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NUCLEAR REGULATORY COMMISSION

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ADVISORY COMMITTEE ON REACTOR SAFEGUARDS

(ACRS)

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SUBCOMMITTEE ON THERMAL HYDRAULIC PHENOMENA

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TRACE APPLICABILITY TO ESBWR LOCA

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FRIDAY

FEBRUARY 27, 2009

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The Subcommittee met at the Nuclear Regulatory Commission, Two White Flint North, Room T2B1, 11545 Rockville Pike, at 8:30 a.m., Sanjoy Banerjee, Chairman, presiding.

SUBCOMMITTEE MEMBERS PRESENT:

SANJOY BANERJEE, Chairman

SAID ABDEL-KHALIK, Member

MICHAEL CORRADINI, Member (via telephone)

ALSO PRESENT:

DAVID BESSETTE, Designated Federal Official

GRAHAM WALLIS, Consultant

RALPH LANDRY, NRO/DSRA

JOHN MAHAFFY, RES/DSA

JOE KELLY, RES/DSA

NATHANAEL HUDSON, RES/DSA/CDB

RON HARRINGTON, RES/DSA/RSAB

ANDREW IRELAND, RES/DSA/CDB

MATTHEW PANICKER, NRR/DSS/SNPB

KATHY GIBSON, RES

JOSEPH BOROWSKY, RES/DSA/RSAB

STEVE BAJOREK, RES/DSA

DON FLETCHER, Information Systems Labs

JENNIFER UHLE, RES

A-G-E-N-D-A

P-R-O-C-E-E-D-I-N-G-S

8:29 a.m.

CHAIRMAN BANERJEE: The meeting will now come to order. This is a meeting of the Advisory Committee on Reactor Safeguards Subcommittee on Thermal Hydraulic Phenomena. I'm Sanjoy Banerjee, chairman of the subcommittee. Subcommittee members in attendance are Said Abdel-Khalik, John Stetkar may join us later, Mike Corradini will be on the phone. I'd also like to welcome ACRS consultant and former ACRS chairman Graham Wallis. David Bessette is the designated federal official for this meeting.

The purpose of today's meeting is to consider the applicability of the TRACE thermal hydraulic system for the loss-of-coolant accident analysis for ESBWRs. The subcommittee will gather information, analyze the relevant issues and facts, and formulate those positions and actions as appropriate for deliberation by the full committee in September. The rules for participation in today's meeting have been announced as part of the notice of this meeting previously published in the Federal Register. Portions of today's meeting will be closed for the discussion of proprietary information. We have received no written comments or requests for time to make oral statements from members of the public regarding today's meeting. A transcript of the meeting is being kept and will be made available as stated in the Federal Register notice. We request that participants in this meeting use one of the available microphones when addressing the subcommittee. The speaker should first identify themselves and speak with sufficient clarity and volume so that they can be readily heard. With that, we can start the meeting and I think the first thing on the agenda is an introduction from Chris Hoxie who is there, and we'll take it from there.

MR. HOXIE: Okay, good morning. My name is Chris Hoxie. I will give you a brief introduction. We are here today to discuss the adequacy of the TRACE computer code for performing confirmatory analyses of design basis loss-of-coolant accidents and the Economical Simplified Boiling Water Reactor, or the ESBWR. As you can see from the agenda, this will take the better part of the day. While TRACE is also used for transient calculations and MELCOR was used for long-term containment pressure calculations, these topics are not covered today. Today we're going to focus on ESBWR LOCA calculations.

In our work we used a phenomena identification ranking cable approach to focus our reviews and evaluations on phenomena, processes and components and systems important to the prediction of ESBWR behavior during LOCAs. The TRACE code governing equations, numerics and closure relations were reviewed for applicability to ESBWR LOCAs. Particular attention was paid, of course, to the new ESBWR design features such as the gravity-driven cooling system, a passive containment cooling system and a chimney region. In terms of development, a measure area of TRACE model development in support of the ESBWR analyses concerned the film condensation model, and Joe Kelly will talk about that shortly. Evaluations were made of integrated code performance for predicting the behavior observed and experiments pertinent to the ESBWR LOCAs. The experiments include tests representing important basic physical processes, separate effects tests simulating performance of unique ESBWR components, and integral effects test configured to represent the overall behavior of the ESBWR reactor system. Assessment of the code against experimental data represents a major portion of the code adequacy demonstration.

A quick review of the agenda and the presentation shows that we'll spend a fair amount of time discussing these topics. To accommodate the ACRS subcommittee's request, we did segregate the presentations so that all of the public stuff is at the beginning and then we go into closed session.

CHAIRMAN BANERJEE: With regard to the agenda items, which ones are going to be in closed session? You have the agenda in front of you, right?

MR. HOXIE: Basically open is up until I believe - let me.

CHAIRMAN BANERJEE: So you see, closed session you have to go in from 8:45. No sorry, the overview - oh, okay.

MR. HOXIE: This is not the -

CHAIRMAN BANERJEE: It doesn't show the closed.

MR. HOXIE: We were planning on following this.

CHAIRMAN BANERJEE: Well, I don't mind following whatever agenda, I mean that would be fine too. Okay, so you want to reorient the agenda to follow the thing that you've got there which is different from what we have in front of us.

MR. HOXIE: The problem is going in and

out of closed session. My understanding was that -

CHAIRMAN BANERJEE: Okay, so what we want is to put Joe Kelly at some point, and that's the only one which is open?

MR. HOXIE: Yes. The very first one.

CHAIRMAN BANERJEE: Which is what, the TRACE Film Condensation Development?

MR. HOXIE: Right.

CHAIRMAN BANERJEE: So, on the agenda that we have in front of us that's shown at 11 o'clock.

MR. HOXIE: That's correct.

CHAIRMAN BANERJEE: So what we could do is we could close the session after this and then open the session at 11:00 through lunch, and then close it after. Is that what you would like? The only open - oh, sorry.

MR. HOXIE: There really were only two. There's Joe Staudenmeier with the separate effects, and there's Joe with the film condensation. Those were the two open pieces. Everything else is closed.

CHAIRMAN BANERJEE: Okay, so we'll open the session after the break. Okay, we can do that. All right, carry on Chris. We'll just reorient the agenda.

MR. HOXIE: All right. Let's see here. I

think then we're pretty much - basically I wanted to mention we view this as an informational briefing for you. We know that there are future meetings scheduled for ESBWR and we view this meeting here as supportive of -

CHAIRMAN BANERJEE: Good morning, Mike.

MEMBER CORRADINI: Good morning.

CHAIRMAN BANERJEE: Okay, the meeting's started. Sorry. Just Mike, Chris Hoxie is on and besides the first introduction section now he's simply saying something about how the staff view the meeting. Are you on, Mike?

MEMBER CORRADINI: Yes sir, I'm on.

CHAIRMAN BANERJEE: Okay Chris, go ahead.

MR. HOXIE: Our hope is that this will be supportive and informative, and that future meetings I believe that are planned for this year, that they can focus on the ESBWR design certification as opposed to having TRACE be the center of attention. So with that I think -

CHAIRMAN BANERJEE: Well, let me ask you before, since Mike is now on, that there are two ways to view this issue. One is that we could put in some conclusions from - which the subcommittee and the full committee then agree to - to an ESBWR report at some point which I have to ask Mike about. The alternative is to aggregate all the passive plants, write something based on, say, TRACE applicability to the various passive cooled plants in one document. So there have been two sort of schools of thought as to how the ACRS as a full committee should respond. Let me ask Mike first to give an opinion on this. Mike?

MEMBER CORRADINI: Well, two things. I apologize for being late. Beautiful weather in the Midwest.

CHAIRMAN BANERJEE: I'm glad I'm missing it.

MEMBER CORRADINI: I was going to say that I think since this is coming to the Thermal Hydraulic subcommittee the focus - I mean, now this is my opinion and I'll let you guys on the subcommittee and full committee decide further, but I think the focus ought to be on TRACE's applicability to passive plants, and any particular technical challenges that raises. And then as those things are identified, try and understand how the staff is going to address those. There were some - there's been a number of things that the subcommittee on ESBWR is interested in where we've asked the applicant to go back and do analysis and we've yet to see it, but those things -

Said is there - those things involve noncondensable gas trapping and then potential reduction in ECCS for - on the GDCS and the PCCS because of that, and we wanted to see calculations. Similarly, I would expect that if the staff is using TRACE to audit the applicant's calculations to make sure that they have comfort that things are working properly and they'd be using TRACE, we want to understand the applicability of TRACE to those same situations. So I think I would be looking for, from my perspective, general technical issues that are unique to passive plants that staff is going to have to look at relative to TRACE applicability and use.

I mean, the one that - I'll give one that I wanted to ask about eventually, and I will apologize since I have to go somewhere in two hours anyway, is the - is this question was raised in the peer review about the momentum equation and corrections that have to be made to make it more consistent. And particularly when I have low pressure driving heads, that could be an issue. So that's an example of something that started off in the peer review as generic, but yet has particular application when I have a passive plant with low pressure heads. Is that enough for now?

CHAIRMAN BANERJEE: Yes, I think that's fine Mike. What I understood from you, and maybe you can write me a brief paragraph or a note at some point.

MEMBER CORRADINI: Yes, I will.

CHAIRMAN BANERJEE: Okay, and just summarizing your views that would be fine.

MEMBER CORRADINI: Yes, that's fine. Sorry.

CHAIRMAN BANERJEE: Thanks.

MEMBER CORRADINI: And I'll listen primarily unless - I just ask everybody if you could for at least the next couple of hours speak up a tad so I can hear, and then I'll try to be quiet unless there's something that you guys aren't going to pick up. But with you folks in the room, I'm guessing you're going to ask similar questions.

CHAIRMAN BANERJEE: All right. Okay. So Chris, are you done now?

MR. HOXIE: Yes.

CHAIRMAN BANERJEE: So, in the revised agenda the next speaker will be - we are still in open session and we will be in open session till the break. The next two speakers will be Joe Kelly on the film condensation model development and followed by Joe

Staudenmeier, who will speak on the separate effects assessment. After that we go into closed session. Why are we into closed session for such a large amount of time? Is it because the data are proprietary data?

MR. HOXIE: It's proprietary.

DR. WALLIS: Do we have a handout from Joe Kelly?

CHAIRMAN BANERJEE: Yes.

DR. WALLIS: We do?

CHAIRMAN BANERJEE: Where is the handout from Joe? I think you can go ahead, Joe.

MR. KELLY: Okay. I'll be speaking to the development that we did for the TRACE condensation model, the applicability to the ESBWR. And this is actually a model that was - the need for which was identified early in the program and we spent a concentrated development effort on it. The work you're going to see was pretty much done five to six years ago, and committee members that have been for awhile have seen elements of this presentation several times already. But I'll see if I can get it better this time.

I'm going to start out with an introduction, very, very brief background, modeling approach, and overview of the model accuracy. The model description, this fits into a two-fluid framework, so there are actually changes to constitutive models in five different areas: wall friction, interfacial shear, wall-to-liquid heat transfer, and then the liquid-to-interface heat transfer, and then finally the noncondensable gas effect. There wasn't enough time in this presentation to go through all five areas, so I flagged the two that I'm going to attack, and that's the -

DR. WALLIS: What's missing in there is buoyancy.

MR. KELLY: Well, if we're talking about a falling film then it's the gravitational -

DR. WALLIS: No, I'm interested in ceilings and water surfaces. One of the problems that TRACE had was modeling the surface of the pool, the condensation on the pool, which is governed very much by buoyancy effects. And ceilings are the same, and I didn't see them in your scope.

MR. KELLY: No, and you won't. I shortened the title of this presentation by one very critical word.

DR. WALLIS: So maybe you have some work to do in those areas, Joe?

MR. KELLY: Well, we've looked at doing,

you know, the turbulent wave jet modeling in the pool and so on, and didn't go to the history effect because we were initially going to use TRACE for the whole containment and so I started doing some work on that and kind of developed a first cut at a model to put into the suppression pool. Then we were going to use the contained code for the containment, so all the TRACE work on that stopped. Then there were problems with the TRACE contained coupling and we went back to using TRACE for everything, but by then it was too late to put that model in.

DR. WALLIS: Well, you're telling me history, but it seems to me that there is work to do on the area where you don't match the data which is predicting surface condensation on the pool.

MR. KELLY: And this is just my recollection because this has been years and I haven't been involved with this work for a couple of years now, to handle that they did some sensitivity studies to see how important the effect was and decided that for, you know, and I don't remember what the data was, but -

DR. WALLIS: Yes, but that still isn't an excuse for why it doesn't model PUMA. So let's not get into that till we talk about the data.

MEMBER CORRADINI: Can I ask a follow-up question just so I'm clear? So is the intent of TRACE to actually do containment condensation phenomena such as the one Graham is suggesting, or is that just an extra benefit?

MR. KELLY: For this particular case where the containment is so intimately coupled to the reactor system, the final position was to use TRACE for the entire containment.

MEMBER CORRADINI: Okay, thank you.

MR. KELLY: And I really should have said this presentation is about film condensation because certainly what Professor Wallis was talking about are important things, but they're different than what I'm going to talk about.

CHAIRMAN BANERJEE: So your interest is primarily here to model the PCCS to ICS and these types of -

MR. KELLY: That was the initial thrust was, you know, we know that the ECCS is a new component, it's an important component, let's make sure TRACE has a model for it. ICS is also there and then later when the scope of the TRACE model was extended to include the containment volumes, then we needed a model for the condensation on the containment walls with the effect of noncondensable on those as well.

CHAIRMAN BANERJEE: So you can handle, if you like, the vertical walls, film condensation on those?

MR. KELLY: Yes, and I'll show some comparisons here.

CHAIRMAN BANERJEE: Whereas what Graham was talking about, on the horizontal surfaces -

MR. KELLY: Yes, evaporation from, you know, a stratified layer on top of the suppression pool, that kind of thing, that's different.

CHAIRMAN BANERJEE: What about the ceilings on things where pre-convection effects and so on could be important?

MR. KELLY: I haven't looked at that in detail so I can't say.

DR. WALLIS: The ceiling is - they'll talk about the ESBWR or not in this stage. Maybe not.

CHAIRMAN BANERJEE: Well, I think we'd better hold it to the closed session. As long as we understand exactly, and I think you brought it up there. PCCS to ICS and the wall - vertical wall condensation.

MR. KELLY: All right. So this is what I

was tasked to do was first to come up with a model in TRACE, or just show that the existing TRACE model was adequate for the passive containment cooling system. What you have here is obviously condensation and the presence of noncondensable gases, but it's co-current downflow and the films tend to be laminar and the Reynolds number of the gas mixture is fairly modest so the gas velocities are low, so interfacial shear does not play much of a role. It's pretty much a falling laminar film with noncondensable gas condensation. If we start condensing noncondensable gases we've got a problem. But at the same time we wanted to be able to model the Isolation Condenser System. That tends to be more pure steam because the primary system is still closed up, at least when that tends to be important. And what you have here is a highly sheared turbulent film, more of the classical kind of in-tube condensation.

DR. WALLIS: Now, the argument that's used later in the report is it doesn't matter because the PCCS is self-regulating anyway.

MR. KELLY: I think that's probably true. Quite often there are system effects that one thing balances the other.

DR. WALLIS: Or if it goes too far one way it corrects itself.

MR. KELLY: Right.

CHAIRMAN BANERJEE: How - I guess this was my question and Graham has asked it already, is how sensitive are the overall results to getting this right?

MR. KELLY: I don't know. I haven't been involved with this for awhile and I didn't do the ESBWR calculations. There are people here in the room who can answer that question better than I can. And I think when they start doing the plant calculations and show those to you they'll be able to answer that.

CHAIRMAN BANERJEE: The sensitivity to these condensation -

MR. KELLY: Right.

CHAIRMAN BANERJEE: Okay.

DR. WALLIS: Have you been out of the picture for awhile?

MR. KELLY: I've been working on gas reactors for more than a year now and my last year really with the TRACE development team was dedicated to documentation.

DR. WALLIS: Nobody comes back to you and says Joe, why did you do this? They just accept whatever you did?

MR. KELLY: No, they do, and that's kind of why I'm here today. Later on, pretty late in the process it was decided to extend the scope of TRACE to the wall condensation. And so you'll see how I modified the model to account for that as well. The first thing you do is, you know, does what we have existing in TRACE work well enough that we just go with that? So I took a look at what was in the model and kind of went well, I don't think so, but just because you'll see like the effect of noncondensable gases was an empirical correlation and it was for condensation on a turbulent liquid jet in a cross-air stream which you don't expect to work in PCCS.

DR. WALLIS: So the question here is that TRACE has to be rebuilt when you get to ESBWR. Is it going to have to be redone when you get to some other design?

MEMBER CORRADINI: Graham, can you say that again? I didn't understand what you just said.

DR. WALLIS: I was just saying, if you found that the old model didn't work for ESBWR and you fix it.

MEMBER CORRADINI: Okay.

DR. WALLIS: I'm just wondering if it has to be fixed again when we get another design. Because these phenomena are so design-specific.

MR. KELLY: If we talk about tube condensation, I'll say no. If we talk about something completely different, you know, say it's an innovative design for the flow of an accumulator, for example. Well, if that's something that's never been studied before, yes. But I didn't just go on reviewing the model. I went ahead and did some calculations with TRACE against data to see if it did work, and the performance was pretty poor. And so that led to a model development effort. And so again, I've already said this a couple of times, in-tube condensation, the presence of noncondensable gases for the PCCS, but I also make that same model applicable to the isolation condenser, later on modify it for wall condensation for large containment models.

TRACE is a two-fluid code, as you know. Most literature correlations are really built for, I don't want to say homogenous things, but you know, where you assume things are at saturation temperature. So a lot the literature models don't go into a two-fluid code very easily. You have to shoehorn them in, and play some tricks, and so on. So what I wanted to do since I had the chance to possibly develop a new model was try to make it consistent with the two-fluid framework. And another part of that is when you're solving the two-fluid equations, you're already providing a lot of the information that some of the empirical models really would need. Now what I'm talking about here is by solving the mass and energy equations you get the axial distribution of the condensive flow rate. You don't have to calculation it like you do in the Nusselt model. By solving the momentum equation you now get to liquid film thickness, assuming you do that right, okay? If you get to liquid film thickness, the Nusselt formula is just the liquid connectivity divided by the film thickness.

CHAIRMAN BANERJEE: But that's only for laminar.

MR. KELLY: That's correct.

CHAIRMAN BANERJEE: Assuming the film is laminar.

MR. KELLY: That's correct. But I'm replacing that final formula with K over delta because I'm taking advantage of what - everything the code is going to go through to calculate the delta anyway.

DR. WALLIS: It's not just gravity that's pulling the film. The second equation follows from the first by doing momentum balance, but it's only gravity that's working.

MR. KELLY: Right. Yes, there's no interfacial drag in the Nusselt, it's only gravity and wall drag. Whereas if you're using a film thickness that comes out of solving the two-fluid conservation equations, if you do it right you at least can get the effect of interfacial drag on the liquid film thickness.

CHAIRMAN BANERJEE: Well, there is a problem with the Nusselt formula because as soon as you get significant waves you get stirring. So the delta there is an effective delta and not a -

MR. KELLY: That's exactly right. You know, it kind of all comes out in the wash because you change the Nusselt formula part, but.

CHAIRMAN BANERJEE: All right. Anyway, I think we get the picture here.

MR. KELLY: Right. Well, so I want to give you an overview of the model accuracy. Now, this is the set of test data that I used for the condensation database, and the calculations I'm going to show you in just a second are more spreadsheet-based. What I did was for each data point I took the local conditions from the experimental data and stuck them in the new model and I'm going to compare the

TRACE model to several empirical correlations.

CHAIRMAN BANERJEE: So, this - the ABDHBI stuff in the report, is that the MIT stuff?

MR. KELLY: Yes. I will refer to it as MIT-Dehbi and I'll show some - a plot of that later. I can describe. I don't have drawings, but I can tell you what it looked like later. This was done for N2 because this is what I started with, and the Dehbi is more, you know, a vessel test. I wanted to do pure steam condensation first. Well, for the ESBWR conditions there was a whole series of tests at Berkeley, and the last one which was the one that got the best data because they learned progressively how to do the tests better was by a student named Kuhn, and he did both pure steam condensation tests as well as steam air and steam helium. And you'll notice all three of those show up. So for pure steam, that gives me the conditions that you're more likely to see in something like the PCCS. The NASA Goodykoontz test, those are, you know, highly sheared turbulent-type films into small diameter.

DR. WALLIS: That's very old.

CHAIRMAN BANERJEE: Even the UCB-Kuhn data, falling liquid - if you're basing that Reynolds number on the film thickness, falling liquid films become turbulent actually at close to 400 to 800. So the 2000 data is well turbulent.

MR. KELLY: Yes. Well, as in - I brought along a bunch of extra slides so I can show you how the wall liquid correlation goes, and it does - first, it goes above Nusselt through the effect of the holes on the film, and then as the Reynolds number increases it gets higher, then it gets fair into a correlation for a turbulent film. So that's there, and that's why we can do these tests, okay? Then for air steam, again the UCB-Kuhn, but they had some companion tests at MIT, two students named Siddique and Hasanein, and then helium steam, those three. I'm still an old style presenter. I like getting up and pointing.

CHAIRMAN BANERJEE: We can get you a laser pointer.

MR. KELLY: Sitting in front of a laptop just doesn't - isn't -

CHAIRMAN BANERJEE: As long as the court reporter can hear, that's okay.

MR. KELLY: Oh, that's right.

CHAIRMAN BANERJEE: That's all that really matters.

DR. WALLIS: So everybody uses about the same tube size. How does that compare with the ESBWR tube size?

MR. KELLY: As far as I remember, the UCB-Kuhn is prototypic.

DR. WALLIS: Okay.

MR. KELLY: Because these tests were sponsored by GE.

MEMBER CORRADINI: If I may break in. Joe, just to follow on Graham's question, this - to get back to the bigger picture, if I understand it, the periphery tests were done literally so that you could get an in-tube condensation heat transfer with noncondensables and they stuck with the same weight scale because it's pretty unclear as to when you go from essentially in-tube to an in-vessel phenomenon. Is that correct?

MR. KELLY: The part about doing the test to develop heat transfer correlations to use in their model for the ESBWR is correct. The choice of the length scale I can't say. I just know they tried to make the tubes prototypic.

MEMBER CORRADINI: Okay. Well, the reason I bring it up like that is the Dehbi experiments at MIT, the length scale there was only about 30 centimeters even though it was viewed as a vessel, whereas Tagami-Uchida from long ago were much larger vessels. So I'm only getting at that we're focused on something where length scales are matched to the actual design to get around the question of how to scale.

MR. KELLY: That's correct.

MEMBER CORRADINI: Okay.

CHAIRMAN BANERJEE: Yes, actually of course, as you get towards the bottom of the tubes your Reynolds numbers for liquid film will increase significantly.

MR. KELLY: Yes, actually on my previous slide, those Reynolds numbers were the tube exit. The film was the tube exit, the gas was the tube ventilate. Okay? So, here are the experiments again. The pointer doesn't work on white. So, you see the number of data points, and there's three different correlations, the Vierow-Schrock which was developed from the first set of UCB tests, the Kuhn-Schrock-Peterson which was the last set, and then the well known Shah correlation for pure steam condensation in tubes. And what you'll see if you'll compare the - I'm showing here the average error and the RMS error for each of these compared to all the data points in those sets. And let's just for example look at the Kuhn-Schrock model compared to their own data. The errors are very small, but you're fitting an empirical model to one set of data, so that's kind of what you'd expect. What is surprising is we're able to do almost as good with what I'll call a semi-mechanistic model in TRACE. Now, if you take a model, an empirical model and apply it to a different data set, like you take this model which was for lightly sheared laminar films and you apply it to a highly sheared turbulent film, the errors are pretty large, whereas again, the TRACE errors aren't too bad.

