider -- or can you please consider adding in the maximum set point limitation for overhead systems using the language in 90.1, so that this actually works better?

MR. STEIN: Sure, yeah. I mean, what Steve is talking about is the maximum supplier temperature in 90.1, I think, is 20 degrees over –

MR. TAYLOR: Twenty degrees above base set point, yeah.

MR. STEIN: Base set point. And there’s no cost associated with that, so I don’t think it’s going to require a lot of analysis for us to do, it’s really kind of a sequenced issue.

MR. TAYLOR: It’s probably what you modeled, in fact.

MR. STEIN: What’s that?

MR. TAYLOR: It’s probably what you modeled.

MR. STEIN: Yeah, it is what we modeled.

MR. TAYLOR: So it’s just finding what that maximum set point is in line 3 there, if you just add that requirement to that same sentence – then the maximum shed point shall be no more than 20 degrees for overhead systems,” or whatever 90.1 says.

MR. STEIN: Okay.

MR. SHIRAKH: Any other questions online? It’s interesting, I think in 2008 we actually came up with a compliance credit for UFAD’s. Maybe we should take a look at that. If no more questions, we’re magically back on schedule again. Thank you, Jeff.

MR. STEIN: I told you I could do it.

MR. SHIRAKH: So, Matt Tyler [sic] from PECI is going to present the HVAC controls and economizing. Okay, is he online?

MR. HART: Yeah, I’m online, do you have me on?

MR. SHIRAKH: Yes. You are unmuted now.

MR. HART: I am, okay.

MR. SHIRAKH: We’re going to bring up your presentation in a second.

MR. YASNY: Mazi, let me just take one second to capture this presentation, this past one.

MR. SHIRAKH: Okay.

MR. YASNY: Okay, and which presentation are we looking for now?

MR. HART: I think the title is – it has PECI in the title.

MR. YASNY: Okay.

MR. HART: This is the HVAC Controls and Economizing. And you're going to be changing the slides on this, right?

MR. YASNY: Yep.

MR. SHIRAKH: Okay.

MR. HART: Okay, great. So, we're presenting on some additional economizing measures, I'm going to focus more on smaller rooftop units and we've got several items, basically we're looking at reducing the size threshold in terms of cooling capacity for economizers, getting some more clarification for these smaller units on integration and what that means, also talking about improving the quality of economizers; there have been several studies that show we've got some problems out there, and then we're looking also for multi-purpose rooms and classrooms, and conference rooms where there is some occupancy sensor requirements for lighting, that those also have some temperature adjustment requirements, and heating temperature set points, as well as ventilation related to the occupancy sensor. And then we're going to have Mark Cherniak talking with us about some new fault detection and diagnostic requirements.

Next slide.

So, the basic move here, which is in alignment with 2010 ASHRAE 90.1 is moving the economizer down to 5,400 Btu’s per hour, that’s basically the same as the ASHRAE requirement and I think, yeah, let’s go to the next slide. The analysis was done and basically anything below 50,000 Btu’s makes sense with a pretty healthy cost in the analysis for economizer maintenance, so we could make sure they were being maintained and operating properly over time. So, the limit is about 4.5 tons, 1,800 CFM, and the rest of the requirement there is as it is in the current Code. Next.

And again, the significant comment we received on this was from HARI and they just wanted to see some alignment with ASHRAE and we were able to meet that with this language. Our next proposal relates to clarifying integration, it’s actually always been required, but sometimes Code officials have difficulty seeing for these electromechanically controlled devices that it actually is operating in an integrated fashion, not fully integrated, but with a two-stage thermostat if it’s wired up correctly, you do get what would be called “alternating or partial integration.” And so this allows it to be done any way, but the idea is, when that compressor is not operating, we do get the economizer effect operating between mechanical cooling cycles and that allow us to get a significantly higher benefit out of the economizer than if it just turns off when the economizer is operating, so this is really more of a clarification than a new requirement. Next slide.

And you know, we can see how this also fits into another section of the Code, again, it's basically a clarification that indicates the economizer is providing partial cooling and, between economizer cycles, it's providing as much economization as you can get.