DR. WALLIS: That's the old TRACE?

MR. KELLY: No, that's the new TRACE.

CHAIRMAN BANERJEE: You'll show us the formative correlation at some point?

MR. KELLY: Yes. It's not one correlation. What it is, it's models for those five different categories I show, and I'm planning on showing you some of them. If we have time I can go ahead and try to - but you know, that's a 2- to 3-hour presentation, so.

CHAIRMAN BANERJEE: Right. The other thing you could do is just give us the backup slides. I mean, we've been reading this report, but you know, we've got thousands of pages to read through.

MR. KELLY: You don't have the TRACE

theory manual by heart now?

CHAIRMAN BANERJEE: Almost.

DR. WALLIS: It's okay. It doesn't really tell us applicability to the range of variables we're interested in because the - I don't know if RMS tells us much about where ESBWR fits on the statistical curve, on the tail of your distribution. Perhaps more - we need to know that.

MR. KELLY: Well, these NASA tests, that's pure steam, high flow rates - that's ICS, not PCCS. And I don't remember the degree to which the conditions overlap. However, both the UCB and the MIT tests were designed for -

DR. WALLIS: They top out the right range.

MR. KELLY: The MIT ones tend to cover about the right range. The Kuhn ones go a little bit past that. So for example, pressure goes from one bar to five bar, but we're not planning on having five bar inside the containment. So they tried to cover the range very well, but also bracket it.

So, for the pure steam test - actually, the point I really wanted to make here is that when you now go to air-steam condensation, again the empirical model gets its own database, it's very good. TRACE is equally as good. And if you then go to a test outside that database, TRACE is much better than the other correlations. So this just shows that comparison. So this is the calculated heat transfer coefficient versus measured. The error bars are plus or minus 25. This is the UCB-Kuhn pure steam data. You see it lines up very nicely except for a few points here, and those are right in the laminar turbulent film transition where we're going from one model to the other. We over-predict the heat transfer by a bit.

CHAIRMAN BANERJEE: That's interesting. So, you'd expect the laminar ones to be on the left-hand side, but lower.

MR. KELLY: Well, this has to do with the way the models in TRACE work. Oh no, I see what you mean. You're expecting -

CHAIRMAN BANERJEE: It's at the lower end of the - the turbulent along the -

MR. KELLY: Yes. Now, I'm overlaying on the same plot the results of the Kuhn air steam test. And you see again they line up very well with a slight bias, but there are some points out here where TRACE under-predicts fairly significantly. Those points tend to be at the end of the tube, and so then all the errors, you know, for the mass and energy as you integrate down from the top of the tube to where you're actually running out of steam, those errors have accumulated. That's part of it.

CHAIRMAN BANERJEE: Let me ask you a question here. Obviously you've now got a resistance on the gas side which becomes significant due to the accumulation of the air.

MR. KELLY: That model I'm going to show in detail later.

CHAIRMAN BANERJEE: Okay. So it's sort of diffusion control.

MR. KELLY: It's a mass transfer of conducted -

CHAIRMAN BANERJEE: Yes, you're going from a heat transfer controlled condensation to mass transfer.

MR. KELLY: Exactly.

CHAIRMAN BANERJEE: And that - is that sort of where the big errors arise?

MR. KELLY: Well, no, but close, because what you're - for these points, if you go and look at where they actually are, the local gas mixture Reynolds number is in the hundreds. So you're in the laminar regime. And if you back out from the data what the Sherwood number ought to be you expect it to be on the order of four because it's laminar in a tube, but instead it's on the order of 10 to 12 which doesn't make a whole lot of sense. So you have to think, well is it the data, or are we missing something phenomenologically? And I don't have any proof, but from what I've looked at, what I think it is is the persistence of turbulence, it's a history effect.

CHAIRMAN BANERJEE: Or it could be the stirring effect of the liquid film.

DR. WALLIS: Maybe it's not a steady flow.

CHAIRMAN BANERJEE: Well, you've got - that's what I'm saying, you've got large waves by then. The film is fairly thick and it could be stirring the -

MR. KELLY: That's possible as well.

CHAIRMAN BANERJEE: It's a classical problem.

MR. KELLY: My explanation also covers one other case though.

CHAIRMAN BANERJEE: Could be also fossil turbulence.

MR. KELLY: When we first looked at the PUMA PCCS condensers, and now these are quarter height so everything is shortened up, but the tubes are still a pretty good diameter. I don't remember. When they scaled that facility for a noncondensable conduction heat transfer model they were using the Vierow-Schrock, and that model is only - it's a function of the laminar film Reynolds number and a function of the gas - you know, mole fraction of noncondensable gas. It doesn't have any effect of the gas mixture Reynolds number in it. So when they scaled the facility, they didn't worry about the gas mixture Reynolds number in the PCCS tubes. They just wanted to get the right noncondensable gas concentration there. Well, so they actually come in laminar, if you look at the local Reynolds number. And our model, which is only a local conditions model, says our laminar Nusselt number should be 4, Sherwood number should be 4, and we under-predict that, just like we under-predict this. But if you go back and look at the facility, and I don't remember this exactly, but you know, size of the pipe coming in and what the Reynolds number of the pipe going into the header is, that flow is highly turbulent before it enters into the header. And now instead of the header being the size of the plant, the header is reduced because it's quarter height. So you have a highly turbulent flow coming in, going to the set of tubes. They're going to have a history effect and have some turbulent kinetic energy in them even though the local Reynolds number says hey, I'm laminar. So -

CHAIRMAN BANERJEE: How long - are other deviations occurring near the entrance or near the exit of the type?

MR. KELLY: In these tests where the Reynolds number was about 20,000, these are all at the exit.

CHAIRMAN BANERJEE: Right.

MR. KELLY: And that's one thing where the experimental error has been magnified because you're integrating it from the top of the tube to the bottom, but that's also where the local gas Reynolds number says it would be laminar, and like you said, that's also where the liquid film thickness is at its maximum.

CHAIRMAN BANERJEE: There are - I mean, there's a lot of evidence that the liquid film, the waves can stir up the gas. You get an effect even on laminar liquid film mass transfer due to that which gives you over 50 percent, 60 percent higher than you'd expect from Nusselt theory. So on the liquid film side I'm not sure. On the gas side, the same sort of effect is seen due to these large waves. So anyway, it's an interesting problem, but I don't know how germane it is to our prediction for ESBWR.

MEMBER ABDEL-KHALIK: I have sort of a big picture question. I'm trying to understand what it means for the TRACE model to either under-predict or over-predict the heat transfer coefficient by 40 percent. Is there a presentation later on where a sensitivity study will be done to look at the effect of that much deviation in various parameters on the overall plant response - on the calculated overall plant response?

CHAIRMAN BANERJEE: I guess we asked that question earlier and we said we're going to defer that to.

MEMBER ABDEL-KHALIK: I'm just trying to see, you know, how much I should keep track of these individual deviations.

MR. STAUDENMEIER: I don't think we have any sensitivities on that because what that really affects is going into long-term cooling and we don't have plant calculations doing that. We have some -

MEMBER CORRADINI: Can you speak up? And I didn't hear, you will have plant calculations?

CHAIRMAN BANERJEE: Joe Staudenmeier.

MR. STAUDENMEIER: Joe Staudenmeier. Our

plant calculations don't extend into the long-term cooling period where this becomes more important. We had some integral test facility calculations that do, but we don't have sensitivity on that. But essentially what you do if you're under-predicting a heat transfer coefficient or over-predicting, what you're doing is changing the temperature difference that it's floating by, and then you can translate that into a difference in t-sat and a difference in pressure if you want, and that's really the way you have to look at it.

MEMBER CORRADINI: I understand your explanation. I think Said's point is - I mean, this is exactly the sort of stuff the TH community should worry about, but eventually if we ever get back to an ESBWR subcommittee, somebody in that meeting is going to ask, this is all very good for the physics of it, but how much does it impact? And I think that's Said's major point. And I think downstream that's going to be a question.

MR. STAUDENMEIER: Yes, that will be a question. I think for long-term cooling that's really a question more for MELCOR than for TRACE since we're not responsible for the long-term cooling calculation.

MEMBER CORRADINI: Hold on there. That's

the - well, I guess - this is Mr. Chairman Banerjee's issue here, but that was part of the reason I think that we had this - or that he had had this meeting is that it was my understanding and Joe's response to my original question that you have to couple into the containment. That means TRACE has at least got to be able to do this. That leads us to the question of if you do it, how much sensitivity you have to these models. I mean, that's kind of going back to Said's original question, if I understood it.

MR. STAUDENMEIER: Yes, and you'll get to see some integral test predictions and I'll discuss sensitivities of that during that presentation.

CHAIRMAN BANERJEE: Mike, do you have access to the slides being shown?

MEMBER CORRADINI: They just came. I'm sorry, if I'm taking you off track just tell me to be quiet.

CHAIRMAN BANERJEE: No, no, it's fine, you're on track. You're not going to be with us through the day?

MEMBER CORRADINI: I go into budget crisis meetings and doctor things in about an hour and a half, and I'll come back to you after that.

CHAIRMAN BANERJEE: All right.

MR. KELLY: The one thing that I can say about the plant response, at least the last time I checked, things like the PCCS tube heat transfer condenser error - area were over-designed. There's a safety margin built in. So as we talked later, if you under-predict the condensation heat transfer, what happens is you just open up a little bit more of the area. Typically, this completes say about two-thirds of the way down the tube and the last third of the tube is pretty much all noncondensable gases. Now, you just move that interface down a little bit further until you condense all the steam. You know, all the steam up to the partial pressure corresponding to the temperature in the PCCS pool.

DR. WALLIS: So, sort of self-controlled.

MR. KELLY: Exactly. Which is nice.

DR. WALLIS: Now, as I read before, it seems that you fixed the driving force by using the partial pressure.

MR. KELLY: For the noncondensable gas effect, that's true.

DR. WALLIS: It may be okay when you have a very turbulent flow with everything well mixed in the tube, but when you're dealing with containment, the buildup of noncondensables is more governed by diffusion and you get this layer which snuffs out condensation in a way it does not do in a tube. So I wouldn't think that this model that you're developing for the tube would apply to these big air, big volumes and walls and surfaces.

MR. KELLY: Well -

DR. WALLIS: Is it the same model, or do you have a completely different model?

MR. KELLY: It's the same model except for how the Sherwood number is calculated. In that case, the Sherwood number is calculated by a natural convection correlation, and it turns out that it basically reproduces Uchida when you compare it to the conditions Uchida is for.

DR. WALLIS: Are you going to get to that later?

MR. KELLY: Yes.

DR. WALLIS: Okay.

MR. KELLY: So in summary, we developed the model. It's consistent with two-fluid. We take advantage of the solution and conservation equations, and the accuracy is as good as empirical correlations when it's compared against the database those correlations came from and it's superior to empirical models when you go to other databases.

CHAIRMAN BANERJEE: Now, within the two-fluid model carrier concentration fields for the noncondensables, you just get the average concentration?

MR. KELLY: Yes.

CHAIRMAN BANERJEE: So somehow you have to change that.

MR. KELLY: Into a concentration interface. And I will show you how we do that. Not in great detail, but I think in enough. So as I mentioned, we had to make changes to five different constitutive models. I'm going to talk about wall friction, a noncondensable gas effect. And you may say, why would you worry about wall friction? And I'll show you. You know, obviously it's important to the film thickness, but I'll show why I worried about it. Then I'll give you a sample of the TRACE assessment results, and these are the ones that I did as part of the developmental assessment. Other people will show you the ESBWR-specific ones.

DR. WALLIS: Now, does shear govern essentially rather than gravity? Gravity doesn't have much effect in these tubes.

MR. KELLY: For PCCS tubes it's gravity.

DR. WALLIS: It is?

MR. KELLY: Because the gas mixture velocity is relatively slow, so gravity is more important than interfacial drag. But if you go to ICS conditions where you have high-speed velocities, then you're back more to the traditional tube condensation stuff where it's a highly sheared film.

DR. WALLIS: Thank you.

MR. KELLY: And I'll show you a slide later which will kind of show that some. So, this is what I talked about earlier. If you look in the literature for a condensation heat transfer model, you get heat flux is equal to the heat transfer coefficient times t-wall minus t-sat. And then of course if there are noncondensables in here there's going to be some extra term for the effect of the noncondensables. But when you look at the two-fluid representation, we have a wall, a liquid and a vapor, co-current downflow from PCCS tubes, and we have three different heat transfer processes. First you have to remove heat from the liquid to the wall. You're also removing heat from the interface to the liquid, and this is really the primary heat removal that causes the condensation at the interface. If for example the vapor would have to be super-heated, you can also be transferring some heat to this interface which could be causing evaporation so you end up taking the net. So in the two-fluid model the wall heat flux is actually the wall to liquid heat transfer coefficient times wall temperature minus the condensate temperature. Condensation rate is the sum of those two interfacial processes divided by the latent heat. And this is something that Professor Banerjee asked earlier. This interface temperature which is really t-sat at the partial pressure of the steam, and it should be the partial pressure of the steam at the interface. But within the two-fluid numerical framework it's assuming that it's at the bulk vapor partial pressure. So I have to make an adjustment for that, and I make that adjustment explicitly and then add in a heat transfer resistance here to account for that. And I'll show you how that's done.

DR. WALLIS: TNI is the saturation temperature at the bulk partial pressure?

MR. KELLY: In reality, no.

DR. WALLIS: Shouldn't it be at the actual -

CHAIRMAN BANERJEE: That is the issue.

MR. KELLY: That's exactly what I just said. And so to account for that I'm going to use a mass transfer conductance model and locally calculate what it really is.

DR. WALLIS: So you're going to do it right.

MR. KELLY: Yes.

CHAIRMAN BANERJEE: The only way you can do that is to get the concentration of the interface normal.

MR. KELLY: And that's exactly -

CHAIRMAN BANERJEE: And then you can always calculate.

MR. KELLY: So with this framework, the first thing we're going to do, you know, the model first. It had to be for both pure steam and steam-noncondensable gas mixtures. It had to be applicable to both falling and sheared films. So now I have to worry about both gravity and interfacial, and that's what makes just picking one correlation from the literature kind of difficult. So I tried to put together a model that would handle all of this. For film thickness, that's a function obviously of gravity, wall friction, and interfacial shear. As we've discussed for PCCS conditions, it's primarily wall friction. And these are the other models we needed. I'm going to talk about the wall drag now. And here we're going to talk first about what happens with the legacy code version which in some of the slides you'll see with the plots labeled Original TRACE. So, the plot on the left-hand slide is the phase velocity in meters per second versus the axial position. This is for a pure steam condensation run. You come into the tube, the blue curve is the vapor velocity. Comes in at 6 meters per second. As you condense the vapor, the mass flow rate hence the velocity decreases. Okay, well and good. Look at what TRACE was calculating for the liquid film velocity. It's not right. It's actually unphysical. You don't have a thin film along the wall falling at 5 meter a second. You know, it just doesn't happen.

DR. WALLIS: How did TRACE ever predict that?

MR. KELLY: Well, it predicts it because this is the legacy TRACE model. Because it partitions the wall drag between liquid and vapor. So - and it uses the void fraction to do that. When you have very high void fractions, which you are with thin liquid films, there's basically no wall drag on the liquid.

DR. WALLIS: Pretty cool.

MR. KELLY: Yes. And it was the same thing in RELAP when we started doing RELAP for the 8600 and the SPWR back 13 years ago. This was something we had to look at in RELAP too. So it was something that I expected to stumble on in TRACE as well.

CHAIRMAN BANERJEE: The history of this goes back to Solberg who partitioned these things.

MR. KELLY: You know you're right at the limits, you know, and it's - and they weren't thinking of trying to calculate a film thickness. That wasn't what they were worried about. They were worried about blowing down a plant, large break LOCA, what's the PCT. You know, not going to criticize what someone did 30 years ago because from their focus they were doing something that was reasonable, but it's not reasonable when it's applied to this.

DR. WALLIS: It just shows why you shouldn't use a code without understanding what's in it.

MR. KELLY: I agree with that. The plot on the right-hand side is the film thickness in millimeters versus the axial position for the same calculation. The legacy TRACE one is the blue curve. This red one is a hand calculation where what I did is just simply took the liquid condensate flow rate, divide - you know, and solved gravity versus - so it's a Nusselt thing in effect. So there's no interfacial drag on this. You notice the film thickness is between 100 and 150 microns, which is kind of what you'd expect.

So, the revised model which is what's now the default model in TRACE. Parameter film is just the parallel plate formula for smooth laminar film and as Professor Banerjee stated, that will slightly over-predict the film thickness because it's neglecting the effect of ripples. Rather than put that into the wall drag, I'm going to make that up when I go to the wall heat transfer model. So rather than it being K over delta, it'll be K over delta times 1 plus Reynolds number to a power, you know, to make up for the rippling on the film, where that Reynolds number is the film Reynolds number.

DR. WALLIS: So this is restricted to down flow?

MR. KELLY: This wall drag model -

DR. WALLIS: Well, it assumes a sort of linear profile.

CHAIRMAN BANERJEE: This is just a laminar prediction.

MR. KELLY: That's where this comes from, but this model is used everywhere except as long as you have a film. When the film grows large enough so that we're getting to void fractions greater than 0.9 and we're starting to look more like pipe flow, then the laminar goes to 60 and over Reynolds number.

DR. WALLIS: But I'm just saying in up flow, or in the sort of low velocity region you can get the liquid going down at the wall and up in the middle. So then this wouldn't make any sense.

MR. KELLY: No, that's true. Fortunately the PCCS - co-current down flow. And just, you know, an explicit approximation, the Colebrook-White for the turbulent film, and in a power wall combination. You know, pretty standard stuff.

CHAIRMAN BANERJEE: Why did you have to get such a - go back to the previous slide. I mean, there are simpler ways to make things turbulent. Why did you have to go through?

MR. KELLY: Oh, I could have used the Lazius. That would work equally well.

CHAIRMAN BANERJEE: Lazius or something like that.

MR. KELLY: But I was - what I was trying to do was remove uncertainty where I could, you know, and that's not that - it's an explicit approximation. It's not that hard for the computer to evaluate it, and I can also start using the same models across the board, rather than have a turbulent wall drag model that we only use for liquid film and PCCS tubes. I can use that for two-phase flow once we get to, you know, two-phase flow in the pipe.

CHAIRMAN BANERJEE: So that's why the epsilon is there.

MR. KELLY: Right.

DR. WALLIS: Does epsilon over D have any effect in PCCS tubes?

MR. KELLY: We - no. None whatsoever.

DR. WALLIS: Why did you put it there?

CHAIRMAN BANERJEE: Well, as I said, to make it more universal.

MR. KELLY: Right. So, this is a non-dimensional film thickness plus the film thickness divided by the Nusselt parameter, you know, the little viscosity squared over G delta rho, all over one-third power. Plot against the film Reynolds number. I found all the data that at least when I was looking at that time I could, and I had to digitize a lot of old -

DR. WALLIS: This is with no vapor doing anything, right?

MR. KELLY: Right. This is falling films. Because that's the first thing. I mean, I want to make sure I can at least get the film thickness for the falling film. So I pulled as much as I could out of the literature so I could go from very low parameter to rippling films to highly turbulent films, okay?

DR. WALLIS: I'm just looking to see if the best data isn't the old data.

MR. KELLY: In some cases. People took more time. There's some old data that is a little suspicious.

DR. WALLIS: It was out on the West Coast, wasn't it?

MR. KELLY: Well, maybe they were measuring the wave crests, you know, I don't remember. So that's the same data, but I took away the legend so I could easily overlay the TRACE calculation. So it's a TRACE calculation with the interfacial drag set to zero and you see it matches very well. So now I have a reasonably good approximation for what the film thickness is going to be and I think I can now do say laminar film condensation equally as well as Nusselt model without some of the restrictions that are built into Nusselt model like, you know, uniform plate temperature.

CHAIRMAN BANERJEE: So this is on a mock

scale, of course.

MR. KELLY: Right. That's why I used the word "reasonable." But it's - the error versus the data is smaller than the scatter in the data. It's hard to get better than that. So, that finishes the wall friction section. Now what I'm going to talk about is the noncondensable gas effect, and that's where we're going to calculate the -

DR. WALLIS: I have a question for you. These tubes are vertical?

MR. KELLY: In the - yes.

DR. WALLIS: They're vertical? Because the tube is not quite vertical. There's quite a big effect.

MR. KELLY: There's a section - I'm trying to think because this is open session. There is a header above and below. Some of the tubes go straight down. There are tubes that have an incline section coming out from the header, but for most of their length they're vertical, and they're inclined going back into the other header.

DR. WALLIS: Because all of this assumes uniform film around the tube.

MR. KELLY: Exactly. So it would mis-predict if the film thickness is the limiting heat transfer resistance, which it would be for pure steam condensation. Then you would mis-predict in that incline section, that's correct.

CHAIRMAN BANERJEE: I guess the concern in a very macroscopic way is accumulation of noncondensables towards the exits, and if it's not completely vertical, how does this clear? You know, now it has to somehow get pushed out. Is this a sort of a sporadically occurring phenomena, or is it sort of continuous drag sporadically happening where you clear it, and you accumulate it, and you clear it.

MR. KELLY: I don't remember all the details. I know from - because this has been years ago the last time I did like a calculation for PUMA, but when we did you see it in the test data and you see it in the code calculations. It does just what you're saying. You know, the noncondensables will build up, condensation rate will slow down, pressure will go up a tick until you get just enough to purge it all, and so it's a very oscillatory phenomenon.

CHAIRMAN BANERJEE: And the code predicts that?

MR. KELLY: With a reasonable, you know, guess as to the amplitude and frequency. Now, they don't start at the same time, they're not in a locked phase, that's not going to happen, but yes.

CHAIRMAN BANERJEE: Authoritatively it gets the reality.

MR. KELLY: Because of just what was said earlier. It's a self-regulating system.

CHAIRMAN BANERJEE: Right. All right.

MR. KELLY: So now we'll talk about the noncondensable gas effect and how we end up calculating the mole concentration of the gas at the interface which then gives us that. So what I'm going to do is take a mechanistic approach similar, very similar I might add to the mass transfer conductance model which was described by Kuhn, Schrock and Peterson, and it actually came from Kuhn's PhD thesis. So if you go and get a copy of his thesis you'll see that it's almost exactly this model inside it, and then he makes the point that this model describes his data more accurately than his empirical correlation. I kind of thought, well that sounds like a good idea.

These are the same assessment things I've talked about before, except with the containment wall condensation. I will later show a comparison to the Uchida formula and the MIT-Dehbi test. They were - this was all compared to the Wisconsin Flat Plate Test. When I was preparing the presentation I didn't have those plots, but I can at least talk about it. So, you start out - we need the liquid interface heat transfer because that's - the amount of heat you remove from that interface is what really is controlling this. As the liquid to interface heat transfer coefficient, t-interface minus t-liquid.

CHAIRMAN BANERJEE: You're going to show us what you used there?

MR. KELLY: Yes. Because you will notice that's what it says. That's the unknown here. For the heat flux from the gas mixture to the interface it has two components, one is condensation and one is sensible heat transfer. The condensation is obviously - heat flux is obviously the condensation mass flux times the latent heat, and the sensible heat just looks like your normal heat transfer. This is typically negligible. It's built into the model but you could leave it out and you wouldn't see it. So what you're going to do is equate -

CHAIRMAN BANERJEE: That's on the gas side, obviously.

MR. KELLY: Yes. You're going to equate the heat removed from the interface to the heat provided to the interface, and that gives you a system of equations that you solve with interface temperature. So what are the models that go into those equations? The condensation mass flux is given here. The letter B here is the mass transfer driving potential and these are just the mole fractions. This is the vapor in the bulk and this is the vapor at the interface, and this is why you have to iterate because the temperature is a function of the mole concentration and you know, that's not straightforward. The mass transfer, you know, this is the Sherwood number and this ratio of the molecular weights - this is molecular weight of the mixture at the interface to the molecular weight of the mixture, and in this case it's to the 0.4 power. I should say, what I'm talking about now is for the tube model. I'm going to make a change to this when I get to the wall.

DR. WALLIS: That must be a small effect of molecular weight.

MR. KELLY: It's - what this takes account of is the change in the properties as you go from the bulk across the boundary layer. So it's, you know, a 10-20 percent kind of effect. It's not major, but it's not negligible.

CHAIRMAN BANERJEE: Steam to air, or steam to whatever.

DR. WALLIS: The big thing is beta, isn't it? Isn't beta the big deal? Beta is the concentration-polarization effect.

MR. KELLY: Well, it's not as large as you expect.

CHAIRMAN BANERJEE: I would say that somewhere you have to tell us what HLI is.

MR. KELLY: Well, that's actually my backup slides, okay?

CHAIRMAN BANERJEE: Because that it seems to me is where people get things wrong.

MR. KELLY: We'll talk about that. Let me finish that - this part and bring that back up. For most of these conditions in the PCCS tube you're limited by the mass transfer driving potential, and what happens in the liquid film is inconsequential. For other things, it's different.

CHAIRMAN BANERJEE: I can believe that because you're getting a fairly thick noncondensable there.

MR. KELLY: Yes, because we're up at, you know, mass fractions 20-30 percent, you know, so the noncondensable is the major potential.

CHAIRMAN BANERJEE: So that correction is just a correction for properties?

MR. KELLY: Correct. It's like a

viscosity ratio and a Nusselt number, you know, or if you're doing a Nusselt number for gas-heat transfer you do a temperature ratio. Where it really comes from, if you write down, you know, everything in the Sherwood number, all the property groups and then you look at how they are - how they change as a function of the gas concentration as you go from the bulk to the interface, you can do a first order correlation of that property group as a function of the molecular weight. And the - it comes out in the molecular weight to be 0.4 power. This was in a textbook by, I believe it was Eckert, and I just pulled it out.

CHAIRMAN BANERJEE: Yes, I think -

DR. WALLIS: I'm just puzzled here because I see beta and B, but beta is log 1 plus B over B, so doesn't the B cancel out and you just get log 1 plus B? Is that true?

MEMBER CORRADINI: Are you guys talking about the suction effect? Is that what you're going through?

DR. WALLIS: We're talking about the mass transfer and the polarization.

CHAIRMAN BANERJEE: Suction and blowing.

MEMBER CORRADINI: Yes, yes, yes. I don't think it cancels out, Graham.

DR. WALLIS: Well, it does.

MEMBER CORRADINI: No.

DR. WALLIS: It does in the equations he's presenting on the board.

MEMBER CORRADINI: Well, then the equations are wrong.

DR. WALLIS: That's what puzzles me. I expected to see an exponential, because once it builds up in the wall it has a huge effect.

CHAIRMAN BANERJEE: I think that B may be - maybe within the log. Is that B within the log or outside the log?

MR. KELLY: Well, it's a natural log of 1 plus B divided by -

DR. WALLIS: It's within the log. It's inside the log?

CHAIRMAN BANERJEE: No.

MR. KELLY: The denominator is not. The numerator is.

DR. WALLIS: So it cancels out when you put it in the -

MR. KELLY: Yes. And then this ends up being times the natural log of 1 plus B.

DR. WALLIS: So that's a very weak effect of B.

MEMBER CORRADINI: That's what he said at the beginning, yes. It's about a - the suction effect is about a 10 percent effect. It tends to get bigger than just a few percent as the pressure rises. So atmospheric pressure is about 2 percent effect at a few bars, it's a 10-ish percent effect or something of that order if I remember this from the experiment.

MR. KELLY: And you know, I understand what you're saying, but these are literature type things and it works.

DR. WALLIS: So I remember doing calculations in Sherwood - like homework problems where the effect could be very big in some mass transfer operations.

MEMBER CORRADINI: Right, but I think - I think Joe's point at this juncture is with these concentrations of a large amount of noncondensable and at these pressures, we're talking, you know, 5 to 15 percent effect, of that order.

MR. KELLY: That's correct.

CHAIRMAN BANERJEE: Now, one of the things of B is really the driving force for mass transfer, right?

MR. KELLY: That's correct.

CHAIRMAN BANERJEE: And the Sherwood

number is basically the mass transfer coefficient.

MR. KELLY: Yes.

CHAIRMAN BANERJEE: Corrected for -

MR. KELLY: Once you bring in the diffusivity and the density over the 2.

CHAIRMAN BANERJEE: Right. And gamma is really the mass flux in some sense.

MR. KELLY: That's correct.

CHAIRMAN BANERJEE: Okay.

MR. KELLY: So what we're doing is solving for the diffusion of steam molecules - the interface and in effect the air molecules back away from the interface. It's that equilibrium situation that gives you this effect.

CHAIRMAN BANERJEE: Yes, so it's sort of counter diffusion, if you wish?

MR. KELLY: Exactly.

DR. WALLIS: Something doesn't make sense. Without any beta, gamma is proportional to B which makes sense.

CHAIRMAN BANERJEE: What makes sense?

DR. WALLIS: When you put in the beta, the B's cancel, you get log 1 plus B. It looks as if the B has less effect because it's a mock.

MR. KELLY: Yes. There's also one other

thing that's dependent upon the mole concentration: mixture gas density.

CHAIRMAN BANERJEE: It's true, but where I'm losing this is that without all these complexities gamma would be proportional to the Sherwood number row D divided by D into B.

MR. KELLY: Yes.

CHAIRMAN BANERJEE: Okay. So why does this blowing effect cancel B which is your driving potential? Somehow I'm not understanding something. Imagine that it was very, very slow, the process, so there's no blowing. I mean, this is exactly what you would expect in a fairly thick gas layer. The flowing thing is a small effect.

DR. WALLIS: But if you expand log plus B you get B as the first term, don't you?

MR. KELLY: Exactly. Exactly, thank you.

CHAIRMAN BANERJEE: So, the empiricism is hidden, so the reality of that physics is obscured by the blowing factor. Which is that B is really - because as you go to very low condensation rate, that blowing factor is 1 basically. It's nothing.

MR. KELLY: So you know, I took this kind of as the - and I never looked into it. I looked into the empirical parts of it here, but the other part I just -

CHAIRMAN BANERJEE: What happens is beta becomes 1 as you blow slowly.

DR. WALLIS: Beta becomes zero, doesn't it? Or 1.

CHAIRMAN BANERJEE: No, if it comes here it would be -

MR. KELLY: It becomes -

DR. WALLIS: - do I find this beta?

MR. KELLY: Yes.

CHAIRMAN BANERJEE: And what -

MR. KELLY: Calculated, you know, for a very specific situation, and I'm sure it's exactly the same formulation.

CHAIRMAN BANERJEE: So the classical formulation for the Sherwood number is of course 3.66. What is this Gnielinski or something?

MR. KELLY: Okay. Remember, we have to go laminar to turbulent conditions in the gas mixture. So what you're seeing is 3.66 for laminar flow in the tube. Gnielinski, this is actually a turbulent heat transfer flow that I'm using as a mass transfer analogy. So I'm evaluating it as a function of the Reynolds number and the SPT number instead of Reynolds and Prandtl.

Yes. Actually, this is the one that everyone recommends nowadays because it's much more accurate. You know, it's more accurate at higher Reynolds numbers, but in particular there's - you know, it's the series of those Russian ones that start out as, you know, functions of the friction factor. This one has a Reynolds number -1000 stuck into it so it - no, it's good because it fares into the laminar region better. Because if you just take the defaulter and this laminar Nusselt number and plot them like this, in this region you over-predict significantly, whereas the Gnielinski fares in and matches the data in that transition region better.

CHAIRMAN BANERJEE: The reality is that that transition rate region, the reason none of these really work very well is even the data has a huge scatter because you get slugs of turbulence, slugs of - it's a complicated business.

MR. KELLY: Yes, and if you can't measure, you can't correlate it.

CHAIRMAN BANERJEE: Yes. So this is sort of a fake correlation to make it work. But nevertheless it helps the core which is what matters.

MR. KELLY: So this is really the meat of it, is how do you calculate the condensation mass flux to sensible heat flux. It just looks like the same thing. You notice some - here it's Reynolds and Prandtl. And I'm not going to spend time on this because it's inconsequential.

CHAIRMAN BANERJEE: But eventually I do want to get back to the HLI or whatever.

MR. KELLY: I'm almost finished. After I show the data comparisons I have some backup slides and we'll go into that.

CHAIRMAN BANERJEE: Because that will become important in some cases where you have high shear. If the gas is very turbulent, you see, then the gas resistance will be relatively small.

MR. KELLY: I'll go you one better. I'm going to go to pure steam condensation.

CHAIRMAN BANERJEE: Oh, pure steam, yes.

MR. KELLY: Then all you've got is that, the HLI and the H-wall liquid.

CHAIRMAN BANERJEE: Right.

MR. KELLY: And it's those two resistances that govern it. And you'll see an example of that. We don't do it perfectly, but it's not horrible.

CHAIRMAN BANERJEE: Yes, what worried me is say I remember the condensation of steam bubbles. If you don't get that HLI right you get a completely wrong condensation rate. I think you showed this once in a slide.

MR. KELLY: And fortunately in this case - the interfacial heat transfer is pretty - typically if you get it to the right order of magnitude you're pretty happy. But fortunately in this case that's not driving the resistance, the finite resistance, so we can model this better.

CHAIRMAN BANERJEE: Well, what you're saying it doesn't matter in this.

MR. KELLY: Right.

CHAIRMAN BANERJEE: Okay.

MR. KELLY: Once you go above half a percent weight fraction in noncondensable gas, that takes over. So this is how I modified this to work for a vertical wall. Same formula, except now the Sherwood number and the exponent on the ratio of molecular weights has changed. The Sherwood number for natural convection, this is just the standard correlation natural convection on a simple vertical wall. Function of the Grashof number, the link scale. This is for turbulent film having turbulent conditions, so the link scale doesn't really matter, it ends up getting canceled out. And -

DR. WALLIS: I'm sorry. This XV, is it

null fraction of the vapor?

MR. KELLY: In the bulk. That's correct.

DR. WALLIS: And XVI is at the interface where it's less?

MR. KELLY: That's correct. That's a vapor, yes.

DR. WALLIS: So B is negative?

MR. KELLY: Let's see, that's positive and that's negative, yes.

DR. WALLIS: B is not negative. You don't - there's a minus up there in the gamma too. It's a funny way to do it. Okay.

MR. KELLY: I agree, it's a funny way to do it.

DR. WALLIS: So log 1 plus B is log 1 minus something?

MR. KELLY: 1 minus a number that's less than 1.

CHAIRMAN BANERJEE: But the gamma is the mass flux out of the phase, is it? The wave is extended, or is it the mass flux into the phase? I mean, physically when you do these equations it's the N dot row K into BK minus VI, so it's out of the phase, but you reverse the sign I think.

MR. KELLY: Yes. So negative means you're condensing, and gamma will be negative here because as we discussed earlier the B's cancel. The B's cancel and you've got 1 minus a small number, and so -

DR. WALLIS: When you're evaporating is not a problem. When you're condensing you really build up more condensables. It's okay.

CHAIRMAN BANERJEE: I think the signs must be right, or if not it's just an oversight.

MR. KELLY: Or it's a typo on my slide, but I know it's right in the code, otherwise it wouldn't work. So, this is simple natural convection from the vertical wall, you know, using again the heat transfer, mass transfer analogy. This exponent C I calculated and I explained how I did that before. I looked at the property groups in this and I correlated this function of the molecular weight, and as you'll see this tends to work out great.

CHAIRMAN BANERJEE: So the length scale cancels - the Grashof, that has to be the vertical length scale, right? L.

MR. KELLY: Right, in which case you then turn that into a vertical length scale. This length is cubed to the one-third power. It's first power and it cancels. That's one of the nice things about turbulent convection. Otherwise -

CHAIRMAN BANERJEE: The inner scale in this problem.

MR. KELLY: That isn't a code where you do is it a vertical length scale, when you have a node is it the node length, you know. Fortunately they cancel and I don't have to worry about that. And so what we do in the model is we just simply take the maximum of a natural convection Sherwood number and the force convection, and that way it works either for the wall or for the tube.

CHAIRMAN BANERJEE: Can you go back to that, please? So you have some sort of a switch here in the code?

MR. KELLY: And the switch is by taking a maximum, because that's nice and continuous.

DR. WALLIS: Did I miss something? How do you know what XVI is? Is it written somewhere? How do you know what XVI is?

MR. KELLY: Well, I can't - it's right here.

DR. WALLIS: That's temperature. That's not concentration.

CHAIRMAN BANERJEE: Well, it is related through TI and some sort of Henry's law -

DR. WALLIS: Oh, I see. This is another

equation which you have to tell us -

CHAIRMAN BANERJEE: There's an equation.

MR. KELLY: This is the equation I solve, but the guts of this equation are an iterative procedure to find the temperature which is a partial pressure of the steam at the interface which is the mole fracture.

CHAIRMAN BANERJEE: We're running slightly behind schedule, so let's -

DR. WALLIS: Okay, let's move on.

CHAIRMAN BANERJEE: I think we understand the procedure.

MR. KELLY: So I've already shown this. This just tells you the average RMS error again. This is the steam air. This you haven't seen, which is steam helium. This is calculated versus measured. It does very well except again for those cases I told you about in very low gas to mixture Reynolds numbers. Now, everything I've shown so far was actually calculated basically by hand, you know, in a spreadsheet, checking the model out before I put it into TRACE. Now I'm going to show a sample of what happens when we actually put the model into TRACE and run it. I'm going to show presets. This is UCB-Kuhn steam only, NASA Goodykoontz, that's steam only, and for noncondensable gas effect the air steam or Kuhn. So you notice here I go from a pressure of 1 to 5 atmospheres. The film Reynolds number, that changes just because you get higher, more condensation. So you have warmed up the flow at the bottom. That's the turbulent film, you know, highly sheared flows, piercing. For noncondensable gas I picked a pressure of 4 bar because I had a very nice parametric on that, on a noncondensable gas mass fraction. I'm going to show you results from 1 to 40 percent mass fraction.

MEMBER ABDEL-KHALIK: When you say a TRACE model for these experiments, are you actually simulating the test facility?

MR. KELLY: Yes, which is just in this case a single tube. So, this is the pure steam laminar film, so these are the Kuhn steam tests. Calculated heat transfer versus measured. This is the original model. I've got all five tests here going from 1 to 5 atmospheres. You can see the comparison is fairly miserable, except for a few points where we'll assume the code got lucky. With the revised model which is what the default model now, it's variable PCCS condensation model here, you see a very, very close calculation just for a couple of points.

CHAIRMAN BANERJEE: Which is what you saw in the film thickness.

MR. KELLY: Well, it's also nice to know that when you stick it in the code you get the same answers you got in the spreadsheet.

DR. WALLIS: So the higher noncondensable fraction is way down on the left.

MR. KELLY: We haven't gotten to that yet.

DR. WALLIS: Haven't gotten to that.

MR. KELLY: That was a pure steam. On Page 27 this is pure steam.

DR. WALLIS: Okay.

MR. KELLY: And this is pure steam also, showing you the heat transfer coefficient and its axial trend. This was the old model and this is the new. The TRACE calculation is the red curve, notice the very large effect at the inlet. This is the liquid film thickness. The plot I showed before, that's the old model. This is the hand calculation, ignoring interfacial drag. The yellow curve is what was calculated by TRACE which has both the wall drag and interfacial shear in it.

DR. WALLIS: Well apparently if you used this one you'd do about as well.

MR. KELLY: Exactly.

DR. WALLIS: So you don't need all that

stuff.

MR. KELLY: Well, this is pure steam.

CHAIRMAN BANERJEE: But he wants to keep things that he could drag in.

MR. KELLY: You need it for this. This is the highly sheared turbulent film and these are two different tests that an orange line is the original TRACE model, blue line is the beta, yellow line is the new model. You notice the new model does not reproduce -

DR. WALLIS: That's interesting because I think Goodykoontz had such high velocities that he probably entrained some liquid.

MR. KELLY: Could be because I think we got about 100 meters a second or something. And so what you'll see is we under-predict near the inlet of the two where the film is laminar, and once it becomes turbulent we do pretty well.

MEMBER ABDEL-KHALIK: If I go back to slide 28, why is this any different than the comparison made by the model developer in his own thesis?

DR. WALLIS: You mean Kuhn?

MEMBER ABDEL-KHALIK: Why would it be?

MR. KELLY: In this particular case

there's a difference because the model developer had an empirical correlation for the heat transfer, okay? Just wall to t-sat. In this case, this is within the numerical framework of TRACE and it's using the calculated film thickness as the characteristic dimension. So you've got -

MEMBER ABDEL-KHALIK: I'm just trying to understand if this is simply a circular argument. You're using a set of external data, a model based on a set of external data, you put in the code, you go back and use the same code to model the same experiment, then you get good comparison. You say voila, this is great.

MR. KELLY: Well, that's the reason why you should also look at other databases.

CHAIRMAN BANERJEE: Are you actually doing that, or you're putting in the wall friction and the interfacial friction, calculating the film thickness as a part of the process?

MR. KELLY: That's exactly what we do.

CHAIRMAN BANERJEE: And then using a wall heat transfer coefficient and interfacial heat transfer coefficient which you haven't shown us yet.

MR. KELLY: Right.

CHAIRMAN BANERJEE: But, so it's not, he's not using that correlation. He's actually doing it right.

DR. WALLIS: Doing it from fundamentals.

CHAIRMAN BANERJEE: Yes, correctly.

MR. KELLY: To the best that I can, yes.

CHAIRMAN BANERJEE: I don't see that you're using anything wrong, other than the data. He's putting in wall friction, interfacial friction.

MR. KELLY: Yes, there's one thing you haven't seen and that's the heat transfer coefficient used between the wall to the liquid and the liquid to the interface.

CHAIRMAN BANERJEE: He's asking for that, of course.

MR. KELLY: For the laminar - and for the laminar film I use a correlation recommended by Kuhn in his data.

CHAIRMAN BANERJEE: That's -

MR. KELLY: But, there's a difference in what we use for the film thickness. He used a Nusselt kind of film thickness with a multiplier on it for the effects of interfacial shear. So he had a separate way of calculating the film thickness that he used. I used a solution of TRACE equations with different models for wall drag and interfacial shear to get me my film thickness. And in this particular case, the film thickness is it. You know, it's pure steam condensation, the films are laminar, they're a little wavy, but the waviness is a 20 percent kind of thing.

CHAIRMAN BANERJEE: So if I understand you, the heat transfer coefficients that you're using, we haven't seen them so we don't know what you're using, but one set is based - all heat transfer coefficients are based on some form of heat transfer enhanced due to the ripples or whatever which comes from Kuhn's work.

MR. KELLY: And I'll show you that compared to other models very quickly, as soon as I finish going through this.

CHAIRMAN BANERJEE: Okay, so let's defer that till we come to that then, and we take up Said's question as to whether it's circular or not at that point.

MR. KELLY: So this is compared to the air steam data -

DR. WALLIS: Excuse me. Can you do the Goodykoontz measure pressure drop too? What if you compared that? That's another check on your model because you're doing wall shear. You just focused on heat transfer?

MR. KELLY: I focused on heat transfer. I - now we're going back six years.

DR. WALLIS: Yes. With Goodykoontz you're going back 40.

MR. KELLY: Yes, well six years in my memory might be 40 to some. I, you know I think I looked at it, but you also - it's not just heat. It's hard to think up here, but remember you're also decelerating the vapor because you're condensing it all. So you have a very large pressure drop because of that, because of the acceleration.

DR. WALLIS: Pressure rise.

MR. KELLY: Right, excuse me. So it makes getting the wall drag kind of. So this is the air steam results going with mass fractions of air from 1 percent to 40 percent, and you can see it looks very good. These are TRACE calculations solving the model that I just showed. Now, I'm just - some quick results from the wall condensation. This is heat transfer coefficient versus noncondensable mass fraction. The blue diamonds are evaluated using the Uchida correlation and Uchida is kind of the base for containment analysis. And the red line is the TRACE calculation.

CHAIRMAN BANERJEE: That's using the

Grashof?

MR. KELLY: It looks - yes, that's using the Grashof. There's a really good paper, I don't remember the author's name, but the title was something like - because the Uchida correlation is a very simple empirical thing. You wonder how it can ever work. And the title of the paper was something like The Physical Basis Behind the Uchida Correlation. And what he did, he started with a model that looks just like mine, does some simplifications like ignore the heat transfer resistance of the liquid film, turning it into just a mass transfer model, makes a couple of simplifications in that, comes up with a property group and shows how that property group is like a coefficient in Uchida to a certain power - under certain pressure, excuse me. So what he makes the point is he has a fundamental basis, even though the original author didn't make that point, but it's only valid over a fairly narrow pressure range.

DR. WALLIS: This is if there are no drafts in the rule. Fill in with all these natural convection things is in reality almost always something else going on that stirs things up.

MR. KELLY: And I don't have it to show you, but I'll comment on that in just a second.

MEMBER CORRADINI: If I might just break in. Graham, I think in this regard I think - I'm not sure if Joe, but I seem to remember this was Whitley from UCLA or Denny Mills, or Mills or Denny from UCLA, but I think the point is because it's so simple and because it's natural circulation, this minimizes the heat transfer to the wall which maximizes the pressure, and that's why Uchida has always been the default heat transfer coefficient that's used in all containment heat transfer. It tends to maximize pressure.

MR. KELLY: I think that's exactly right, Mike, because it's - I think it's most correct at low pressure -

MEMBER CORRADINI: Yes.

MR. KELLY: - and as you go to a higher pressure it underestimates.

MEMBER CORRADINI: And the reason it underestimates, Graham, even if there was no circulation is back to your point about the suction effect. At higher pressures the suction effect goes from a few percent effect to 10 or 20 percent effect and it starts deviating from what is measured pressures.

CHAIRMAN BANERJEE: So Mike, this comment of yours, if you carry through to the containment over-pressure calculations, what would you say then?

MEMBER CORRADINI: I hesitated talking about this because I had this bad feeling you were going to go in another direction. I think staff is forcing all the applicants to use traditionally Tagami-Uchida and all their stuff. I mean, you have to check with Almeida and Francesca to make sure I'm not misstating this, but that is historically what staff requires the applicant to do.

CHAIRMAN BANERJEE: So if you have a few drafts around does this mean that we get sort of a conservatively high containment over-pressure?

MEMBER CORRADINI: For calculations where you are purposely trying to get a high containment pressure -

CHAIRMAN BANERJEE: This is fine, but where you're trying to get a purposely low containment pressure.

MEMBER CORRADINI: I don't want to even dare to say that. I think we should bring in the staff and have them explain that.

CHAIRMAN BANERJEE: All right. Carry on. And that's another subject which doesn't -

MEMBER CORRADINI: Yes, sir. Yes, sir.