Okay, next Code proposal, this relates to the actual economizer quality and early on in this process, there were some suggestions that all economizers should be factory installed, along with some quality requirements and those were dropped just because there's some shipping issues with that, and flexibility issues, but we did work with industry to arrive at a series of quality components that could be applied to economizers, basically have a series five-year warranty, drive mechanisms that are gear-driven, rather than having linkages, which can get loose or jammed, and actual reliability testing that manufacturers have to take care of in their lab, verify that this economizer is going to open and close 100,000 times, which should give it – I think we estimated about an 18-year life, based on typical cycling. There is a requirement that matches also what we see in ASHRAE 90.1 as far as damper leakage and so that’s what we’re looking for. The next slide has the actual code language – oh, no, we capture a little more – okay, the controls themselves require an adjustable set point or reflectible set point, so that someone can match the newer economizer set point requirements with it. And then it’s also important for these DX systems that the primary control be located after the cooling coil, otherwise we end up with a comfort issue with the economizers and they get disconnected in the field. There are some requirements around sensor accuracy, we have seen some issues in lab testing of sensors on the lowest economizers and these bring requirements up to a reasonable accuracy that will provide better control. And the sensor calibration are plotted on a sensor performance curve so that information is available from the manufacturer so that someone can actually look at, say, the amp draw or voltage of the sensor and actually verify its calibration in the field, and we also need to have the outside air sensor located to prevent false readings. In other words, either be in the hood, or shielded from direct sunlight, and that there be some sort of relief air system built in to provide relief air so that we can actually get the outside air into the building. And that is Code proposal 3. Code proposal 4 gets into some ventilation issues. One thing that is currently unclear in the current ventilation code is that, when the building is scheduled, but not occupied, when a space is vacant, it's a little bit unclear about whether you can reduce ventilation when it's unoccupied, so we've added some language here. We're using an occupancy sensor in the space and there's nobody in the room, we can actually turn off the fan or shut off the ventilation. These requirements are – this would allow it to be done for a package unit, and we'll get into the other requirements later, but we're requiring it for VAV reheat systems, but for the package unit it would be allowed by this language, or at least it would be clarified that it was allowed.

We do have some provisions in here that the occupancy sensors meet some requirements that already exist in the lighting section, we didn't reproduce all those details here, so that we have a decent quality of occupant sensor, and we also made it clear that, if you had a manual on-type, some of the lighting occupancy sensors keep the lights off unless someone manually turns it on, and that wouldn't work for ventilation control, which may mean that the easiest thing to do where ventilation control is required, occupancy sensor control you would actually put a sensor in, and there are 24-volt occupancy sensors available that are more compatible with package unit or DBC control systems, and it may well be as cost-effective to put that in, rather than try and re-use information from the lighting sensor, just because you have different trades, and we costed – when we analyzed this, we costed it up as if we had a new 24-volt occupancy sensor going in where these requirements were applied.  And then – go ahead.

MR. SHIRAKH:  I wonder if this language has been cleared by the Cal OSHA folks, they may have some issues with this.

MR. HART:  Yeah, at this point, it's been out in the stakeholder groups, we haven't heard specifically from them at this point.

MR. SHIRAKH:  Yeah, I don't think they're aware of it.

MR. HART:  Yeah, and we should probably loop them in.

MR. SHIRAKH:  They've engaged us on other parts of the Code, the garages, labs, and other areas, and they're very vocal and it appears to me that they would be interested in this, as well, they just don't know that we're doing this.

MR. HART:  Right.  And, again, reading the current language, it looks like the current language could be interpreted to allow this, so we're positing it as a clarification. In addition, this is compatible with the National Consensus Standard, ASHRAE 52.1, that specifically has examples in Users Manual –

MR. SHIRAKH: Usually, they don't buy those arguments, so….

MR. HART: Yeah, well, okay. And we also require that there is a purge cycle that does occur daily as required in the ventilation code that the occupancy sensor does not lock that out. And so, anyway, these are just basically allowing circumstances.

The next part of the Code proposal and the next slide, we're looking at multi-purpose rooms that are smaller than 1,000-square-feet, so you can actually get an occupancy sensor of that room, classrooms and conference rooms of any size, when we've got a variable air volume system, we'll have an occupant sensor and there are two things that is going to create, one is actually broadening out the deadband a little bit, this just allows that room to float a little bit, it's still actually within the comfort range and shouldn't take very long to recover from this slight adjustment in temperature, as well as close its own damper so that, when that space is unoccupied, we don't continue to provide ventilation air, which also needs to be reheated into that space, typically when it’s empty. And I think that’s the conclusion of that measure.

I don’t know if we want to answer questions on those first four proposals before FED, or answer all the questions.

MR. SHIRAKH: No, we can take questions on the first four. Jon McHugh has a question.

MR. MCHUGH: Jon McHugh, McHugh Energy. Reid, you know, this proposal has a huge impact on the state. Just to start with, approximately what fraction of the energy consumption by air-conditioning is saved by this particular measure?

MR. HART: You know, we analyzed it on a per case basis. Matt, are you online? Did you work up any statewide numbers?

MR. TYLER: Yeah, this is Matt. Jon, we don’t have that just yet for our statewide, but that’s certainly something that we could provide.

MR. MCHUGH: I guess the next question is, you saw the presentation earlier by Taylor Engineering about what they’re proposing for integrating economizing with the loading of the air-conditioner compressor, and so I just make the comment that I think you two need to organize your language there about, you know, cycling –

the cycling type control of the compressor relative to the control that Taylor Systems has pointed out.

MR. HART: Yeah, I think some overview of the language is probably important. Their variable capacity provisions don't come into effect until 2015, and these other provisions would come in as soon as the Code is adopted, and also, theirs, I believe, stop at a certain size, around six tons, and this would capture actually where about the majority of units are in that four and five-ton range, requiring to have variable speed. But it probably makes sense to look at how all this language fits together so it is clear in the Code.

MR. MCHUGH: And then, finally, you have a series of requirements to actually make sure that the economizer works in terms of its physical capabilities. Is there any reason to place a lower limit on that? I mean, if someone makes a really small economizer, wouldn't you want to make sure that the damper is able to cycle and all the various things that you have to make sure that the economizer works?

MR. HART: Yeah, I think that's a straightaway economizer requirement, so I think it applies to any economizer, not –

MR. MCHUGH: I thought the language you showed here on the screen had a minimum size that it applied to.

MR. HART: Oh, I see, okay.
MR. TYLER: That’s just how the cost-effectiveness works out and that we needed to establish a lower limit for cost-effectiveness.
MR. MCHUGH: Yeah, I would just ask that maybe you revisit that question around the issue of public safety in terms of, you know, that the economizer actually works correctly and maybe some of this is actually justified in terms of if someone – so you have requirements that require an economizer, you know, then you need to show cost-effectiveness, but when someone chooses to install an economizer, should there be some minimum requirements for that economizer for public safety issues that I think actually trumps the cost-effectiveness argument? Thanks. I guess the final thing is, that would certainly make those requirements a lot easier to enforce, that those requirements apply across the board. Thank you.
MR. HART: Yeah, hopefully we’ll get some spillover even if there is a minimum size, so that once manufacturers are providing this testing, you know, pretty much it’ll be done for their economizers across the board, a lot of these requirements are [inaudible].
MR. OATMAN: Hello?
MR. SHIRAKH: Go ahead.

MR. OATMAN: Yes, Ron, were you calling on me? This is Kirk Oatman.

MR. YASNY: Yeah, go ahead.

MR. OATMAN: Yeah, so the chat message that I sent was I had an exchange with Gary Flamm yesterday. It looks like he is going to consider adjusting the definitions of things like occupant sensor and some things like that, to specifically, well, to define them more as the functionality rather than as a device, and that comes from the fact that IMN [ph.] Control offers an EMS that can provide all those capabilities, not just as a direct connection from a sensor to turning something on and off, but taking many more things into effect. So, we would like to see language which kind of throughout, where there is a consideration of the fact that there is an EMS operating at a higher level, rather than a simple device like, you know, the one slide says “two-stage or electronic thermostat,” you know, another says “setback thermostat” where we can provide all the setback capabilities as long as the thermostat is a communicating thermostat. Does that make sense to you?

MR. HART: Yeah, I think we tried to revise the language, I’m back on slide 7, so it says “shall have control systems,” and then it says, “…including two stage or electronic thermostats.” And we did get this feedback earlier and, you know, if you have suggested better wording on that, but the idea of control systems, and then it goes to what the functionality is, but it’s just trying to indicate, well, two stage or electronic thermostats. We’ll provide that functionality because that is typically on the lower end units of what goes in and trying to make it easier for the Code Official to understand for a lot of these smaller package units.

MR. OATMAN: Yeah, and that certainly is the challenge in all of this, to define it so that it can apply both to low cost installations and to an installation which has an installed EMS. Now, just as a reminder, a system like ours can be very inexpensive, so you know, we don’t want to get stuck in thinking that EMS’s are only for very large buildings, they can end up applying to some pretty small installations. So, I mean, for instance, on slide 7 here, you know, it does say “shall be equipped with a setback thermostat,” it doesn’t say, “or system,” and I think there are references that I saw other places in the chat I mentioned, Section 144, 149, and 121, 122, that have references to devices instead of functionalities, so again, Gary Flamm, I think, if – you mentioned you might be referring to some of his definitions, at least for occupancy sensor; maybe that would be the way to approach it for this, to have a definition which prescribes the functionality for the thermostat kind of devices or functions, and then refer just to those definitions.

MR. HART: Yeah, you know, this is existing language and we try to be consistent and I think we can certainly look through that as we're polishing up the language and make it more general.

MR. OATMAN: Yeah, I don't mean to make a huge amount of work and change, but I think it's really important that this revision really specifically acknowledges and enables EMS's to frankly provide even greater savings than simplistic individual devices can, so we kind of want to encourage that, rather than make it difficult or, unfortunately, sometimes people will just interpret language that is not explicit, their own way, and perhaps not approve an EMS installation in which the intention at this moment, as we're all talking, might have been to have approved.

MR. SHIRAKH: So, Reid, do you know Gary Flamm? He is on our staff here.

MR. REID: Yeah, we'll get in touch with Gary and see what efforts are going on and make sure we coordinate.

MR. OATMAN: Thanks, gentlemen.

MR. SHIRAKH: Jon.

MR. MCHUGH: Jon McHugh, McHugh Energy. I think the language that's currently there is pretty important and I think the main part of this section, so it doesn't describe the control system, but it is indicating two stage or electronic thermostats, the primary issue here is that, for these small economizers, a very common failure mode is that these systems have been installed with a single stage thermostat, so that's the primary purpose of the electronic or two stage thermostat that is in here, so it's actually a pretty important part of the language. Thanks.