MR. KELLY: So this is the condensation heat transfer coefficient versus air mass fraction for the MIT-Dehbi test data. And as Professor Corradini said, this is a relatively short heat transfer surface to spend in a large containment volume where they vary the pressure from 1.5 atmospheres to 4.5 and a very wide range of air mass fraction. And the blue symbols are the three different pressures. That's a fit, a curved fit to the data, because he had a lot of data points. And the red lines are the TRACE calculation. You notice there is an under-prediction at low pressure, but it matches very well at the 3.0 and 4.5 atmospheres. The plot I don't have in comparison to is the University of Wisconsin Flat Plate Test. My apologies, Professor Corradini.

MEMBER CORRADINI: No, don't apologize.

MR. KELLY: And so we looked at the vertical ones. And so here the air vapor mixture is forced convection, not natural, and they use two different velocities, or at least two that we looked at. One was 1 meter a second and the other was 3. The model I've shown you compares very well with the 1 meter per second test. It under-predicts by about 40 percent for the 3 meter per second test. And because of that, we built in a sensitivity parameter so that -

to make it easy for people to do sensitivity studies on the wall condensation so that they could quantify the effect should, you know, if they want to.

CHAIRMAN BANERJEE: So, now, let me understand the data which you didn't show sounds interesting. Why do you physically under-predict the 3 meters per second? Is there some aspect like turbulence in the -

MR. KELLY: In the gas vapor mixture?

CHAIRMAN BANERJEE: - gas, yes. Which you don't really account for except through - so you don't have a free plus force convection sort of expression for the heat transfer?

MR. KELLY: Well, we do, but in this particular case because of the way the Reynolds number came out - let me back up. The model was done in two stages. At one point in history there was the PCCS model and there was the wall model, and at that time the wall model only had the Grashof number in it. Now it has the maximum of that and a turbulent force convection, so it kind of has reinforced.

CHAIRMAN BANERJEE: We don't have mixed convection.

MR. KELLY: No. Only by taking the maximum of those two.

CHAIRMAN BANERJEE: I see.

MR. KELLY: But at the time of the data comparison that I'm remembering, I think a model was only a function of the Grashof number, in which case you would expect it to under-predict.

CHAIRMAN BANERJEE: So if you redid it now you'd get a better prediction?

MR. KELLY: Well, not as poor a prediction. I don't know, you know, I'm not going to speculate on how good it would be. I don't know when that model change was. I mean, I think the calculations that are in the ESBWR applicability report were with the final version of the code, but I don't remember.

CHAIRMAN BANERJEE: Are there scenarios in the ESBWR where you might have combined free force convection sort of scenarios? Or is it always going to be pre-convection?

MR. KELLY: I think when it's important it's primarily free. I mean, you are boiling down into these containments, so obviously you, you know, there are periods where you -

DR. WALLIS: Well, the thermal transient in the wall governs after awhile and this doesn't matter at all.

MR. KELLY: That's true. That's true as well.

CHAIRMAN BANERJEE: So Mike, why did you do these experiments?

MEMBER CORRADINI: Why did I?

CHAIRMAN BANERJEE: Yes.

MEMBER CORRADINI: Well, do you want the honest answer or do you want an answer for the record?

CHAIRMAN BANERJEE: I want an answer - well, honest answer you can tell me over dinner, so give us an answer for the record.

MEMBER CORRADINI: Well, these experiments - the experiments that I think Joe is referring to were sponsored by Westinghouse for the AP600.

MR. KELLY: Exactly.

MEMBER CORRADINI: Giving a series of experiments to look at flow over inclined plates both facing downward and up to 90 degrees vertical to try to get an estimate - there was a series of experiments, there were actually four different experimental devices that were built, all focused on AP600 heat transfer from the cooled dome to the steam air atmosphere. And I think the ones he's - I mean, I think the ones you're referring to are the original experiments done 20 years ago by Jim Barry.

MR. KELLY: Yes, the ones on the vertical plate.

MEMBER CORRADINI: Yes.

CHAIRMAN BANERJEE: Okay. So that explains it.

MR. KELLY: This is the summary slide. I have said some of this so many times I'm simply not going to repeat it. But the one thing I want to most note is that everything I've shown has been against single tube or simple wall tests. There has been more extensive ESBWR-specific testing, including multi-tube heat exchangers, for example, the full height PANTHERS facility, and I believe you'll see some of that later today in the closed session. You asked about the liquid to interface heat transfer.

CHAIRMAN BANERJEE: And the wall, if you could show us both. What was that?

MR. KELLY: Okay, this is a section on the descriptions of the wall heat transfer. This is old legacy data. This is a non-dimensional Nusselt number averaged over the heat transfer surface, that's what the brackets are for. The non-dimensional means that the length scale in it is the Nusselt parameter, you know, the velocity squared over g delta rho thing, plotted against the film Reynolds number. I mean, some of this data is people like Kutateladze, okay? This is a simple Nusselt correlation. So it always under-predicts and that's the rippling effect. When this levels out it starts to turn around, and for this data which is FREON data, you notice it starts going back up. This is turbulent film. These are just falling films.

Whole lot of different models. Just look at a couple of them against the data, you know, it's, depending on what, you know, here's Nusselt, the green one is Kutateladze and that's Labuntzov which seems to be a little bit better fit. Then I plotted them against the UCB-Kuhn pure steam test and so that's that data. And you see the models, that's a pretty large uncertainty. This is that data with no correction for interfacial shears. There was some interfacial shear in the test, a fairly small effect, but what I've done now is I've actually calculated the film thickness based on the local conditions and plotted a Nusselt number versus film Reynolds number. When you do that all of these old equations over-predict. That's the UCB fit. So that's what I chose to use.

CHAIRMAN BANERJEE: So what you're seeing is the Reynolds number effect which takes some turbulence into account. Because the Reynolds numbers actually start to go higher.

MR. KELLY: You notice this is 1.

CHAIRMAN BANERJEE: Yes.

MR. KELLY: That's k over delta. As a film Reynolds number comes up you start to increase. And this is a log scale here, so.

CHAIRMAN BANERJEE: Yes, I think what you're seeing is the rippling effect actually starts around 20. You know, if you look at Benjamin's solution for the problem.

MR. KELLY: And that's what these models are supposed to be doing is taking into account the rippling.

CHAIRMAN BANERJEE: They don't.

MR. KELLY: Well, they do against their database, but they don't against the one from Kuhn.

CHAIRMAN BANERJEE: What you've shown there is Kuhn's database.

MR. KELLY: Which I figured was most applicable to PCCS conditions.

DR. WALLIS: What's really surprising is that a lot of the data below 1, so it's worse than k over delta? How can you ever get worse than k over delta?

CHAIRMAN BANERJEE: That shows you the scatter in the experimental data.

MR. KELLY: Yes. There's experimental data and I'm also calculating that film thickness.

DR. WALLIS: You must have calculated delta wrong.

MR. KELLY: That's possible. We're also doing - you know, I didn't want - Mike, how easy are condensation heat transfer tests to do?

MEMBER CORRADINI: I'm sorry, Joe?

MR. KELLY: No, we're talking about experimental error in condensation tests and -

DR. WALLIS: I don't think you have to figure that.

CHAIRMAN BANERJEE: They didn't measure the film thickness directly.

MEMBER CORRADINI: Well, I think - I guess, I don't know what Graham - I apologize that I don't have anything laid up on my screen as you guys are talking, but I think in most of the containment work, as you drive the noncondensable gas fraction to zero, your heat transfer coefficients get so large your error starts really becoming large because most of these facilities were not built to remove that amount of heat. So instead of getting 5, 10, 15 percent error, you're going to get a much larger error in your measurement. And so as I drive it to pure steam, I would not believe many of these experimental data because your error rate, your scatter gets very big.

DR. WALLIS: I think we might sort of establish that once the heat transfer coefficient is above some value you don't really care what it is.

MEMBER CORRADINI: Right. Well, I think that's the practical conclusion Graham, I agree.

CHAIRMAN BANERJEE: Well, but there is actually if you really look at this equation, it is fundamentally wrong up there, and the reason for it is as soon as you start to get some stirring there's a Prandtl number effect that comes in and turbulence, it's clearly Prandtl to the two-thirds that has to be there. So, I mean, it is trying to fit it without a sort of Reynolds analogy is wrong from first principles as soon as it goes turbulent. You have no Prandtl number in that.

MEMBER CORRADINI: But - I think that's a good point. You're absolutely right.

CHAIRMAN BANERJEE: So let's move on anyway.

MR. KELLY: That was for the laminar film turbulent channel. If you look at condensation correlations, now the spread is much worse. So what I chose to do was look at heating correlations, models where you heat a film, a fallen film. Because here you can control the wall heat flux instead of having to try to back calculate it from whatever happened on the secondary side. Here, the models are very, very close together. There's the Wilke Film heating correlation which is in four different parts pieced together, the Gimbutis, and it turns out the Gnielinski correlation which we used throughout TRACE code for forced convection turbulent heat transfer, if you divide it by four you reproduce this. Where does the fourth come from? Think of how you calculate a hydraulic diameter. You know, now I'm using a film thickness as a characteristic link -

CHAIRMAN BANERJEE: This is the gas side.

MR. KELLY: No, this is for the liquid film.

CHAIRMAN BANERJEE: On the interface? Or which part of the liquid film are we talking about? The wall or the interface?

MR. KELLY: Wall to film.

CHAIRMAN BANERJEE: Wall to film. But I thought you were already using -

MR. KELLY: That's for laminar. That only goes up. Remember it only goes up to about 1,000, 1,200, and the Reynolds number and then starts to not look so good.

CHAIRMAN BANERJEE: Okay. Because turbulent liquid films go turbulent - I mean, liquid films go turbulent about 800.

MR. KELLY: Right.

CHAIRMAN BANERJEE: So you're well into the turbulent region. So there's an overlap.

MR. KELLY: Right. So I have the laminar correlation and a turbulent one, and of course there's going to be a power wall combination. This is just showing the comparison data. Now you also asked about the interfacial.

CHAIRMAN BANERJEE: Yes.

MR. KELLY: Well, there's a lot of correlations out, you know, for mass transfer that you can make turn into interfacial ones. You know, any diffusivity models, gas diffusivity models. Again, pretty large scatter. So -

CHAIRMAN BANERJEE: But gas has a Schmidt number of 400.

MR. KELLY: Yes, well you correct it and turn it in. You know, this is turned into a Nusselt number, okay? So I wanted to go look at some experimental data to help me choose one of these. And it's hard to find interfacial heat transfer data. It's hard to measure. So I went to the Northwestern test which I know you know very well, Professor Banerjee. So this is Nusselt number, Prandtl number to the half versus film Reynolds number. There were five measurement stations and you notice there is a - this is horizontal cocurrent flow. There's an entrance effect. Then this is the model, the very first model he quoted where he uses the liquid film as the characteristic link and a turbulent Reynolds number that's really nothing more than a film Reynolds number. I modified that correlation. All I did was change the lead coefficient to correlate the data for these last two stations so that I wouldn't see the larger entrance effect. Then I went and looked at some more of his data. So there were three different series of tests, horizontal cocurrent, horizontal counter-current and vertical counter-current. That's all on here. You notice it's a fairly limited film Reynolds number range from about, you know, 6,000 up to 50. It doesn't hit this transition region. That's where the UCB-Kuhn data is, is down here. So what I do between those two lines, okay?

CHAIRMAN BANERJEE: The Kuhn data is wall heat transfer, right?

MR. KELLY: No.

CHAIRMAN BANERJEE: Some of it is -

MR. KELLY: It's both.

CHAIRMAN BANERJEE: Okay.

MR. KELLY: Okay? If you do the Nusselt solution, you know, where you're assuming it's laminar film flow, conduction across the film, I can't remember which is which.

CHAIRMAN BANERJEE: With the noncondensables it would be -

MR. KELLY: That's - we're just talking pure steam here. But if you look at the thermal resistance inside the film it's a three-eighths, five-eighths, distribution. One is one the wall side and one is interfacial. I don't remember which is which. That's just pure Nusselt.

DR. WALLIS: How do you do a horizontal counter-current fluid test? What's driving the film? It's slightly tilted, is it?

CHAIRMAN BANERJEE: It's slightly tilted, yes. George did these a long time ago. Beautiful experiments.

MR. KELLY: So what I did here was I just simply divided the heat transfer resistance 50/50 between wall liquid and liquid interface.

CHAIRMAN BANERJEE: So this is sort of an arbitrary -

MR. KELLY: It's arbitrary, but you know. And I worried about this laminar turbulent transition region, and if I bring in Gnielinski it looks like that. At least it hits this, whereas if I use the Bankoff I don't. And it manages to hit the Bankoff data where the Bankoff model looked good, so that's -

CHAIRMAN BANERJEE: Gnielinski is for the wall, right?

MR. KELLY: It's for the wall. Now I'm using it, splitting it again. I'm using it for the wall and for the interface.

CHAIRMAN BANERJEE: You're using it for both now.

MR. KELLY: Right. Yes.

CHAIRMAN BANERJEE: Gnielinski, can you show me the form of the correlation again?

MR. KELLY: I don't - did I have it? I don't remember.

CHAIRMAN BANERJEE: I don't think you showed it.

MR. KELLY: Yes, I didn't show it. It's

one of those complicated, you know, friction factor and natural log things.

CHAIRMAN BANERJEE: The worry about all these correlations is that they are for the liquid - vapor-liquid interface any form of Reynolds analogy breaks down which is - and you go into different dependence on Schmidt number and Prandtl number. And where -

MEMBER CORRADINI: Where they break down, Sanjoy, is why?

CHAIRMAN BANERJEE: Because the interface to the liquid is not a solid boundary.

MEMBER CORRADINI: Okay, sorry.

CHAIRMAN BANERJEE: You can actually move parallel to the interface, but not easily normal to the interface.

MEMBER CORRADINI: What you're really saying is the analogy is modified based on the surface structure.

CHAIRMAN BANERJEE: Yes. It's completely - any relationship, for example, will vary as Prandtl number to the half rather than Prandtl to the two-thirds just because of this, or Schmidt number. So if you put in Reynolds analogy which this is trying to do you will get condensation of bubbles, for example, if you use it universally wrong. Or you'll get condensation on a stratified layer wrong, which is mainly turbulent-centered, you know.

DR. WALLIS: So what are you doing here, Joe? I mean, you showed us in the first 30 pages your early work. Now you're showing us that some of the details don't work very well when examined carefully.

MR. KELLY: And this is why the comparisons to the Goodykoontz test. Well, there's possibly the entrainment like you noted, but I think this is why the Goodykoontz tests don't work as well.

DR. WALLIS: For the PCCS you're in the sort of red region there, aren't you? So it works very well.

MR. KELLY: Yes.

DR. WALLIS: How about the other one, the one where you have labels.

MR. KELLY: That's the NASA Goodykoontz. And that's where at the tube -

DR. WALLIS: Then off to the right here and Bankoff would work.

CHAIRMAN BANERJEE: The ICS would work.

DR. WALLIS: The ICS would work.

CHAIRMAN BANERJEE: Well, you know, I think - we're going to have to end it, Joe.

MR. KELLY: Given the uncertainties in condensation heat transfer data anyway, the model is representative of that. It's not going to be better.

CHAIRMAN BANERJEE: So we're running - Joe, thank you very much. This was most illuminating and we'd like to have these slides as well if you would give it to me or whatever so that we have access to them. We'd be very grateful.

MR. KELLY: Do they have the electronic copy right here?

CHAIRMAN BANERJEE: Whatever. I mean, just.

MR. KELLY: It's right here. They're tacked onto the end.

CHAIRMAN BANERJEE: Okay. I think what we need to do now - thank you very much; really appreciate that - is we are running roughly 45 minutes behind schedule. So what we might need to do as Professor Abdel-Khalik will need to leave around 5:00, we might go over a little bit, is perhaps we should bring the TRACE momentum equation discussion which Steve Bajorek will do somewhat earlier to make sure that it covers that before he has to take off. Okay? But right now what I would suggest is that we take a little break for about 10 minutes say, we reassemble at 25 to 11:00 and then we go into Joe's talk, Joe, and we'll try to keep that half an hour, hopefully, if not we'll do whatever we can. And then after that we close the session. So this will still be in open session and I'm just going to go for a break now for 10 minutes.

MEMBER CORRADINI: Sanjoy?

CHAIRMAN BANERJEE: Yes.

MEMBER CORRADINI: I'm going to have to pull away for a couple of hours. I will call you.

(Whereupon, the foregoing matter went off the record at 10:23 a.m. and resumed at 10:37 a.m.)

CHAIRMAN BANERJEE: All right, we are back in session. On the record. We are still in open session and we'll have Joe tell us about the separate effects test now. These are not in color, the slides.

MR. STAUDENMEIER: No, they're not, and unfortunately when I closed down PowerPoint last night I didn't save it so I don't have the electronic version of this here, so we're going to have to go from the paper slides. So fortunately it's straightforward material.

CHAIRMAN BANERJEE: Is this due to a a glitch in Windows or a glitch?

MR. STAUDENMEIER: I think a glitch in the operator. So, for TRACE we've done quite a bit of separate effects assessment that applies to lots of different reactors and we have assessment for a void fraction, heat transfer, critical flow. For ESBWR the two that really matter are voice fraction and critical flow. Heat transfer isn't as big a deal because we never get - recovery. So there is going to be some critical - a little bit of critical flow covered by Don Fletcher later, but I'm going to go over our void fraction assessments, both in rod bundles and pipes since the main parameter we're looking at is predicting minimum level inside the reactor system.

CHAIRMAN BANERJEE: Joe, just to interrupt, we're also interested in the chimney, right?

MR. STAUDENMEIER: Yes, the chimney is a region of interest and that's where we'll end up looking at the minimum levels is - on top of the four, we end up looking at minimum level up in the chimney above the cooler. Move to the third slide. I'm going to run through a sample of void fraction tests that we've done TRACE assessment against. Some of them are in our assessment manual. Two that aren't are FLECHT test for rod bundle and some recent Purdue, marked hydraulic diameter data that we've done through our thermal hydraulic institute testing. Next slide, it should be the FRIGG rod bundle. It's just a picture of the FRIGG facility and a cross-section of the rod bundle. As you know, FRIGG, at least this version of FRIGG was a model of a single bundle in a Marviken reactor that was going to be built. It was a natural circulation BWR. The nice thing about FRIGG is it's at high pressure and full power. The hydraulic diameter in the FRIGG bundle is a little larger than what it is in a typical BWR rod bundle. That's essentially the major distortion between FRIGG and the regular BWR rod bundle, but it does have full power which is - it's hard to get tests with full power.

CHAIRMAN BANERJEE: Remind us about the length here of the bundle.

MR. STAUDENMEIER: Okay. The length, it's as long as a BWR rod bundle would be. So it's, I believe it's about 12 feet. I don't remember the exact dimensions.

MR. KELLY: The next slide shows about 4.5 meters.

MR. STAUDENMEIER: Okay, so yes a little bit longer than that.

CHAIRMAN BANERJEE: Okay, thanks.

MR. STAUDENMEIER: Okay, next slide is -

DR. WALLIS: You don't even have page numbers on the slides so I don't know what the next slide is.

MR. STAUDENMEIER: FRIGG rod bundle plot. It shows an axial void profile for one test calculated in an experimental -

DR. WALLIS: So you do better with this complicated geometry than a university does with a straight button?

MR. STAUDENMEIER: I don't know what a university does with a straight button or what you're referring to.

DR. WALLIS: I'm just saying that two-phase flow void fraction, even in a very simple geometry does not get - very well. You seem to do very well with this rather complicated geometry.

MR. STAUDENMEIER: We have really good predictions in rod bundle -

DR. WALLIS: You have to - you have to attune things in the code to that?

MR. STAUDENMEIER: We use the Bestion - simple Bestion correlation -

DR. WALLIS: Which is for -

MR. STAUDENMEIER: - equal to 1 is our rod bundle void correlation.

DR. WALLIS: So it is based on rod bundle data.

MR. STAUDENMEIER: It's based on rod bundle data.

DR. WALLIS: That's why it works.

MR. STAUDENMEIER: But this hydraulic diameter for FRIGG is much larger than the database that SDL was based on, so it is not in the same range of conditions that the data was based on, or the correlation.

MR. KELLY: And this is Joe Kelly. Remember, you're looking at the bundle average void fraction. If you were to look at the void fraction in any one subchannel or say the subchannel up against the wall which is the subchannel in the center, TRACE isn't going to calculate that difference for you.

CHAIRMAN BANERJEE: Remind me of the Dominique correlation. Is this sort of a drift flux sort of correlation, or what is it?

MR. STAUDENMEIER: Yes, it's a drift flux correlation.

CHAIRMAN BANERJEE: And you back out the interfacial friction from that then?

MR. STAUDENMEIER: Yes.

CHAIRMAN BANERJEE: So this is different

from that guy in EPRI. And that also is a drift flux type correlation.

MR. KELLY: And like in the RELAP 5 code they do the same kind of thing backing that out into an interfacial drag correlation. It's a very complicated correlation, it has switches in it so it has discontinuities, and we found we could get the same or better accuracy with the simple model so we went with the simple model. We actually compared both models against the data set before we chose one.

CHAIRMAN BANERJEE: Thank you. All right. Let's keep on going.

MR. STAUDENMEIER: Okay. The next slide is just predicted versus experimental void fraction for essentially the whole test series of FRIGG tests that had been measured void fraction. And that goes over a wide range of subcoolings.

CHAIRMAN BANERJEE: Do you also - I mean, in these correlations, is it primarily the interfacial shear that is adjusted, or do you also adjust the wall friction in some way? Or do you keep the wall friction and back out the interfacial shear from the drift flux correlation?

MR. KELLY: Do you want me to answer?

CHAIRMAN BANERJEE: Either.

MR. STAUDENMEIER: Yes, I mean this - the correlation doesn't depend on the wall friction, and we assess over a wide range of flow rates. In something like the FRIGG rod bundle, or BWR full flow conditions, the wall friction can give a significant pressure drop. In more stagnant conditions like the FLECHT low pressure thing where you're sitting under decay heat, wall friction isn't so big. So the correlation wasn't developed with a wall friction component in it, but we have assessed over a range of conditions with different impacts other than the wall friction.

MR. KELLY: This is Joe Kelly again. What Joe is saying is completely correct. When you start, however, getting to the higher void fractions, 0.8 or so, you're going into the annular flow regime and here the wall friction is very important, and here we're actually using the - excuse me, the Wallis interfacial drag model with the two-phase flow wall drag model. And it's the - how those two interact that gives you the void fraction at these highest factors.

CHAIRMAN BANERJEE: So -

MR. KELLY: So wall drag is being computed in all of this. In some places it's important, in others it isn't. If I were to develop a drift flux model from scratch, I would take the data, subtract out the effect of the wall drag, then correlate the interfacial drag. That was not done in the development of the models that we chose to use, but what Joe is showing you is that when you put that interfacial drag model together with the wall drag model, you get this.

MR. STAUDENMEIER: Okay, the next data I'm going to show assessment against was the THTF rod bundle, the level swell test. This is also a high-powered bundle, and it's a medium range pressure condition, I think about four megapascals. You can get all the details of these assessments in the Appendix B of our assessment manual. I guess the main mission of this facility was also blow-down cooling for PWRs under large break LOCA but the level swell tests were done for PWR small break LOCA conditions.

Next slide, it shows a plot. It's giving an example from one test showing comparison of void fraction predictions with both the channel and the vessel component in TRACE versus experimental data. This is fairly representative of our predictions. We get good predictions. Next slide shows a comparison of some kind of global parameters that are calculated from the test. The left plot is experimental collapse level, and you can see that we're predicting that fairly well. The right slide is mixture level, two-phase mixture level, and TRACE is either predicting it very well or under-predicting it which would be conservative.