MR. HART: Yeah and I think we want to leave that in, I think the issue as I understood it was more actually with the existing language under number 1 for setback thermostat and, you know, I'd have to sit down and look at Section 112(c) and see how that describes things, and make sure we do allow a more, you know, the control functionality, not just a setback thermostat, so we'll take a look at that.

MR. SHIRAKH: Okay, any other questions on the first four proposals online?

MR. YASNY: Yeah, let's see, John Douglass (ph.) wants to make a comment. John?

MR. SHIRAKH: John, if you're muted, you need to unmute yourself. We can't hear you, if you're still interested in making a comment, why don't you send us a chat message?

MR. YASNY: Okay.

MR. SHIRAKH: Any other comments on the first four proposals? And if we get a comment, we can always come back to it, so I'm going to move forward to proposal 5, Fault detection. Go ahead, Reid.

MR. CHERNIAK: Mazi, this is Mark. I presume you can hear me?

MR. SHIRAKH: Yes.

MR. CHERNIAK: Just a note, by the way, John Douglass said he's trying to talk to everyone. It did come through on the chat, so I didn't know if you wanted to try him again.

MR. SHIRAKH: If he's still online, he can –

MR. YASNY: Yeah, he's unmuted, so I don't know what to do.

MR. SHIRAKH: We haven't muted anyone, everyone is open now.

MR. CHERNIAK: Okay, good. Thanks. So, this is Mark Cherniak, New Buildings Institute. I'm representing the PIER part of the team, the case team and PIER team put together this Code change proposal for RTUSDD. I was part of the 2008 Title 24 initiative that got FTD for rooftop units, and terminal air handling units, and as a compliance option, but since that time, we've been looking at moving ahead with a proposal for a prescriptive requirement, again, not that this would not be a mandatory requirement, but prescriptive, but these may tend and do tend to set the basis for performance standards. Next slide.

And here is the proposed language, and we are, by the way, since we have also been talking to folks, we're also putting our eggs in the 54,000, the 4.5-ton basket, in terms of FTD requirements, along with, again the economizer requirement. Next.

And these sensors, we would like to be permanently installed in the unit, you can see all of those right there. And next.

System requirements that we would like to see, first, that the unit controller can initiate, in other words, an operator technician can initiate the operating sequences to make sure things are actually operating as required, or as designed, that we get the information from the unit off the roof either into the building in some manner, or to a more remote location, however that might happen. We're certainly not prescribing the method that could take, but anyway, to get the information off the roof to an owner or a service contractor, a building manager.

A performance indicator that allows simply understanding the efficiency of what’s going on in the unit, and having the system itself certified by CEC and verified to be installed correctly. The next stage, of course, in addressing these particular issues have to do with the development of methods of test, as well as alarm or fault detection thresholds. Southern California Edison has completed a Statement of Work and will begin work very shortly on developing methods of tests, as well as test thresholds.

We’ve asked U.S. DOE if they would like to collaborate with us on the development of the test methods and thresholds due to their rooftop units, high performance specification, it came out a month or a month and a half ago. We have selected six faults at this point, we originally started with a list of 13, and we’ve paired them down largely in response to the reality right now that the OEMs have all of the major manufacturers offering rooftop units of a certain class, certainly, have one or more of these faults already embedded in their controls and controllers, and we thought this would be a good place for us to start in terms of moving ahead, or potentially an additional number of faults or alarms to be embedded at a future date. But we know the market is moving along. Next slide, please.

There are third-party providers, as well, of some of these fault detections, so, again, currently available products certainly by the OEMs that list on the tops the bullets, and at the bottom we’ve got three additional parties, at least, at this point in time, who can put these capabilities to use currently, certainly in retrofit equipment -- for retrofit, as well as for new equipment.  Next.

And there are pieces of this FTD in development, the smart monitoring and diagnostic systems from PNNL, it’s been under development and continues in development, NILM, Non-Intrusive Load Monitoring, an idea that came up at MIT, one that Virtjoule and Company is pursuing at this point in time, so we’re very confident in terms of the availability of product by the time the 2013 implementation date comes around.  As I said, we will have methods of tests and thresholds clarifications, or, sorry, I should say “thresholds metrics” available later this year.  So, any questions?

MR. SHIRAKH:  Jon?  You’re the only audience here.  Okay, no questions from the audience here. Anybody online?  Okay, well, I think we’re all good, then.  So, thank you for that presentation and we’re going to move through our last item on the agenda, Air Compressors and Russ Torres from Energy Solutions is going to present that. Russ, are you online? Can you try Russ again?

MR. YASNY: Russ?

MR. TORRES: Hi, can everyone hear me?

MR. YASNY: Yes.

MR. SHIRAKH: Yes, we can.

MR. TORRES: Oh, great.

MR. SHIRAKH: So we're going to load your presentation, just give us a moment.

MR. TORRES: That's fine.

MR. YASNY: Actually, this looks like Version 2, I think there is a newer version, isn't there?

MR. TORRES: There is a newer version, it should be the PDF, version 3.