CHAIRMAN BANERJEE: What do you mean by Chan model and vessel model?

MR. STAUDENMEIER: Well, there's two places where we can have rod bundle interfacial drag. We can have it in our vessel component and our BWR channel model. Actually, since this was done we can also have it in a height component. You have to designate a region where you're using the rod bundle interfacial drag, and these are - when we were testing this we wanted to make sure that both the vessel model and the Chan model were both giving predictions as they should be since -

DR. WALLIS: Is the chimney a vessel or a pipe?

MR. STAUDENMEIER: - in our ESBWR input deck is modeled in the vessel component, so but we've also done testing with vessel and pipes and get the same void fractions in those in non-rod bundle regions also.

CHAIRMAN BANERJEE: Is there going to be

some discussion of critical flow as a separate effects thing, or are you just focusing on void fractions?

MR. STAUDENMEIER: I'm just focusing on void fraction. I think Don Fletcher goes into a little bit about critical flow predictions. Actually, I have a little bit that's related to critical flow later in the GE level swell test which is partly dependant on vapor critical flow.

CHAIRMAN BANERJEE: And are you going to also discuss the void fraction for Ontario Hydro data?

MR. STAUDENMEIER: Not in this presentation I'm not. Don Fletcher has that one in his presentation.

MEMBER ABDEL-KHALIK: Why is that? Why doesn't it belong here?

MR. STAUDENMEIER: Well, I'd rather not talk about why I don't think it belongs here since it's open session.

CHAIRMAN BANERJEE: But you'll come back to it in the closed session?

MR. STAUDENMEIER: Yes.

CHAIRMAN BANERJEE: Okay. The data - are these proprietary data, the Hydro data?

MR. STAUDENMEIER: They actually were presented in an open conference at one time and there is a conference paper that has it.

MEMBER ABDEL-KHALIK: So if that's the case, why aren't they being included in this open discussion?

MS. UHLE: Because if there's anything specific to ESBWR -

MEMBER ABDEL-KHALIK: There's nothing specific in this comparison.

MS. UHLE: - a particular test to that, that could create an uncomfortable situation.

CHAIRMAN BANERJEE: Let's defer it to the closed session. That's fine.

MR. STAUDENMEIER: We didn't discover the data.

CHAIRMAN BANERJEE: Fine, okay. That's fine.

MR. STAUDENMEIER: RBHT rod bundle tests. Left plot is an example of one prediction of void fraction prediction and measurements. The right thing is a summary of all TRACE predictions versus -

DR. WALLIS: These are local void fractions, are they?

MR. STAUDENMEIER: Yes, they're local void fractions.

MR. KELLY: These are going down to three inches and there's also grid spacer effects in there.

MR. STAUDENMEIER: And some of the data is noisy. They had trouble controlling pressure on some of the experiments also, and nobody has really gone through and sorted out which are the ones you really need to keep and which ones maybe should be discarded since it's drawing all of them together, so.

MEMBER ABDEL-KHALIK: I guess I have a philosophical question which is in some sense related to the comparison with the Ontario Hydro data. What do you do when you compare the code against two sets of data and in one case it says excellent comparison, and in the other case it says inadequate comparison?

MR. STAUDENMEIER: Well, I guess you try to -

MEMBER ABDEL-KHALIK: For the same physical phenomenon and the same range of parameters, presumably.

MR. STAUDENMEIER: I guess you try to see, well, is - are both sets of test data valid, and if they are both valid then you have to look into the reasons, or start looking into the reasons why you're not predicting one set of data very well. What is it about that data that may be slightly different, or there may be something in those conditions that you didn't consider in the correlations you had built.

MEMBER ABDEL-KHALIK: Is that part of the presentation that we will hear later on today? Have you faced that situation in any of these comparisons at all?

MR. STAUDENMEIER: I don't think we've faced a situation where we think that we're just not getting good answers at all. Well, I'll show you in a couple of slides later that large hydraulic diameters, we tend to under-predict void fractions up in the large void fraction range, and that's something we're investigating in the code now is to why we're doing that. And we actually went out - that was one reason why we went out and got this Purdue large hydraulic diameter data to try and look at developing a better correlation that applies to high void fractions and large hydraulic diameters.

MEMBER ABDEL-KHALIK: So in all the comparisons that you've made in the experimental data, all of these comparisons had found the data from various sources to be consistent? Is that what you're telling me? Vis-a-vis the code predictions?

MR. BAJOREK: No, I don't think we have found any data to be dichotomous, in that one set is predicting one thing and another one is predicting something else. We do see quite a bit, like on the RBHT where there's a lot of scatter in the data. Hopefully in the long run when we develop an uncertain methodology we would be looking at some of those correlations be it drag or heat transfer, and ranging that over that entire set of data to get all of the bad actors, as you will, regardless of whether the code is better for one or the other.

MEMBER ABDEL-KHALIK: Well, I have two sets of data and that's why I was wondering why you explicitly excluded the Ontario Hydro test results from this open presentation. Because in the comparison here it says the calculated ESBWR Hydro chimney level are judged to be minimal when compared against that set of data. And then when you compare the data against Wilson bubble rods data which you elect to show us, it says the TRACE capabilities for predicting the collapse ESBWR RPD chimney level are judged to be excellent. So I'm just wondering if you're being intellectually honest in this presentation.

MR. STAUDENMEIER: Well, I think there is a fairly good comparison versus the Wilson data and I won't talk about the other data now, but you'll see hints at the Wilson data high void fractions that we are starting to under-predict, and I think the Wilson data void fractions don't go quite as high.

CHAIRMAN BANERJEE: I think what we can say so we can have a free discussion of this, let's table this till we go into closed session and then I think your point is very well taken, because if you look at the Purdue PhD data, it over-predicts that.

MR. STAUDENMEIER: Probably, yes.

CHAIRMAN BANERJEE: So we can discuss what is happening there. I think, let's do it at that point.

MEMBER ABDEL-KHALIK: Sure. Thank you.

MS. UHLE: Can I just advocate something here? I know we're not in the habit of lying to the ACRS or -

MEMBER ABDEL-KHALIK: I'm not suggesting that. I, you know, if you're going to have an open discussion then you should present the whole story in the open discussion, not part of the story.

MR. STAUDENMEIER: Actually, we wanted to close the whole meeting and we were forced to open part of it and so we kept - that was one reason why I kept that out of this presentation, because they wanted as much open as they could.

CHAIRMAN BANERJEE: Right. We understand

that.

MR. LANDRY: This is Ralph Landry from the New Reactors office. The data that are being referred to as Ontario, while it's been reported in an open paper many years ago, the data were provided to us through proprietary documentation from the applicant. Therefore we are treating the material as proprietary so that we can't discuss that in open session.

MEMBER ABDEL-KHALIK: I think we'll wait till that closed session then and discuss.

CHAIRMAN BANERJEE: I think we'll just -

MEMBER ABDEL-KHALIK: I just want to make sure that whatever story you're presenting is not selective.

MR. LANDRY: No, the Office of Research is trying to be honest with the applicant, that the applicant has provided the material under proprietary documentation, so they're trying to be honest and not present it in open session, even though they are going to present it later.

CHAIRMAN BANERJEE: I think it's fine, thanks. Let's move on. Go ahead.

MR. STAUDENMEIER: Okay.

CHAIRMAN BANERJEE: Which slide now? You have to tell us.

MR. STAUDENMEIER: Next slide is the FLECHT rod bundle slide.

CHAIRMAN BANERJEE: Now, we have gone over the RGHT data, right?

MR. STAUDENMEIER: Right.

CHAIRMAN BANERJEE: There is a - I mean, the RGHT data though, there's quite a bit of data that seems to lie below the predictions. Is that sort of correct, or am I seeing it wrong here? If you look at the void fraction -

MR. STAUDENMEIER: Well, we over-predict - I think we over-predict void fractions more than we under-predict it.

CHAIRMAN BANERJEE: Yes, well, I say a lot of the data lies below your prediction.

MR. STAUDENMEIER: Right. And there's a lot of scatter in this data.

CHAIRMAN BANERJEE: Is it the scatter is coming from the experiment, or is it coming from -

MR. KELLY: Almost exclusively. There are very small DP cells over about a 4-foot length of the bundle, every three inches, and they span grid spacers. They're just upstream, just downstream of grid spacers. You go from one DP measurement and for the void fraction, go to the very next one and you see these 10, 20, 30 percent void fraction changes one DP set to the other. So most of this is experimental scatter. And as Joe said, we've never sat down and done a systematic qualification of the data as to what should be used in this kind of comparison.

CHAIRMAN BANERJEE: So you're still using Dominique Bestion's work? Everything is consistent, you haven't adjusted anything?

MR. KELLY: No. What he's doing now is showing you the model for void fraction prediction rod bundles. And he's showing it for higher pressure, for power BWR conditions, transitioning to high pressure low flow DKE, that was the PHDF, that's like a small break LOCA kind, and the RBHT and FLECHT are low pressure EKE so now the ADS flow down is complete and you're just sitting there whole boiling kind of thing the rod bundle. So he's covering the whole span, the trajectory of the transient that you can postulate.

CHAIRMAN BANERJEE: So this is at low pressure now.

MR. KELLY: Right.

MR. STAUDENMEIER: I think if you look at the left plot on the RBHT slide you'll see that's void fraction measurements, across is from one test going up to bundle and you can see just the scatter in that under supposedly steady conditions.

MR. BAJOREK: If you follow those points you'll see the ones where you start to get a lot of decreases in the voids. They're right at the grid spacers.

CHAIRMAN BANERJEE: Yes, I see some dark to the extent I can tell from this black and white flow up there, some thicker crosses and thinner crosses. Do they have any significance, the difference?

MR. STAUDENMEIER: No.

MR. KELLY: That's two crosses on top of each other.

MR. STAUDENMEIER: That's just resolution from the screen grab that I did to paste it in here.

CHAIRMAN BANERJEE: All right. Fine.

MR. STAUDENMEIER: Okay. Next is FLECHT rod bundle. It's not in our assessment manual, but we've done this assessment in looking at selection of Bestion correlations. This shows two different pressures, 20 and 40 PSIs. This is under decay heat conditions. And you see we get good void fraction predictions for both of those.

CHAIRMAN BANERJEE: But these FLECHT data, what sort of rod bundle remind me these were?

MR. KELLY: This is a PWR 17x17 geometry. These were reflood tests, but this is after the reflood is complete and you're sitting there for awhile just simply boiling at low flow rate, low pressure rod bundle.

CHAIRMAN BANERJEE: Okay, go ahead.

MR. STAUDENMEIER: Okay. Next, looking at some pipe data, larger graph broad diameter pipe data versus Wilson bubble rods experiment. 0.46 meter diameter test section, a range of pressures that the data was taken over. Can see, the next slide shows a summary of TRACE predictions versus measurements.

CHAIRMAN BANERJEE: So these were basically sort of experiments where the level rose and then broke - it broke through and you're mainly looking at the relatively low void fraction range with these?

MR. STAUDENMEIER: No, it goes up to fairly high void fractions, but yes, it's essentially putting steam up through - a cone of water sitting there which wouldn't apply to ESBWR under steady state conditions, but after you've broken natural circulation in sitting there, this is kind of the situation we have.

CHAIRMAN BANERJEE: So, you have either

very little water flow or no water flow, right? In this case.

MR. STAUDENMEIER: Yes.

CHAIRMAN BANERJEE: Okay.

MR. STAUDENMEIER: If you look at the slide showing the summary of the test predictions, you see we have reasonable predictions until you get up towards the high void fractions up about 0.7 and above and you see - starts to seem systematically under-predict the data.

MR. KELLY: It's like two data points the void fraction is greater than 70 percent. And so you go is it the code, is it the experiment - because you're using a delta p cell in this column of water, and there's two data points. And there was one other, Allis-Chalmers, so we're going way back in history. Same thing. Right in that point it looks like there might be a trend away from it. You know, we saw this and we're going is it real? We don't know. So what we did was institute a test program at the Thermal Hydraulic Institute at Purdue and that's what Joe is going to show you in just a minute, to investigate just that area.

CHAIRMAN BANERJEE: Now, in this case, you're still using Bestion's -

MR. STAUDENMEIER: No, this is a light drift flux model. This is a Kataoka-Ishii model.

MR. KELLY: And this is the model that will be used in the chimney region of the ESBWR. That's what you're assessing.

CHAIRMAN BANERJEE: So you're out of the rod bundle region. And you're using a different correlation.

MR. KELLY: And it's for assessment of the chimney region is the reason he's looking at large diameter pipes. We're not looking at 1-inch diameter pipes anymore. We're looking at things as large as we can find. And there's not that much data out there.

CHAIRMAN BANERJEE: Okay.

DR. WALLIS: Is this a correlation they use is drift flux in the C-0 distribution?

MR. KELLY: Yes.

DR. WALLIS: Because if you have a 1.2c you have difficulty getting high void fractions. It cuts everything off.

MR. KELLY: Bingo.

DR. WALLIS: You can't get in that region up there.

MR. STAUDENMEIER: And that's what we're probably going to be looking at in developing correlations -

DR. WALLIS: So you shouldn't use it for high void fractions.

MR. STAUDENMEIER: I mean, if you look at the dimension-less hydraulic diameter in these things that goes into Kataoka-Ishii, when you get up into ESBWR chimney conditions you're up in the range of 500 to 600 I think. Our Purdue test data doesn't go up that high, but we're probably going to be looking at some way, a dimension-less hydraulic diameter tapering off that CNOP from 1.2 down to 1 as you go up into large hydraulic, dimension-less hydraulic diameter.

CHAIRMAN BANERJEE: There's one thing that we should also look at and we've been concerned about as a committee, which is the - when you're in this region of turbulent flows and things, you get a sort of chugging phenomenon. If you ever look at these systems, you see this and what you see is the time constant of these typically is of the order of two seconds. Now, the concern has been with the committee whether you can actually get significant fluctuations in hydraulic head in this because after all, this is a buoyancy-driven system and would these feedback into the sort of time scales associated with the regional instabilities in the core. So if you look at that,

that's of the order of a couple of seconds. So, the coupling is of some interest, or potential coupling. So when you do these experiments, it would be very interesting to know what the hydraulic head fluctuations are like for a large pipe. Now, the data that shall remain nameless right now, they had a couple of gamma densitometers there and if you actually look at it, there are significant fluctuations which are correlated between the two, and the part had an instability. So, but that had a completely different time constant. So it's just interesting to measure these if you can.

MR. KELLY: My experience on the time constant is it's quite often related to the height of the facility. Kind of the slug hits the top, comes back down and reforms. In the Purdue test which he's going to show you some of, the experimental measurements were in - it's inductance for the void fraction and it's always a continuos signal. So you see those traces, variations in time, and they put that signal for a neural net to identify which kind of regime they're in. So they look at frequency response. That's data that they can bring back to you another time so you can look at that.

CHAIRMAN BANERJEE: So this is - the

cross-flow in -

MR. KELLY: It's a large pipe, so it's just two, you know, it's an acrylic pipe, but there are sections where you have measurements, and there it's metal. Two metals and it's, you know, I don't remember the details.

CHAIRMAN BANERJEE: So it's not the Dresden-type -

MR. KELLY: No. You don't get the local information here. But you get the cross-sectional average void fraction from it. But you also get the time signal of that void fraction which you can then look at. We can provide that to you as well as the time signal on the pressure traces. And in the film we got at the Purdue test, okay, they're in water, they're not high pressure steam, so they're low pressure, so that's not correct. But, they cover the high - they go up to the high void fraction range and they cover everything from zero liquid velocity where they look like the Wilson bubble rise test to I believe it's 2 meters a second. So that's mass flux at 2,000 kilograms meter squared.

CHAIRMAN BANERJEE: Is that comparable to -

MR. KELLY: That's BWR operating

conditions. I think it's actually -

CHAIRMAN BANERJEE: And the diameter, is it comparable?

MR. KELLY: Well, I'm trying to - the diameters are not comparable. I mean, we're in the chimney region. Something like this almost a meter across and we can't build a facility that large. So what we did, the testing was done I believe in a 6-inch and a 10-inch pipe. And so you're getting big enough that you no longer can have slugs anymore, but we did it at two different diameters so we could see. You know, one of the problems with going to larger diameters is you can't put that much air through it anymore. The cross-sectional area goes as a square.

CHAIRMAN BANERJEE: You're aware of course that there are major facilities in the world which - in the oil and gas industry, which have as large or larger diameters and can put through the appropriate mass fluxes, and have incredible instrumentation.

MR. KELLY: Actually, I'm not aware of, but that would be -

CHAIRMAN BANERJEE: Yes. SINTEF has it. If you ever go, they have 100-meter long and 50 meters high or something. This is a different ball game.

MR. KELLY: SINTEF?

CHAIRMAN BANERJEE: Yes.

MR. KELLY: T-E-F?

CHAIRMAN BANERJEE: In Norway. Their facilities, their incredible facility is available to do this stuff, except of course the oil industry can afford it, not the nuclear. But anyway, leaving that aside, the reason I'm saying you get this. You see it in the oil business that when you go into these risers, the slug catchers you design have to be, you know, they get very long slugs coming through in exactly this regime, which give you big fluctuations in hydraulic head. Now, whether in GE's case you'll get this is hard to know, but that's what - the question we are asking.

MR. KELLY: When you - when you get - I've looked at a lot of like air-water experiments in different pipes. When you start getting to the larger pipe and you go to these kind of void fractions, and the chimney region if I remember is designed to operate 60, 70, 75 percent void, something like that.

CHAIRMAN BANERJEE: Turbulent, exactly, the wrong region.

MR. KELLY: The wrong region, except you're not going to have intact slugs anymore. And if you - because of the void fraction. And when you talk about having the liquid film bridging.

CHAIRMAN BANERJEE: It's not slugs in that sense. What you get are - actual void weight.

MR. KELLY: Okay, that -

CHAIRMAN BANERJEE: You see, what you get in these, it's very well known, at least my impression is, that you get an agglomeration of liquid into large regions, and you get regions of low liquid. Now, GE has multiple sources coming in, so it's not an oil-gas pipeline. You know, you've got many channels feeding these. So you've got a relatively different dispersed flow at the inlet. So whether these void waves can develop within the length of the chimney which is a relatively short length compared to a pipeline, I don't know. So it's like an entrance region problem. But it's a significant problem for pipelines. So I think we need to take cognizance of this and TRACE's capability to capture these void waves. They're not density waves in the sense that density waves - these are flow regime waves. They're not density waves.

MR. KELLY: Of course, the constitutive models in TRACE were designed to provide the correct time average response, not - you know, they're averaged over volume and time. So they're not to give you the kind of local density waves you're talking about. Ones that come from a system effect, like a condensation-induced instability. Those kind of things we should have a chance of getting, but not what you're talking about.

CHAIRMAN BANERJEE: Well, the multi-field model is known to be able, with high enough resolution and low enough numerical diffusion to be able to capture some of these effects. Because there - well, Jeffrey won't agree to this, but there are many aspects. But some -

MR. KELLY: The first order.

CHAIRMAN BANERJEE: Yes, the first order, you can get them. So, if you ran it in an explicit mode with low numerical diffusion with a high resolution, you probably will see some of this stuff at some point. Anyway, that's a separate discussion. All we're saying is not only average voids, but void waves could be of interest in this problem from an ESBWR point of view. And that question has come up several times. I think when Graham was chairman of the ACRS he raised that question and it's never really been adequately answered I think up till now. Whether there's some coupling between the chimney and the core instability.

MR. KELLY: You know, I don't think the

data that we have can answer that, but we can show you some of the data that we have, not here today, but another time. There is the pipe test we talked about, air water, but also the PUMA facility. It does model a chimney region sitting over the top of a pool. Now again it's not exactly prototypic, it's shorter, but they have pressure taps and they have optical void fraction probes inside the chimney region. So you can look at a temporal trace of void fraction and see what the oscillations are. I don't remember what the -

CHAIRMAN BANERJEE: Were they at full pressure in these?

MR. KELLY: Of course not.

CHAIRMAN BANERJEE: What are we talking about really is a normal operations issue. As you drift, you know, into regions where we are more likely to have instabilities. But let's go around that now.

DR. WALLIS: I think we should move on. I think that this business of building up these regions, concentrations of bubbles in this sort of geometry involves the attraction between the bubbles which isn't in the model at all. In fact, so the wake of one bubble pulls in other ones, particularly if they're big ones, and that's not I think in TRACE at all. So I think we should move on.

MR. STAUDENMEIER: Okay. As I said, you can see at the upper end of the Wilson data that TRACE is starting to it looks like under-predict the void fraction. You can see there was kind of a curvature to the predicted values. It looks like it's starting to roll over up near the high void fractions.

Next slide, GE level swell experiment, large hydraulic diameter pipe void fraction under transient conditions. Also has some steam critical flow in it. You have a large vessel, open up a hole in it, look at the level swell in the tank and as it boils down. Next slide. You can see there were a couple runs that were done. TRACE best was adjusting the discharged coefficient at the nozzle to try to get a better estimate of the depressurization. Base run was just using discharged coefficient of 1.

CHAIRMAN BANERJEE: So the nozzle experiments, you didn't adjust the discharge coefficient, is that it?

MR. STAUDENMEIER: We did. The plots that are going to be shown are with the adjusted discharge coefficient. We try to match deeper, get a better estimate of the depressurization rate. So it'll be - the void fraction thing will be using the best estimate for discharge coefficient. First plot is up near the top of the tank. You can see it gets fairly good prediction of transient void fraction versus firm. Next plot is further down in the tank, so also down there. It's also getting a decent prediction of the level swell.

CHAIRMAN BANERJEE: What are those bars there?

MR. STAUDENMEIER: Bars are uncertainty in the test data.

CHAIRMAN BANERJEE: And if you ran TRACE with the base model, did you get this particularly different, or?

MR. STAUDENMEIER: Well, it wouldn't be - I mean, you'd get the right shape, but you're depressurizing at too fast a rate, so you'd get more level swell in the calculation compared to the test data.

CHAIRMAN BANERJEE: So basically what you're saying is that if you enclose some sort of correct pressure transient you get more or less the right?

MR. STAUDENMEIER: Right. And like in something like ESBWR they'll have specifications on nozzles like SRVs or DTV valves saying that you're going to get this flow rate and at this pressure, and that's part of essentially the design basis of the plant. Their exiting calculations are done assuming you're going to get this flow rate at this pressure. So in a plant calculation you're really - you don't care what the actual area is, you put an area that's nominal, put a discharge coefficient on that's going to give you that flow rate at that pressure.

MEMBER ABDEL-KHALIK: You didn't have that information for this experiment?

MR. STAUDENMEIER: There was - it wasn't good characterization of the discharge coefficient in that documentation we had for the experiment.

MEMBER ABDEL-KHALIK: I'm just trying to - if that is the case, then what do we learn from these comparisons?

MR. STAUDENMEIER: Well, if you're looking at what happens inside the vessel at the right depressurization rate, then you're looking at seeing what the level swell is at that depressurization rate which is prototypical of what happens in a BWR. As you know, the BWR safety systems, you're draining down, hit a level trip, then you start the automatic depressurization system which depressurizes the system at some rate which is close to what this test is showing.

MEMBER ABDEL-KHALIK: I'm just trying to understand. You sort of tweak the loss coefficient.

MR. STAUDENMEIER: A discharge coefficient, not a loss coefficient.

MEMBER ABDEL-KHALIK: A discharge coefficient, excuse me. And you were able to match the pressure history and therefore were able to match the level swell.

MR. STAUDENMEIER: Right.