MR. YASNY: Oh, that's right, we're going to use PDF, right.

MR. TORRES: Sorry about that. We've been having some issues with our Powerpoint.

MR. YASNY: Okay.

MR. SHIRAKH: Do you have a Powerpoint?

MR. YASNY: No.

MR. SHIRAKH: All right.

MR. TORRES: There should be a way to actually put this in a presenter mode. Maybe in tools.

MR. YASNY: Guide me.

MR. TORRES: Actually, we have different versions, don't we? It's okay, we can work off this, this is fine.

MR. SHIRAKH: I couldn't hear what he was saying.

MR. TORRES: We can work off of this, this is fine.

MR. YASNY: I can make it a little larger if that would be helpful. Would you like me to make it larger?

MR. TORRES: No, this is fine, it will be easier to switch pages if you don't make it larger.

MR. YASNY: You're up.

MR. TORRES: Great. Thank you. So, I'm Russell Torres and I am one of the case leads for the Compressed Air Systems Measure. I'm working with Ransom Byers, also from Energy Solutions. And so the scope of our proposal is looking at mandatory requirements for compressed air systems that are at least 25 hp and above. Again, this is helpful for new construction and major renovations.

So, before I go through the presentation, I just want to give you a brief overview of what we'll be looking at. So, first, we're looking at the proposed Code change language and I'll give you a few minutes to look over this. I won't take any questions now, I'll first go over kind of how we got to this proposed Code change language by going through some of our motivations, the methodologies as far as energy savings and incremental cost goes, and then our lifecycle cost analysis. And then we'll turn back to this proposed Code change language and kind of go through some of the issues that we've been resolving and then have a discussion about it. So, again, I'll be giving you a couple of minutes to look at the Code change language. [Pause]

MR. SHIRAKH: Is everyone okay with advancing?

MR. TORRES: So, our motivations behind looking at compressed air is that compressed air is actually new territory for Title 24 and basically Title 24 has been looking at a lot of the low hanging fruit as far as like lighting and HVAC is concerned, but process loads were something that were kind of new. The only process loads that are currently regulated by Title 24 are refrigerator warehouses and, for compressed air, it comprises about 16 percent of industrial motor systems energy and, within that, I think it's about 10 percent of the total industrial energy use. This is about 90,000 gigawatt hours of annual consumption. Now, with cost-effective measures, with less than a three-year payback, savings can be at least 70 percent or more. This is around the 15,000 gigawatt hours annually. Next slide, please.

I mentioned that there are cost-effective measures with less than a three-year payback. Well, those are a lot of measures and this is a lot of term measures that anyone can look at for any compressed air system, and Ransom and I spent some time kind of going through all of these energy efficiency measures, speaking with stakeholders, doing some research, and kind of based off this process and a couple other factors, we nailed it down to two -- next slide, please – the first being requirements for a full range efficient trim compressor, as a designated trim compressor, on all compressed air systems. So, a trim compressor looks at the term load and kind of takes care of that term load above what the base load is. The term load is pretty variable and, so, the compressor for the most part is at part load. Now, [inaudible] machines don’t work very well with part-load, but there are certain machines that do work well with part load, like a variable speed drive compressor. We originally had a proposal to require a VSD compressor, however, we decided to move towards specifications for compressors like a VSD. Basically, we wanted a trim compressor that could achieve the same thing as a VSD, but not limit you to a certain technology. Second, we’re looking at requirements for smart system controls on multi-compressor systems, the minimum requirements being that it’s able to make a decision based on what the current demand is, as measured by a sensor, and then we had looked at a requirement of having it be able to work with various manufacturers and compressor types. But we'll get to that portion when we move forward to the Code change proposal. Next slide, please.

So, as far as energy savings goes for smart controls, we first had to set up our baseline and we came up with four different baselines. This was based off of data from a Utility Voluntary Program; we wanted to get a good sense of kind of what the market was and based off of that, and some stakeholder feedback, we came up with four different baselines that were of different sizes. With these baselines, there were also two different load profiles, one load profile was matched fairly well to what the compressor makeup was, and the second load profile had a slight change from what that original load profile was. This was to model the changes in demand and load as throughout the life of a compressed air system because change does happen. For each of the baselines, the load profiles are pretty much the same, they're just normalized to what the compressed air system capacity is, and then we also included auto shutdown timers. We felt that this was a conservative move on our part because some systems don't always come with auto shutdown timer. We then took these baselines, ran them by our stakeholders to kind of get some feedback, made some changes, then went forward in modeling.  We then modeled each of these baselines in Airmaster with the help of a certified Airmaster instructor, and then applied these smart controls manually because it’s not actually something you can do specifically in Airmaster.  We then compared the energy use to determine the annual savings. Next slide, please.

As far as energy savings for VSDs, we looked at specifically just the trim compressor and our baseline trim compressor was a load/unload lubricant injected, rotary screw compressor with two gallons of CFM storage. This was based on feedback from our stakeholders and it looked more like one gallon of CFM per storage was more typical, but the two gallons of CFM storage were more typical if an audit had been done, so we decided to go, again, with the more conservative route.