MEMBER ABDEL-KHALIK: Then we move on, get another experiment. You tweak a discharge coefficient?

MR. STAUDENMEIER: No. Once you characterize a discharge coefficient for a facility -

MEMBER ABDEL-KHALIK: So you're telling me that this sort of process of tweaking the discharge coefficient happened only in this set of data because you didn't have that information?

MR. STAUDENMEIER: That's right. In pretty much any test or any integral test we have, hopefully the experimenters characterized the discharge coefficient and will give it to you and say if you use this formula for steam critical flow and this discharge coefficient with this area, this will give you the right flow rate like something like ETSI tests with steam generator blow-downs. I know this information is given in other test facilities. You characterize discharge coefficients or data that you can come up with a discharge coefficient for your code, but all nozzles are different. I mean really, you need to know what the hardware is.

MEMBER ABDEL-KHALIK: Okay.

CHAIRMAN BANERJEE: What was - this is still Ishii-Kataoka for the interfacial drag?

MR. STAUDENMEIER: Yes.

CHAIRMAN BANERJEE: Let's go on.

MR. STAUDENMEIER: Okay. The next thing is going over some of the Purdue large hydraulic diameter test data. One of our junior staff members Andrew Ireland has started doing TRACE assessment of it recently. All we have is assessment against the 6-inch pipe at this time. As Joe said, there are air-water tests, large hydraulic diameter, void fractions up to about 80 percent.

CHAIRMAN BANERJEE: Are these similar to the Wilson bubble rods? Or you also have liquid flows, right?

MR. STAUDENMEIER: Yes.

CHAIRMAN BANERJEE: But Wilson bubble rods were steam water and went up to high pressure?

MR. STAUDENMEIER: Right.

CHAIRMAN BANERJEE: Okay.

MR. STAUDENMEIER: Very high pressure. These, I mean, there's a high pressure and a low pressure slide, but both of them are relatively low pressures. The low pressure data was done at 180 kilopascals. You can see there were measurements taken at different axial locations along the pipe. You can see generally TRACE is over-predicting this set of test data, although we don't go up to - you can look at the maximum void fraction and it's up around 0.7. One thing I'll also say is in some of these TRACE assessments up at high elevations you'll get oscillations in the predicted void fraction in the TRACE calculations. We want to look at time traces for the facility to see if there's oscillations in the data. We haven't done that yet. And see if it's - although we don't really have models built in for transient flow regime changes like that, we want to see at least are our oscillations consistent with the facility, or is that just another sort of bug in the code that we have to deal with to get the oscillation.

Next slide is the high pressure data which is at 280 kilopascals. You can see, again, TRACE is generally over-predicting but you can kind of see that effect in there. If you drew the best line through the data you can see the predictions are starting to roll over a little bit at the high void fractions again, where the higher void fraction you get, the less it's over-predicting. Summary of the assessments. We've done a lot, an extensive amount of assessment performed on TRACE and we have reasonable agreement with data that we think is important to ESBWR calculations.

CHAIRMAN BANERJEE: Any comments from subcommittee members on this? Or should we defer them to the closed session?

DR. WALLIS: I would like to get on to the ESBWR.

MR. STAUDENMEIER: One more comment before the session closes. This has to do with some things that came up in the first presentation. Has there been any assessment done for containment? We do have some integral tests for - related to containment predictions in ESBWR, both the PUMA and PANDA predictions. And one comment was made about some of the heat transfer correlations in containments providing for conservative pressure predictions for heat pressure calculations. In ECCS the worst pressure is low pressure for ECCS calculations, both for large-break LOCA and for BWRs. And we have done some calculations where we've looked at making the pressure artificially low in the containment for ESBWR. You'll get to see a calculation where that was done where the suppression pool was kept at atmospheric pressure during the calculations, so it has to boil down further and that - both for BWRs and PWRs it's - large-break LOCA, it's conservative to have low pressures in containment for the in-vessel ECCS response.

CHAIRMAN BANERJEE: Are we going to - I mean, talk about what the staff actually are using for calculations? Because I know that this is related to TRACE applicability, but the hearsay is that the MELCOR is being used for calculating the containment. Is that true?

MR. STAUDENMEIER: Well, I don't think it's hearsay. I think it's actually been presented before the ACRS already, so. But for long-term containment cooling calculations and peak pressure calculations, MELCOR is the code that the staff is using for confirmatory calculations. We have assessment in that period for TRACE for some long-term cooling, but I mean, the things that are going on in those experiments aren't really what's driving the long-term pressure in ESBWR in these calculations. I think it's the addition of the electrolysis noncondensable - or radiolysis noncondensable gas. You keep that and noncondensable gas to a closed system, the pressure's going to keep rising. Our test assessment cases and the test data don't cover that situation where you're constantly adding noncondensable gas to the system over three days.

CHAIRMAN BANERJEE: Now, TRACE has the capability to do this long-term containment pressure calculations, it seems.

MR. STAUDENMEIER: It does.

CHAIRMAN BANERJEE: So I guess this is to be presented to NRO at some point, but why is NRO using MELCOR rather than TRACE?

MR. LANDRY: We can do that at some point when we are up supposed to be talking about this BWR. and the Chapter 21 which I think are in August? We can talk about that a little bit then.

MR. BAJOREK: Ralph, I think part of it though was the assessment and the run time. The higher pressures with ESBWR are going to mean hours, days in the transient, and MELCOR is better suited to look at those very long-running transients. TRACE could get there, but it would take an exceedingly large amount of CP time.

MR. LANDRY: But this is involving our containment branch. I think I'd prefer to make sure that we have the right people here to address any questions that come up.

CHAIRMAN BANERJEE: Well, one of the things that we understand is that GE is using TRACG to do other things.

MR. LANDRY: Right.

CHAIRMAN BANERJEE: So. Slightly different approach. Anyway, let's continue with -thanks Joe, and I think we should probably go into closed session now.

(Whereupon, the foregoing concluded at 11:28 a.m.)

TRACE

TRACE Condensation Model Development for the ESBWR

Presented to the ACRS Subcommittee on Thermal-Hydraulic Phenomena

by

Joseph M. Kelly

Feb. 27, 2009

TRACE Condensation Model Development for ESBWR

- **CONTENTS**
  - Introduction
    - Background
    - Modeling Approach
    - Model Accuracy
  - New Model Description
    - Wall Friction
    - Interfacial Shear
    - Wall-Fluid Heat Transfer
    - Interfacial Heat Transfer
    - Non-Condensible Gas Effect
  - Sample of TRACE Assessment Results
  - Summary

TRACE

Background

- **Proposed ESBWR design:**
  - Tube Condensation
    - Passive Containment Cooling System (PCCS)
      - Condensation in the presence of non-condensible gases.
    - Isolation Condenser System (ICS)
      - Highly sheared turbulent film condensation.
  - Wall Condensation
    - Containment volumes such as the dry well.
      - Falling films with non-condensible gas effect.

- **A model review and assessment was performed:**
  - Significant deficiencies identified in both the modeling approach and predictive capability of legacy TRACE model.

TRACE

Introduction
Model Development Effort

Objective

- Implement a model in TRACE for
  - In-tube condensation that is applicable to the ICS and PCCS systems,
  - Modify tube model for wall condensation in large containment volumes.

Approach

- Model should be compatible with two-fluid numerical framework.
- Model should take advantage of quantities computed by TRACE through the solution of the conservation equations:
  - (e.g.) axial distribution of the condensate flow rate and film thickness
    - then, the Nusselt formula becomes:

$$h = k_l / \delta$$

    - instead of:

$$h = \left[\frac{\rho_l \cdot g \cdot \Delta\rho \cdot h_{fg} \cdot k_l^3}{4 \cdot \mu_l \cdot z \cdot \left(T_{sat} - T_w \right)} \right]^{1/4}$$

TRACE

Introduction
Model Accuracy

Condensation Data Base

Experiment	Tube Diameter (mm)	Pressure (MPa)	Gas Reynolds No.	Film Reynolds No.	NC Mass Fraction (%)
Pure Steam Condensation Tests					
UCB-Kuhn	47.5	0.11 - 0.52	4000 - 34,800	43 - 2000	-
NASA	7.44	0.02 - 0.26	3300 - 237,000	83 - 8400	-
Air-Steam Condensation Tests					
UCB-Kuhn	47.5	0.11 - 0.52	3300 - 46,900	36 - 1800	1 - 56
MIT-Siddique	46	0.11 - 0.49	310 - 23,400	40 - 700	8 - 87
Helium-Steam Condensation Tests					
UCB-Kuhn	47.5	0.39 - 0.43	3100 - 31,200	50 - 1400	0.3 - 26
MIT-Siddique	46	0.11 - 0.47	650 - 9300	73 - 450	2.8 - 46
MIT-Hasanein	46	0.12 - 0.60	1100 - 21,500	50 - 650	2.5 - 66

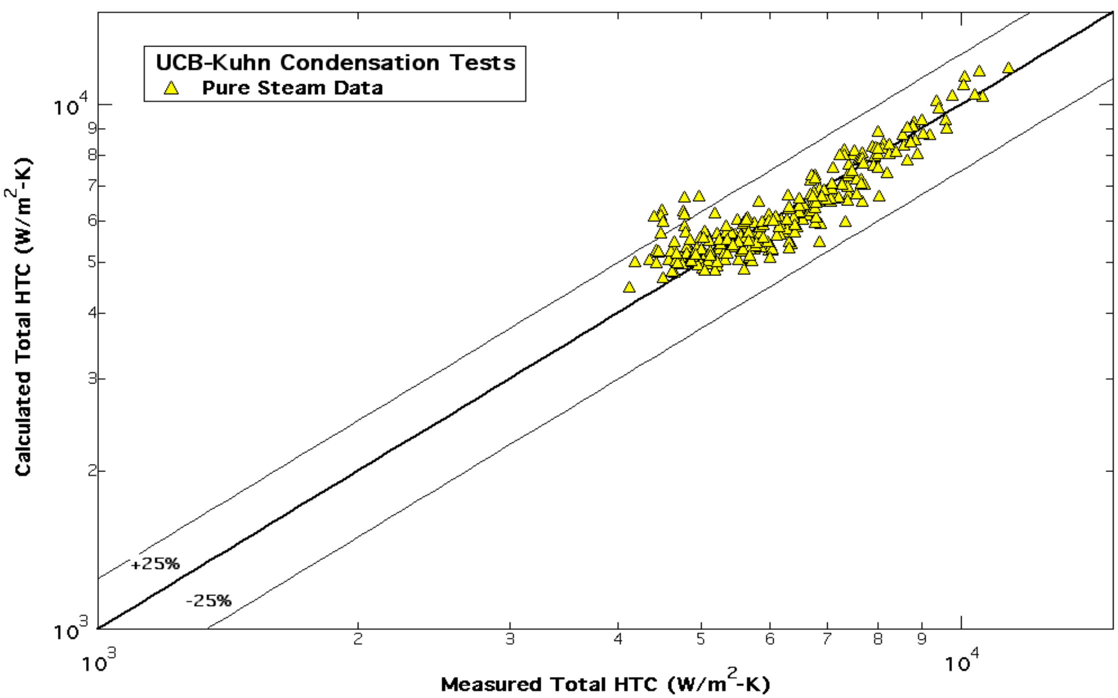
TRACE

Introduction
Model Accuracy

Experiment	No. of Data Points	Vierow-Schrock		Kuhn-Schrock-Peterson		Shah Correlation		TRACE Model	
		Avg.	RMS	Avg.	RMS	Avg.	RMS	Avg.	RMS
Pure Steam Condensation Tests									
UCB-Kuhn	252	2.652	2.975	0.031	0.083	-0.675	0.683	0.018	0.102
NASA	299	-0.021	0.668	-0.468	0.510	0.003	0.244	-0.121	0.300
Air-Steam Condensation Tests									
UCB-Kuhn	571	1.944	2.131	0.067	0.248	-	-	0.077	0.161
MIT-Siddique	255	0.878	1.210	0.770	1.277	-	-	-0.394	0.446
Helium-Steam Condensation Tests									
UCB-Kuhn	192	-	-	0.063	0.162	-	-	-0.026	0.158
MIT-Siddique	68	-	-	0.406	0.788	-	-	-0.481	0.506
MIT-Hasanein	198	-	-	0.755	1.491	-	-	-0.123	0.474

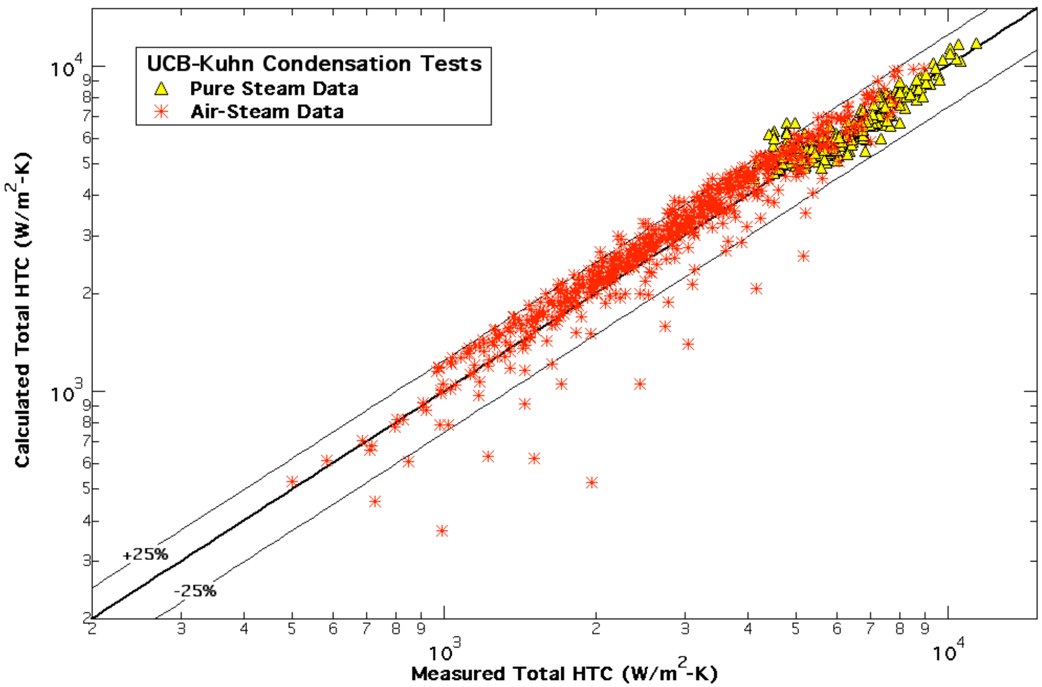
Introduction
Model Accuracy

- **UCB-Kuhn Pure Steam Tests**
  - Over-prediction occurs for points at higher film Reynolds no. and is related to the laminar-turbulent transition for the film.



Introduction
Model Accuracy

- **UCB-Kuhn Air-Steam Tests**
  - Under-prediction for points at low values of the gas/vapor Reynolds no.
    - due to persistence of turbulence (history effect), or
    - neglecting mixed convection in mass transfer.



Introduction

- **Summary:**
  - A model has been developed and implemented in TRACE for in-tube condensation that is applicable to the ICS and PCCS systems of the ESBWR design.
    - compatible with two-fluid numerical framework.
    - takes advantage of quantities computed by TRACE through the solution of the conservation equations.
  - Tube model was extended for condensation on the walls of containment volumes.
  - Accuracy of the new model:
    - Pure steam condensation:
      - nearly as accurate as empirical correlations when compared to the correlation's database.
    - Condensation with non-condensible gases:
      - accuracy is as good or better than empirical models when compared to the correlation's database and superior when compared to other data sources.

TRACE

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Film Condensation

- Normal Representation

$$q''_w = h_{cond} \cdot (T_w - T_{sat})$$

- Two-Fluid Model

$$q''_w = h_{wl} \cdot (T_w - T_l)$$

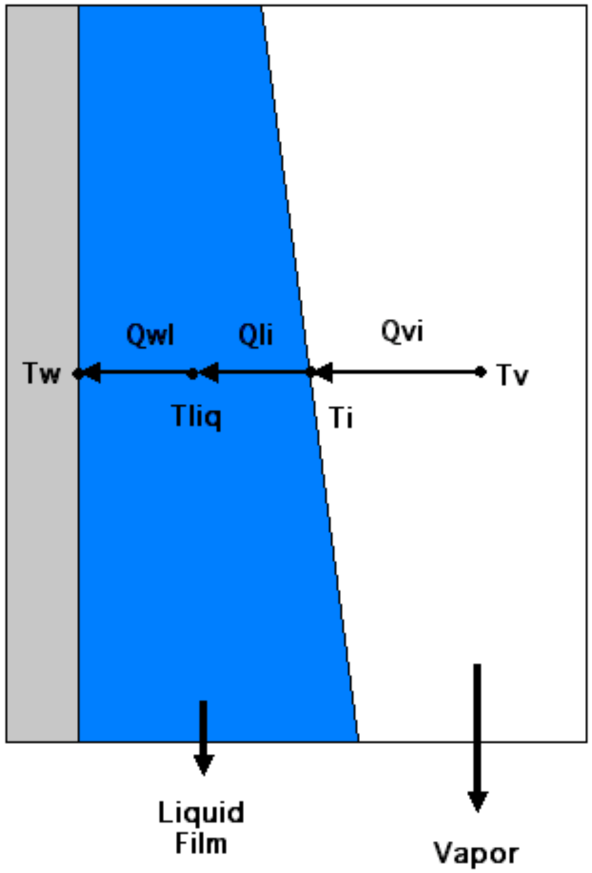
$$\Gamma = \frac{q_{li} + q_{vi}}{h_{fg}}$$

- where

$$q_{li} = h_{li} \cdot A_i \cdot (T_l - T_i)$$

$$q_{vi} = h_{vi} \cdot A_i \cdot (T_v - T_i)$$

- and T_i is the saturation temperature at the bulk vapor partial pressure.

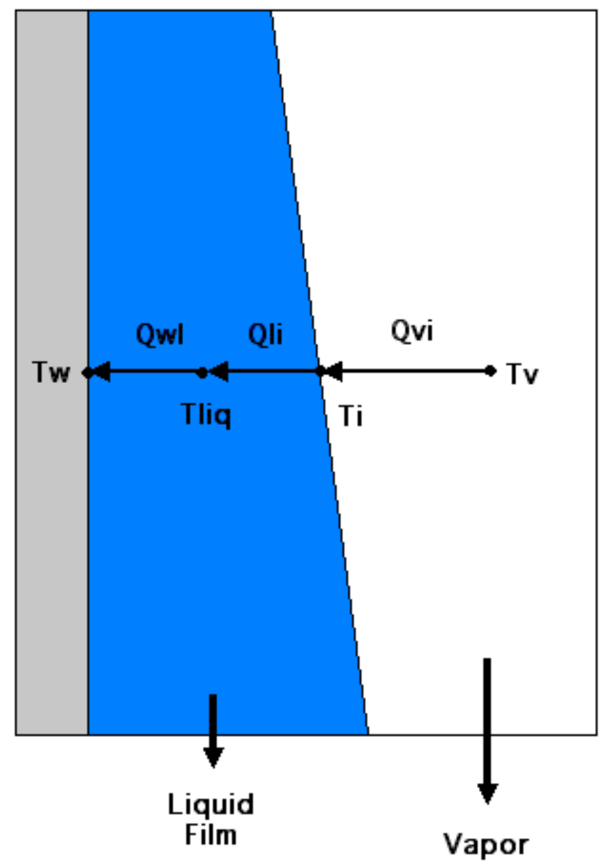


Model Description

Film Condensation

- Model Requirements
  - Condensation with pure steam and steam-NC gas mixtures
  - Applicable to both falling and sheared films

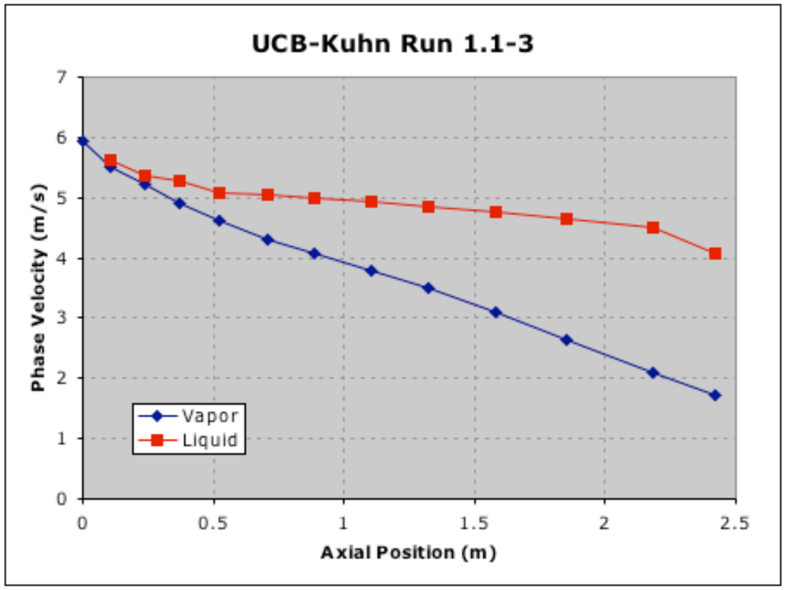
- Models Needed
  - Film Thickness
    - Wall Friction
    - Interfacial Shear
  - Wall Heat Transfer
    - Wall-Liquid HTC
  - Interfacial Heat Transfer
    - Liquid-Interface HTC
    - Vapor-Interface HTC
    - Non-Condensible Gas Effect

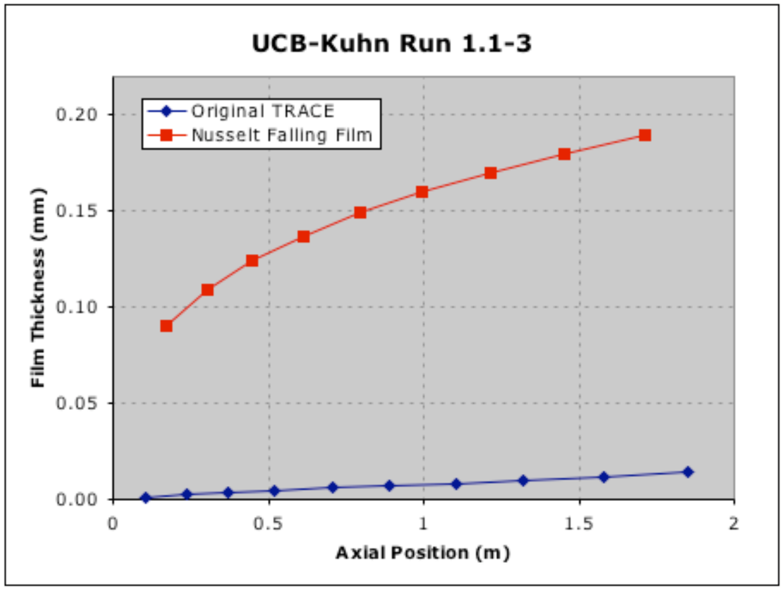


TRACE

Model Description
Wall Friction

- **Legacy TRACE Model**
  - Partitions wall drag between liquid and vapor:
    - Unphysical behavior for liquid film velocity.
    - Film thickness is an order of magnitude too small.