For the modeling plan, we modeled both VSDs and the baseline with two gallons of CFM of storage and we modeled these baselines with Excel formulas.  These formulas were actually based off of information from Airmaster and the reason why we decided to use Excel was so we could do a more broad analysis, or a parametric analysis to look at many different compressors with different load profiles.  We then compared the energy use per hour for each trim compressor and then had our energy savings. Next slide, please.

The incremental costs for smart controls includes a control unit, which makes those decisions, the interface with each of the compressors, any sensors that are necessary, and then also the labor. Now, labor is usually not included for incremental costs, but we thought that there is actually a lot more man hours required to set up a smart control properly. So we decided to include these costs. These costs are based on estimates from three different manufacturers, for each of the model baseline systems; basically, we showed them our baselines and asked each of these manufacturers to give us an estimate. For VSDs, these costs are also based on values from manufacturers and were also shown to various stakeholders for feedback. These costs also include a trendline, again, because we're performing a parametric analysis, this trendline is actually quite necessary. Next slide, please, thank you.

So, in this graph, we looked at the estimated costs to add smart controls for each of the baseline systems. Now, for baseline 1, 2, and 3, each of these systems are a two-compressor system, just of various sizes, and for smart controls what is really driving the cost is the number of components. So, for baseline 1, 2 and 3, because each of them is a two-compressor system, the incremental cost is the same. For our last baseline, this is the three-compressor system, and the cost increases, as you can see. Next slide, please.

For the incremental cost for the VSD compressors, we compared them to cotton [ph.] speed rotary screw compressors, and we had information for both discounted prices and what the original prices were, and since the original prices were more expensive as far as incremental cost goes, we decided to move forward with those incremental costs and then base our trendline off of it. Next slide, please.

For smart controls, we took our incremental costs and our energy cost savings and then compared them, and looked at what the LCC savings were – sorry, the Life Cycle Cost savings were. And because the lifecycle cost savings were all positive, it shows that for each baseline, smart controls are cost-effective. Sorry, can you go back to the last slide?

One thing we also noticed is that – we looked at baselines of different sizes because we believed that the larger the size of the system, the more savings we would get, but it turns out that that's actually not the case. What we ended up finding was that, if the expresser makeup was matched very well to the low profile, and then the savings weren't as high, so we really didn't depend on size, but more on kind of the demand profile and what the compressor makeup is, however, for the most part, a compressor system isn't matched very well to the demand profile. In new construction, the demand profile isn't really known until the system is actually run and, even then, the demand profile changes, as I mentioned before. Next slide, please.

For variable speed drives, we looked at three different profiles operating in various ranges, and for each of those profiles and for a variety of trim compressor sizes, for each of them it shows in this graph that this measure is cost-effective. And, again, we were looking at kind of mostly new construction, but we are also looking at renovations and retrofits. For smart controls, we're looking at having the mandatory four renovation retrofits if the combined horsepower of both compressors is increased. And our reasoning for this is that we believe the smart controls, there isn't any more additional cost to doing it for a retrofit, rather than the new construction, whereas there would be a fee if you have to add in a new compressor, especially a large one, the incremental cost might be a bit more. And so, for renovations and retrofits, we're looking at just smart control being mandatory. Next slide, please.

This is the proposed Code change language that we looked at in the beginning of the presentation. I just want to call out a few things before we get into questions for the trim compressor requirement, so Part B. Again, we were originally looking at requiring a VSD and we decided to go more towards looking at part load performance instead. And we'll go into kind of the numbers. Right now, it says maintaining 22 kilowatts or less of input power per 100 acfm of output. Those numbers are actually still things we need to tweak and, again, that will be talked about in the next slide. And then we also wanted – sorry, can you go back – thank you. We also wanted to make sure that these trim compressors were sized appropriately. Some of the feedback that we got from stakeholders is that we thought a size requirement on the trim compressor, it might be able to provide a loophole for people to perhaps put in a VSD that was really small, like, say, a five horsepower one, and we want to avoid that loophole, so our size requirements can be based off of what we consider is the useful trim load and we're still trying to define this, but basically we're looking at kind of what the largest step size is within a system to avoid any control gaps. For the smart controls requirement, Part A, again, we had originally had the requirement that the control system would be compatible with compressors of different manufacturers and different types, but after speaking with a few stakeholders, we realized that this requirement isn’t actually that important and so we took it out and focused on a control system that can choose the most energy efficient combination of compressors based off of the current air demand. So, given all that, does anybody have any questions?

MR. SHIRAKH: Any questions from the audience? Anybody online? My only comment is – this is Mazi – that some of this language has to be changed to what we consider proper Code language; for instance, we don’t need to have references to building permits and things like that, it’s already assumed. So we’ll clean that up later.

MR. TORRES: Okay.

MR. SHIRAKH: So there are no more comments.

MR. TORRES: So I have, actually, two more slides to go over the issues that I called out. So, for part load performance metrics, we’re looking – we’re wondering if there is an industry standard metric for evaluating trim compressor part load performance. The CAGI datasheets that we’ve been seeing show what the performance is at a certain rate of pressure and so now what we’re looking at is for some continuous range of, say, X percent of the compressor’s total range that the compressor can deliver air at a certain pressure, using less than an efficiency performance value, P KW/100 CFM, at all points within the range. So, in our proposal, X is actually 70 percent of the range, and P is 22 KW/CFM. Some of the feedback that we received is that efficiency metric P might change, depending on what the operating pressure is, and so what is still on our plate has to do with plenty of different variable speed drive compressors at different operating pressures, and looking at that KW/CFM metric. I guess what we're kind of hoping to get from stakeholders is if there is perhaps other sources of part load performance data for VSDs, maybe there is a study done that we just haven't found, or, if going through CAGI datasheets is kind of our best bet for this.

MR. SHIRAKH: There are no comments.

MR. TORRES: So I guess those are the issues that we're looking at for the trim compressor. Next slide, please.

So this has specifically to do with smart controls, but also for trim compressors. So, with smart compressors, there is a set-up process that is required and I guess what we're hoping for is, is there some standard output for all smart control systems based off of what the set-up process is. One of the tests that was suggested was that we run the compressors through – I guess through its full range, going from zero all the way up to its full load, and then back down, and seeing the smart control react to each of those. But if anyone online has any ideas about if there is a standard output to setting up smart control, our case team is curious.

MR. SHIRAKH: Does anyone online have any suggestions? It's all quiet.

MR. TORRES: I believe this is actually my last slide, so if anyone does have any questions or comments that are online, and maybe they're just not talking now, please feel free to give me or Ransom a call, or send us an email; we would definitely appreciate your feedback.

MR. SHIRAKH: Okay, any questions related to the air compressors?

MR. YASNY: It looks like we do have a comment from Eric Bessey.

MR. SHIRAKH: Okay, we're getting a lot of background noise.

MR. BESSEY: Hello?

MR. TORRES: Hi, Eric.

MR. BESSEY: Oh, you can hear me, okay.

MR. TORRES: It's a little rough, actually. Are you driving or –

MR. BESSEY: No, I'm just at my office here. That's interesting, I wonder if my computer microphone might be better. Maybe I can flip open the lid here and go that route. Let's see here –

MR. YASNY: I'm going to mute everybody but Eric.

MR. BESSEY: Hi. Hello, can you hear me?

MR. SHIRAKH: Yes, we can.

MR. BESSEY: The trim compressor, maintaining an efficiency of 22, I don't recall how that 22 was achieved, where that came from, but that's not my point; I wondered if the wording should be an average of P KW/100 CFM instead of having a direct number because the load/unload compressor is going to be loaded sometimes and unloaded sometimes to achieve an average of that value. So, I don't know what your thoughts are about that. That was the comment I had about trim compressors. And then, about smart controllers, there really isn't – you know, the smart controller isn't trying to achieve efficiency, it's just barking out orders, it's loading and unloading compressors, turning them on and off, and as long as the compressor does that, then it is doing its job. It's not like you're tuning the controller to achieve better efficiency, you're just telling it to do things and it's doing discrete things. So, it's not the controller that's efficient, if you know what I mean, it's all about how you program it, and then it just does what it's told to do. So…. Anyway, that's all I had to say. Hello?

MR. SHIRAKH: Yes, we can hear you.

MR. BESSEY: Okay, that’s it.

MR. SHIRAKH: Okay, any other comments or response? Ron?

MR. YASNY: Russ?

MR. TORRES: Yeah?

MR. YASNY: Okay, you’re back on.

MR. SHIRAKH: I don’t know if you have a response to that, Russ?

MR. TORRES: Regarding smart controls? I mean, I understand your point, Eric, and I guess I’m curious if there’s – again, if there is a standard output process for kind of testing smart controls, I imagine that it’s probably different for every manufacturer, kind of based off how they program the smart controller, but, I mean, if you had a sense of maybe if there was a standard test that people do, if you have any thoughts on that, Eric?

MR. BESSEY: Well, it’s – what’s the specs for the smart control, I mean, that’s the first thing – what is a smart control? What does a smart controller do? What’s the spec? And then, can a manufacturer meet that spec? That’s what it is, so you just have to tell them, I mean, you program the thing to do what it’s supposed to do, and then you can verify that it does that, you know, through some commissioning, "Oh, yes, it does indeed keep Compressor A as the trim machine at all times." Certainly, you would not want a smart controller to baseload a VSD, that would be counterproductive, so maybe you specific that the VSD must remain trim at all times. Well, you program it to do it, and then you look at it, you know, it's like lights, you know, motion detectors on lights in rooms, you don't really – you're not programming it to be efficient, you're just programming it to turn the light on and off when it's supposed to, so you verify that through observation more than anything – is the smart controller turning compressors on and off? It doesn't matter what the efficiency of the compressors is, it's just is the smart controller doing what it's supposed to do. So, that's it.

MR. TORRES: Okay, I mean, it does look like we may have to kind of push forward on an acceptance test.