Model Description
Wall Friction

- **Revised Model**
  - Laminar
    - Parallel plate formula for a smooth laminar film

$$f_l = \frac{24}{\mathrm{Re}_l}$$

      - Note: will slightly over-predict film thickness due to neglecting effect of ripples, this effect will be taken into account in the wall heat transfer model.
  - Turbulent
    - Haaland explicit approximation of Colebrook-White

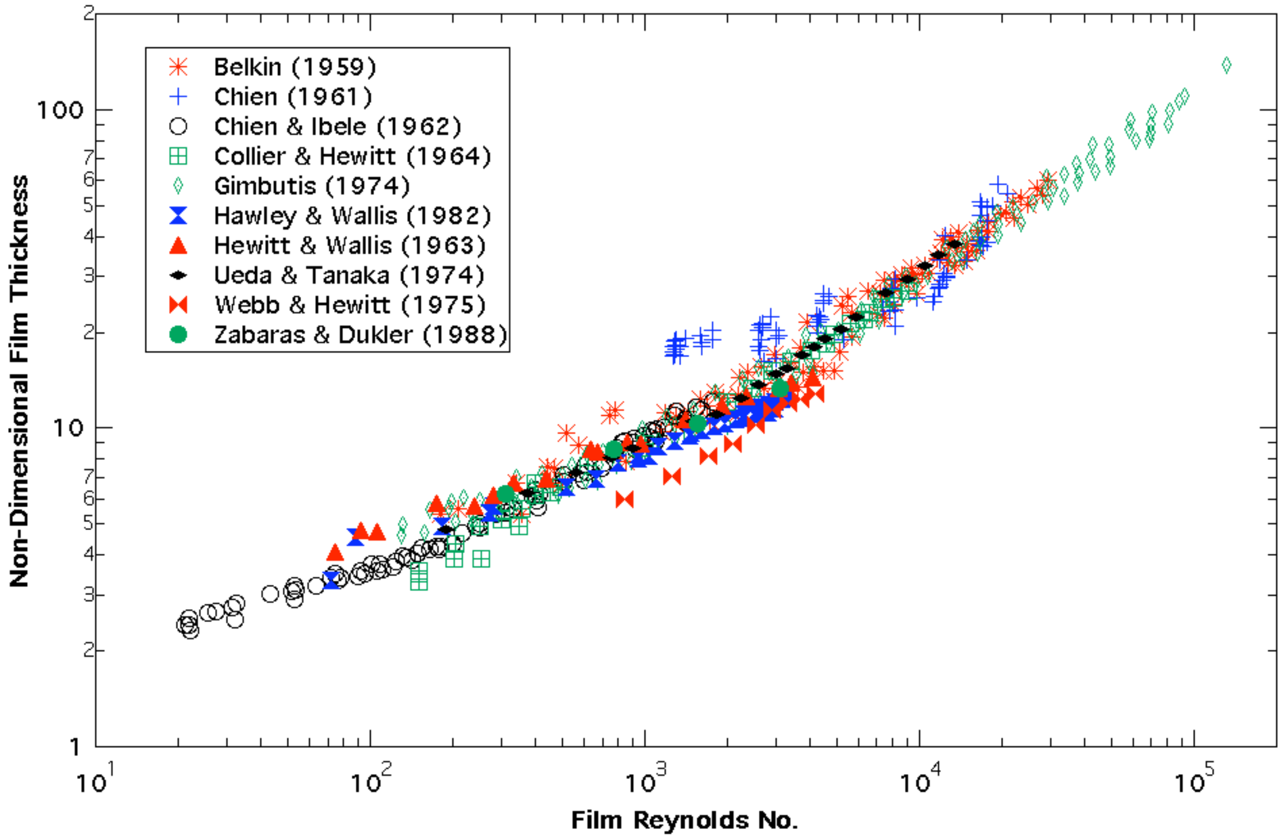
$$f_t = \left[3.6 \cdot \log_{10}\left(\frac{6.9}{\mathrm{Re}_l} + \left[\frac{\varepsilon / D}{3.7}\right]^{1.11}\right)\right]^{-2}$$

  - Power-Law Combination

$$f_w = \left[f_l^3 + f_t^3\right]^{\frac{1}{3}}$$

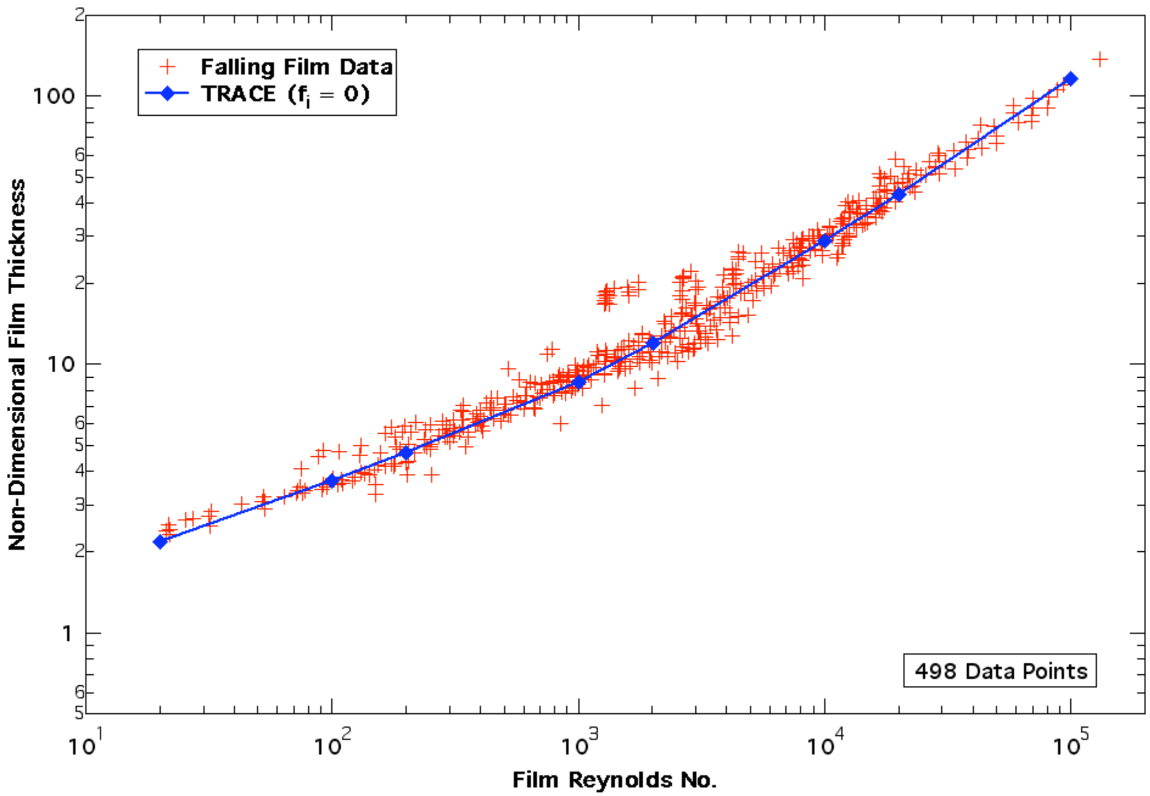
Model Description
Wall Friction

- **Film Thickness: Falling Film Data Base**



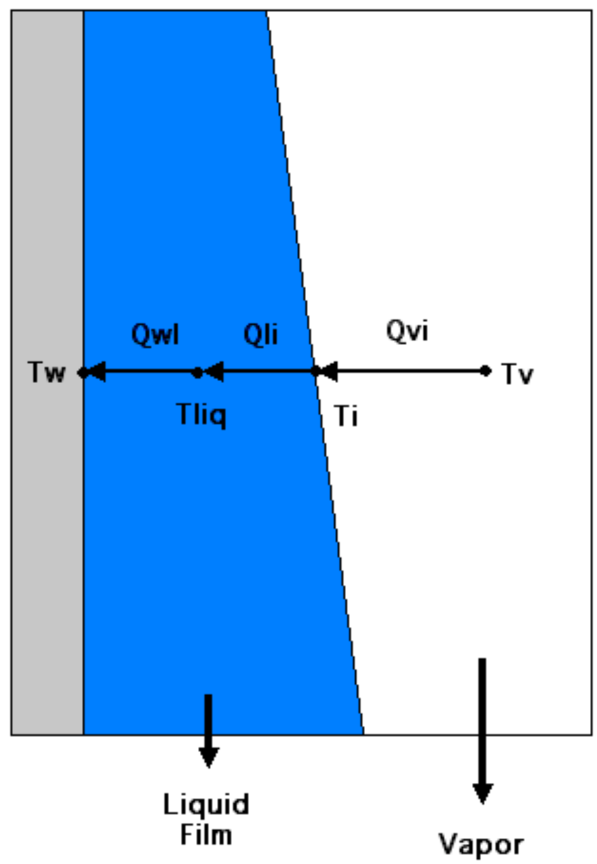
Model Description
Wall Friction

- **Film Thickness: Falling Film**
  - TRACE Results



TRACE Model Description

- **Film Condensation**
  - Model Requirements
    - Condensation with pure steam and steam-NC gas mixtures
    - Applicable to both falling and sheared films
  - Models Needed
    - Film Thickness
      - Wall Friction
      - Interfacial Shear
    - Wall Heat Transfer
      - Wall-Liquid HTC
    - Interfacial Heat Transfer
      - Liquid-Interface HTC
      - Vapor-Interface HTC
      - Non-Condensible Gas Effect



Non-Condensible Gas Effect

- Approach:
  - Use a mechanistic approach similar to the mass transfer conductance model described by Kuhn, Schrock & Peterson (1994).
- Assessment:
  - In-Tube:
    - UCB-Kuhn Steam-Air Tests
    - UCB-Kuhn Steam-Helium Tests
    - MIT-Siddique Steam-Air Tests
    - MIT-Siddique Steam-Helium Tests
    - MIT-Hasanein Steam-Helium Tests
  - Containment Wall Condensation:
    - Comparison to Uchida formula
    - MIT-Dehbi Tests
    - UWisc Flat Plate Tests

Model Description
Non-Condensible Gas Effect

Mass Transfer Conductance Model

- Liquid-Interface Heat Flux:

$$q''_{li} = h_{li} \cdot (T_i - T_l)$$

- Gas Mixture-Interface Heat Flux:

$$q''_{mix} = q''_{cond} + q''_{sens}$$
$$q''_{cond} = \Gamma'' \cdot h_{fg}$$
$$q''_{sens} = h_{sens} \cdot (T_{mix} - T_i)$$

- Iteration Required to Find Interface Temperature (Concentration)

$$q''_{li} = q''_{cond} + q''_{sens}$$

Mass Transfer Conductance: Tube Model

- Condensation Mass Flux:

$$\Gamma'' = -\left(\frac{\rho_m \cdot D}{d}\right) \cdot \beta \cdot Sh \cdot \left(\frac{M_{w,i}}{M_w}\right)^{0.4} \cdot b$$

  - Mass Transfer Driving Potential

$$b = (x_v - x_{v,i})/(x_{v,i} - 1)$$

  - Sherwood No.

$$Sh = \left[(3.66)^3 + (Gnielinski(\mathrm{Re}, Sc))^3\right]^{1/3}$$

  - “Blowing” Factor

$$\beta = \ln(1+b)/b$$

Mass Transfer Conductance: Tube Model

- Sensible Heat Flux:

$$q''_{sens} = f_{fog} \cdot \left(\frac{k_m}{d} \right) \cdot \beta \cdot Nu \cdot (T_m - T_i)$$

- Nusselt No.

$$Nu = Max[4, Gnielinski(\mathrm{Re}, \mathrm{Pr})]$$

- "Blowing" Factor

$$\beta = \ln(1 + b)/b$$

- Fog Factor

$$f_{fog} = 2$$

Model Description
Non-Condensible Gas Effect

- **Mass Transfer Conductance: Modifications for Wall Model**
  - Condensation Mass Flux:

$$\Gamma'' = -\left(\frac{\rho_m \cdot D}{d}\right) \cdot \beta \cdot Sh \cdot \left(\frac{M_{w,i}}{M_w}\right)^c \cdot b$$

  - Sherwood No.
    - Natural Convection

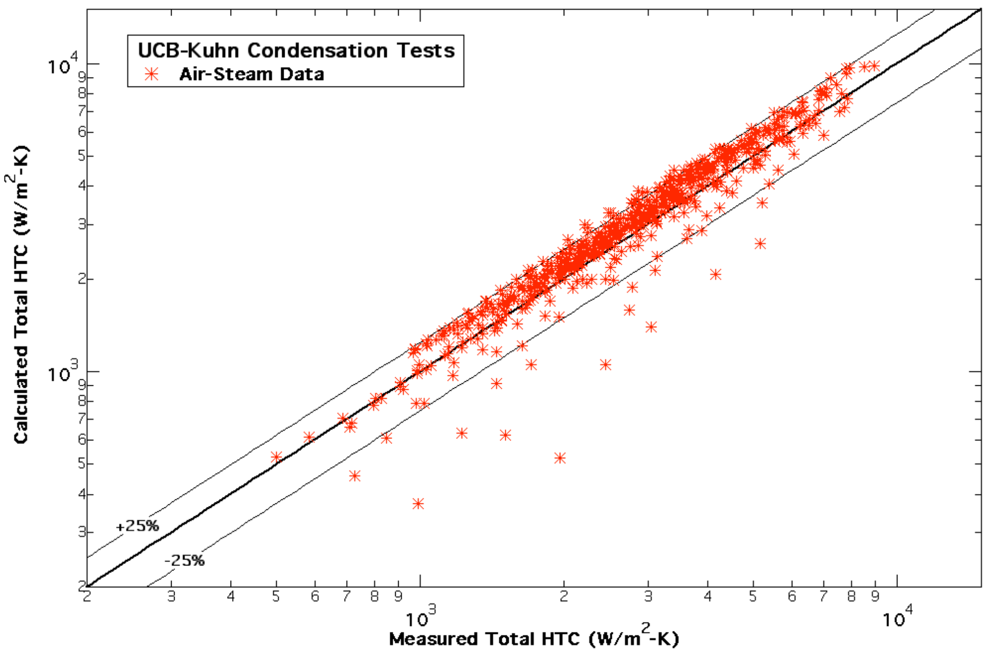
$$Sh_{NC} = 0.13 \cdot \left(Gr_L \cdot Sc\right)^{1/3}$$
$$c = 0.57$$

    - Model

$$Sh \cdot \left(\frac{M_{w,i}}{M_w}\right)^c = Max\left\{Sh_{NC} \cdot \left(\frac{M_{w,i}}{M_w}\right)^{0.57}, \ Sh_{FC} \cdot \left(\frac{M_{w,i}}{M_w}\right)^{0.4}\right\}$$

Model Description
Non-Condensible Gas Effect

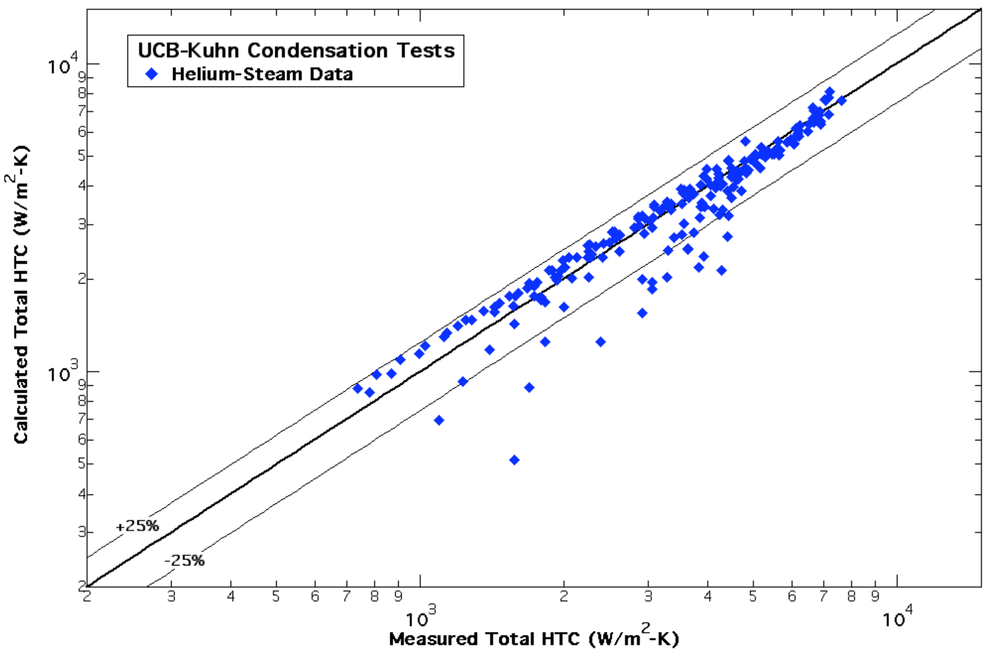
- **Comparison to UCB-Kuhn Experiment**
  - Steam-Air Data: (72 tests, 571 data points)
    - Average Error = 7.7%
    - RMS Error = 16.1%



TRACE

Model Description
Non-Condensible Gas Effect

- **Comparison to UCB-Kuhn Experiment**
  - Steam-Helium Data: (25 tests, 192 data points)
    - Average Error = -2.6%
    - RMS Error = 15.8%



TRACE

TRACE Condensation Model Development for ESBWR

CONTENTS

- Introduction
  - Background and Status
  - Modeling Approach
  - Model Accuracy
- New Model Description
  - Wall Friction
  - Interfacial Shear
  - Wall-Fluid Heat Transfer
  - Interfacial Heat Transfer
  - Non-Condensible Gas Effect
- Sample of TRACE Assessment Results
- Summary

TRACE

Sample of TRACE Assessment Results

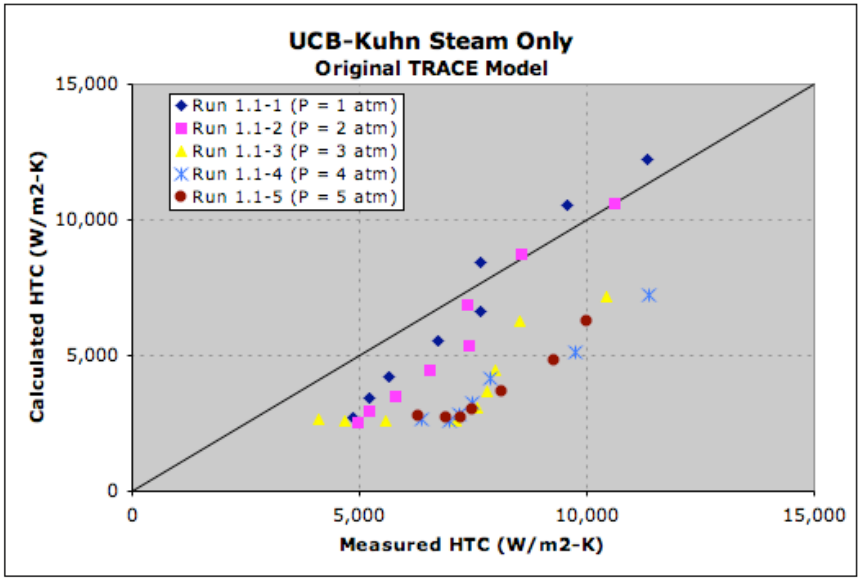
- **Test Matrix: Tube Condensation**

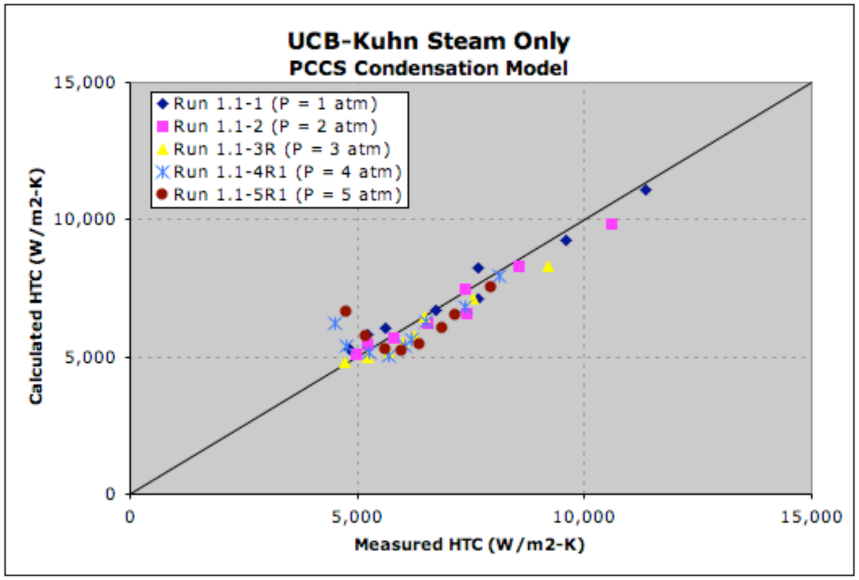
	Run No.	Pressure (bar)	Gas Reynolds No.	Film Reynolds No.	NC Gas Mass Fraction (%)
Laminar Film UCB - Kuhn (Steam Only)	1.1-1	1.16	35,400	450	-
	1.1-2	2.02	33,900	720	-
	1.1-3R	3.20	30,160	1270	-
	1.1-4R1	4.10	29,800	1820	-
	1.1-5R1	5.04	28,930	1970	-
Turbulent Film NASA - Goodykoontz	172	1.77	85,980	3020	-
	174	1.78	105,240	3800	-
NC Gas Effect UCB – Kuhn (Air-Steam)	2.1-1	4.20	23,960	1360	1
	2.1-4	3.93	25,590	1010	4.2
	2.1-7	4.00	26,960	750	10
	2.1-9	4.05	29,280	610	20
	2.1-13	4.15	36,620	440	40

TRACE

Sample of TRACE Assessment Results

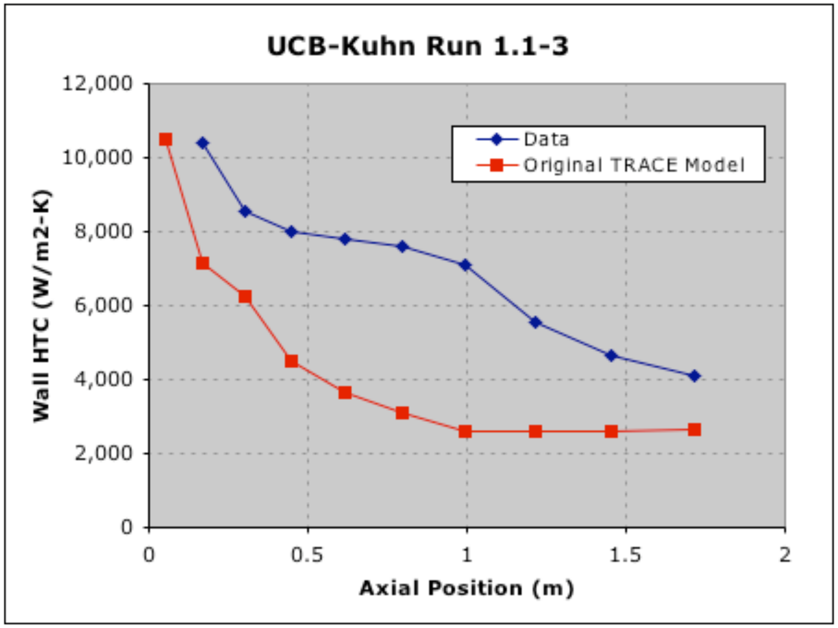
- **Laminar Film Condensation:**
  - Calculation dramatically improved, good prediction over entire pressure range.
  - Few values over-predicted due to laminar-turbulent transition.