MR. BESSEY: Yeah, there probably needs to be some sort of spec, you know? It can be fairly loose so that people – so that all players can play, but certainly some verbiage of not having the 300 hp be the trim machine while the 150 is the base, when it could easily be the other way around, you know, so something like that.

MR. TORRES: Yeah. Ransom, did you have any comments, actually? Ransom?

MR. BYERS: Hello, this is Ransom. Can you hear me?

MR. SHIRAKH: Yes, we can.

MR. BYERS: Okay, great, thanks. I see what you're saying, Eric, and I guess our very very loose wording approach to that was the simple sentence that it has to be able to look at the current load and pick the most efficient set of compressors, and so in an ideal world it seems like that captures essentially what the control should be doing in all situations. Are you suggesting that we should get more specific, like either one direction we have a longer set of cases, or rules that are a little bit more specific in that laying out the things it should be doing to achieve that goal? Or, even more specific, talking about particular types of compressors and how they should be controlled?

MR. BESSEY: Well, maybe for starters. It's kind of a, you know, the fairly open spec you have now, that the controller should maintain the mix of most efficient compressors, well, that might be good during part of the time, but it may be there's another part of the time when that simply doesn't work for a certain load based on, you know, it could be maintenance factors, there could be demand rates of change factors where maybe the most efficient mix of compressors are being met at this particular time, but arguably is the best way to do it based on some other factors? So, there's room for, you know, if you leave it like that, there's room for interpretation and judgment calls, then maybe that works, you know? You're blatantly not making the most efficient compressors, you're baseloading your VS compressor, that's certainly one thing, but there might be another time when, "Okay, I do see why you're running your compressors in this way." Mathematically it's not the most efficient, but it certainly fits the bill best, given this particular situation. So, if you leave it open for judgment like that, then maybe that's okay, you know?

MR. TORRES: Okay, thank you. We'll definitely look at that again and see if maybe expanding that would be helpful. And we'll follow-up with you directly with whatever we think of that seems to make sense.

MR. BESSEY: Sure, yeah. Maybe just a couple of, you know, footnotes on it, you know? An example: "Do not baseload your VS compressor and part load with the constant speed load and hood [ph.]." Maybe just a couple of examples like that to lead people the right way.

MR. TORRES: Right, exactly. So maybe not necessarily covering every single scenario that might be there, but the most common big ones, you know, “This is the sort of thing that it should be doing.”

MR. BESSEY: Yeah, the big follies, there are a couple of big follies that controllers can do and you cover those big ones and maybe that’s okay for now.

MR. SHIRAKH: So generally examples don’t go into the Code language, they go into the Compliance Manuals?

MR. BESSEY: Uh huh.

MR. SHIRAKH: We can probably handle it that way.

MR. TORRES: Okay, that sounds like a good way to do it.

MR. BESSEY: I didn’t hear that, please repeat.

MR. TORRES: So, separately, there’s the Compliance Manuals and then there’s the Code language, itself, and he was saying that the examples and things like that would tend to go in the Compliance Manuals rather than the more synced direct Code language, itself.

MR. BESSEY: Oh, yeah, sure, okay, yeah, that makes sense.

MR. SHIRAKH: Mike.

MR. MCGARAGHAN: Mike McGaraghan, Energy Solutions. Thanks, Eric, for your input here, and I think we can take this offline, I just want to circle back to the main point here, is really a compliance issue, is just how do we ask a Building Inspector to go into a building and confirm relatively quickly that these controls are doing what they're supposed to be doing, and if that requires an on-site test, some sort of acceptance test to do that, we could try to set that up. But if it's something that installing contractors, if there's some series of tests that people do anyways when they're installing the controls, that demonstrate some of these examples that you're talking about, then maybe we don't need to do any additional acceptance testing, we just need some documentation of the fact that these controls have already been set up to do XYZ, and that documentation could be shared with the Building Inspector. And so that's what I think the case team is getting at, you know, how do we summarize that process into some sort of form or to streamline compliance. That's all, thanks.

MR. SHIRAKH: Thank you. Any other questions or comments from the audience in the room? I don't see any hands. What about anyone online? So, I take it that was not a comment. Any other questions or comments about anything related to anything discussed today or to the Standards?

So with that, I'm going to close the workshop for today and we'll reconvene again on May 5th, which is about a week from now and that would be our last Nonresidential workshop before we move to the Res. Thank you so much.

(Adjourned at 4:18 p.m.)

**REPORTER'S CERTIFICATE**

I do hereby certify that the testimony in the foregoing hearing was taken at the time and place therein stated; that the testimony of said witnesses were reported by me, a certified electronic court reporter and a disinterested person, and was under my supervision thereafter transcribed into typewriting.

And I further certify that I am not of counsel or attorney for either or any of the parties to said hearing nor in any way interested in the outcome of the cause named in said caption.

IN WITNESS WHEREOF,

I have hereunto set my hand this 6th day of June, 2011.

Kent Odell

Kent Odell
CER**00548