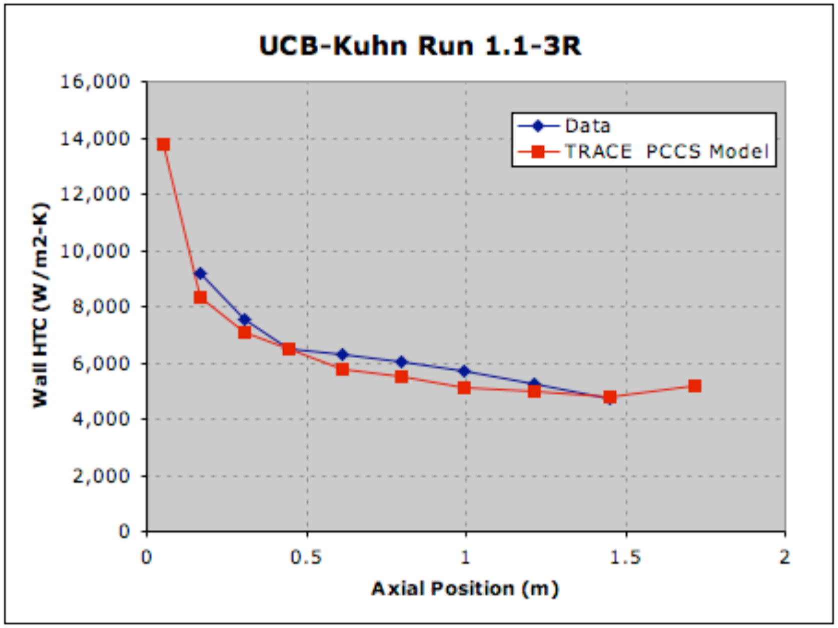




Sample of TRACE Assessment Results

- **Laminar Film Condensation:**
  - Excellent prediction of the heat transfer coefficient both in magnitude and axial trend.

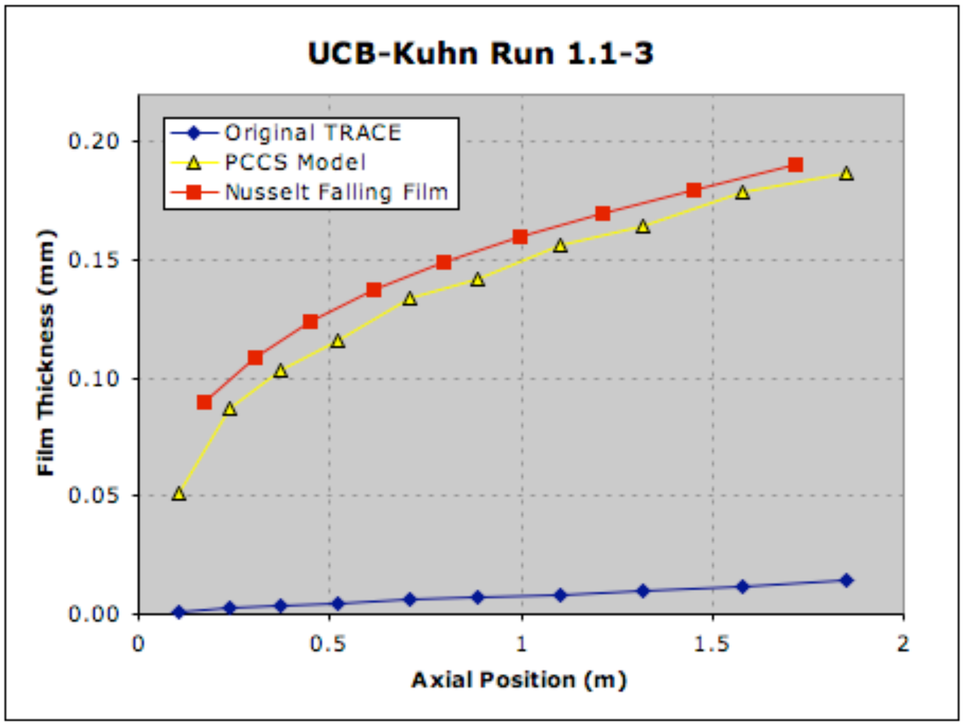




Sample of TRACE Assessment Results

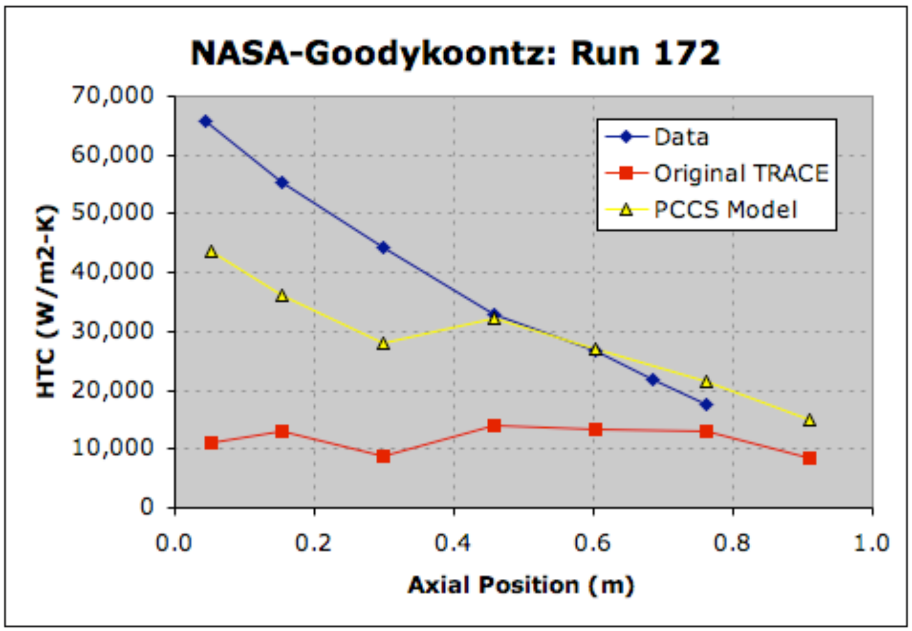
TRACE

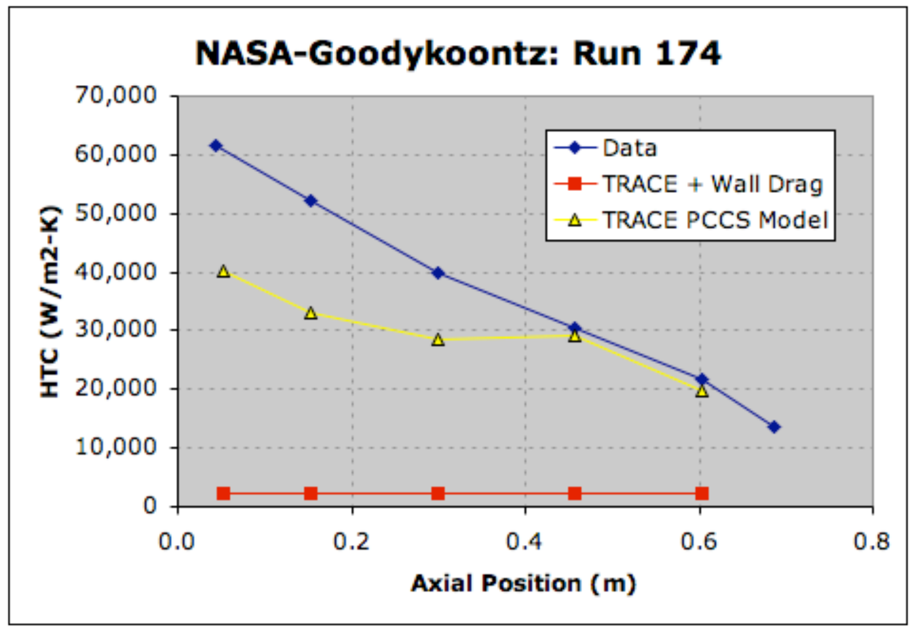
- **Laminar Film Condensation**
  - Realistic calculation of liquid film thickness.



Sample of TRACE Assessment Results

- **Turbulent Film Condensation:**
  - Significantly improved prediction of heat transfer coefficient, but
  - Under-prediction in laminar regime with good prediction once film becomes turbulent.





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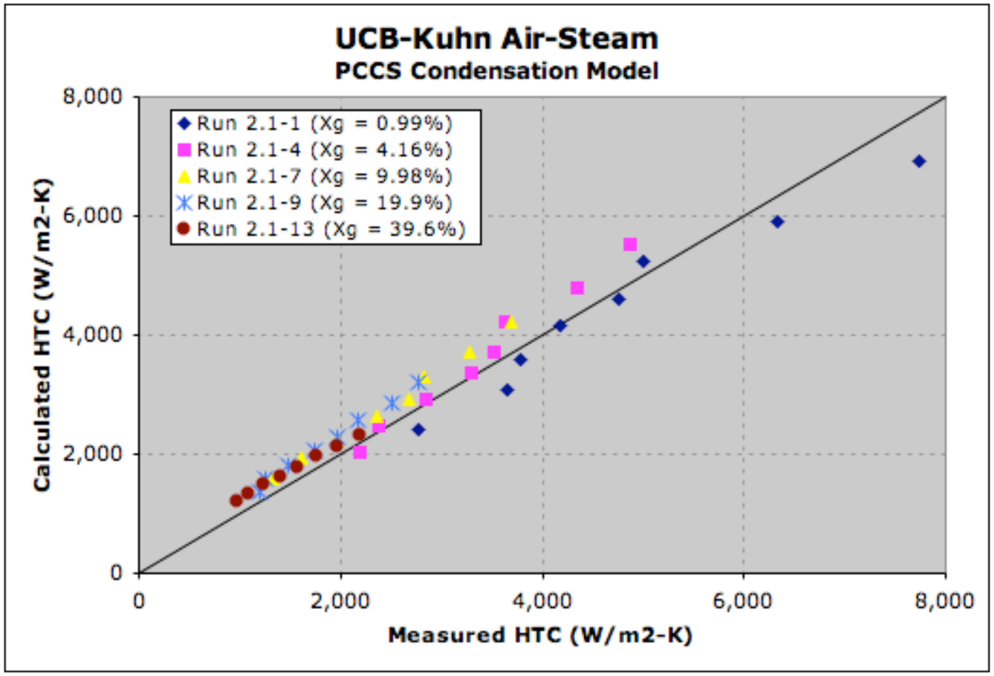
Sample of TRACE Assessment Results

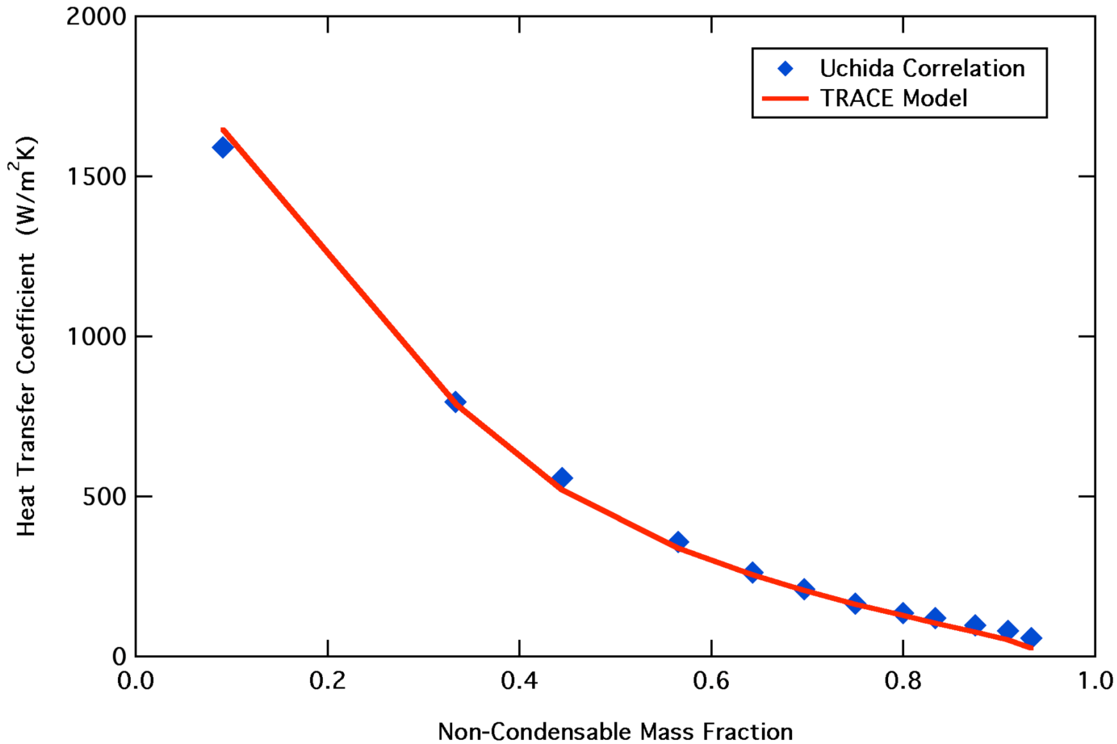
- **Non-Condensible Gas Effect:**
  - Excellent prediction for a range of inlet non-condensible mass fraction from 1% to 40%.



Sample of TRACE Assessment Results

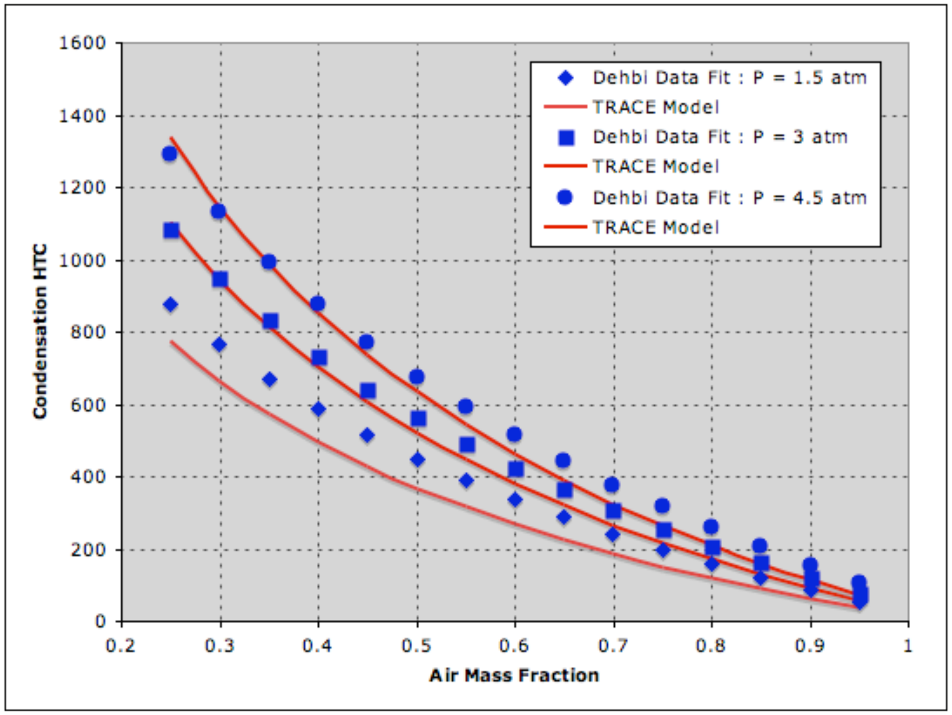
Wall Condensation Model:

- Excellent comparison to empirical model of Uchida.



Sample of TRACE Assessment Results

- **Wall Condensation Model:**
  - Good comparison to MIT-Dehbi test data.



TRACE

TRACE Condensation Model Development for ESBWR

- **Summary:**
  - A new condensation model has been developed and implemented in TRACE.
    - Applicable to the ICS and PCCS systems of the ESBWR design.
    - Extended to wall condensation in large containment volumes.
    - Compatible with two-fluid numerical framework.
    - Takes advantage of quantities computed by TRACE:
      - e.g., axial distribution of the condensate flow rate and film thickness
  - Accuracy of the new model
    - For pure steam condensation, nearly as accurate as empirical correlations when compared to the database of the empirical model.
    - With non-condensible gases, accuracy is as good or better than empirical models developed from that data and superior when compared to other data sources.
  - Assessment
    - Developmental assessment of TRACE has been performed against single-tube experiments.
      - Excellent predictions of UCB-Kuhn pure steam and air-steam tests.
      - Reasonable prediction of NASA pure steam tests.
    - Very good comparisons to containment wall condensation data & empirical models.
    - More extensive ESBWR specific testing has been conducted, including multi-tube exchangers (e.g., PANTHERS).

Selection of Interfacial Friction Model

- Use data of Andreussi-Zanelli for co-current downflow
  - Measured film thickness, pressure gradient and entrainment fraction.
  - Reduced data to give values of the interfacial friction coefficient.
- Compare interfacial friction models of
  - Wallis (1969)
  - Modified Wallis
    - uses friction factor as fn(Re)
  - Henstock-Hanratty (1976)
  - Bharathan (1979)
    - developed for counter-current flow
  - Asali-Hanratty (1985)
    - models with and without entrainment
  - Jayanti-Hewitt (1997)
    - ripple & disturbance wave models

TRACE

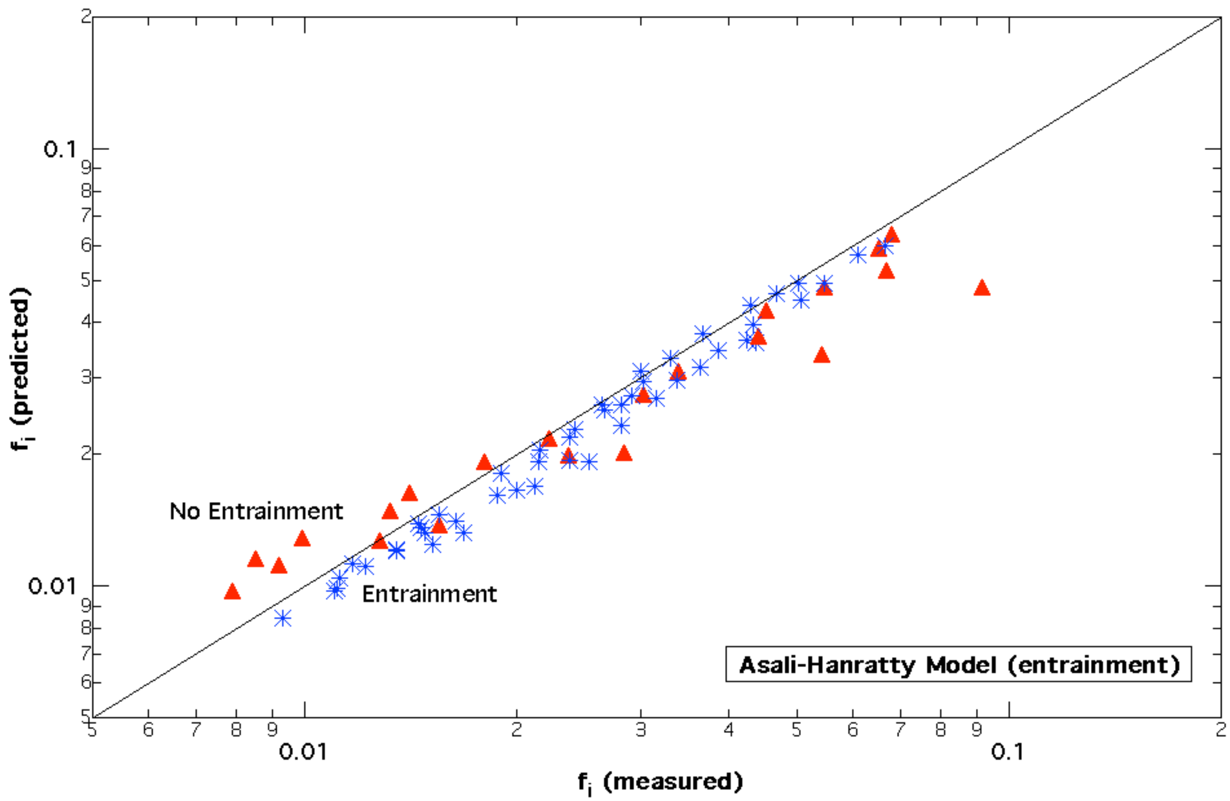
Model Description
Interfacial Friction

Selection of Interfacial Friction Model

Model	Average Error	Maximum Error	RMS Error
Wallis (f_s = 0.005)	−0.176	−2.801	0.517
Wallis (f_s = f(Re))	0.400	5.055	1.480
Henstock–Hanratty	2.266	11.13	1.489
Asali–Hanratty (no entrainment)	−0.165	−0.642	0.049
Asali–Hanratty (entrainment)	−0.076	0.364	0.0226
Bharathan	1.425	6.160	6.612
Jayanti–Hewitt (ripple wave)	−0.396	−0.8252	0.212
Jayanti–Hewitt (disturbance wave)	−0.453	0.718	0.302

Model Description
Interfacial Friction

- **Selection of Interfacial Friction Model**



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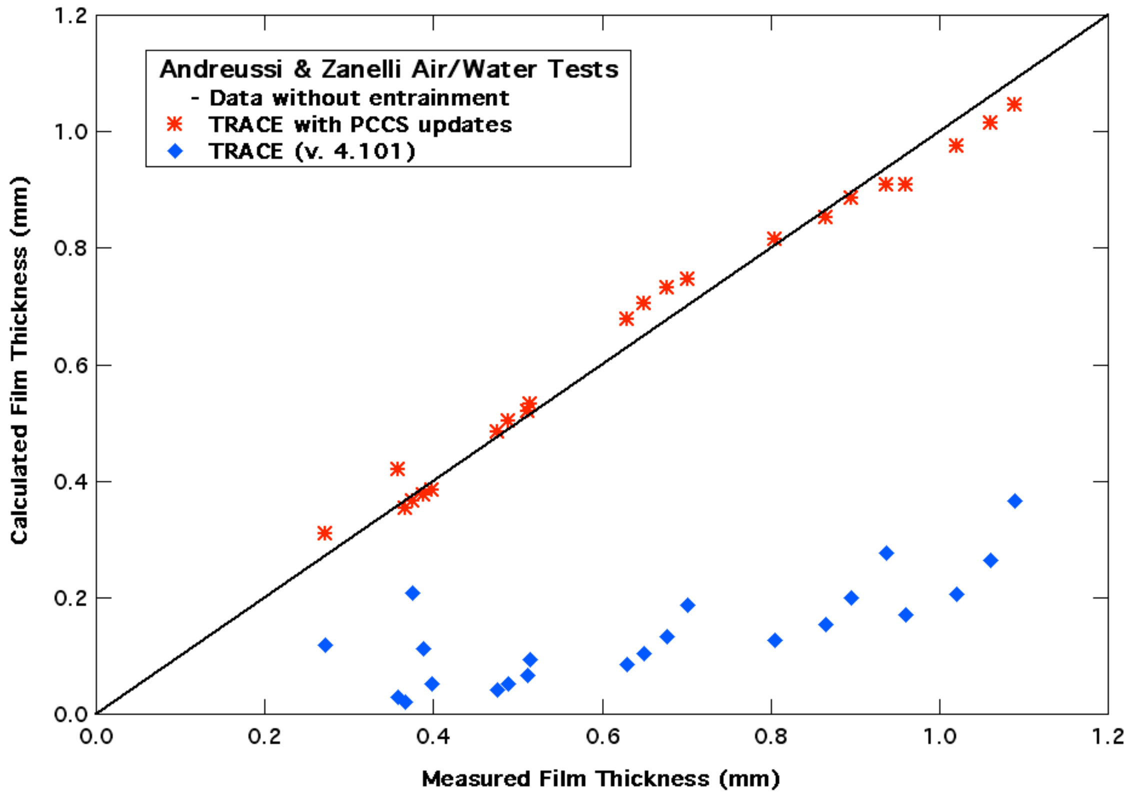
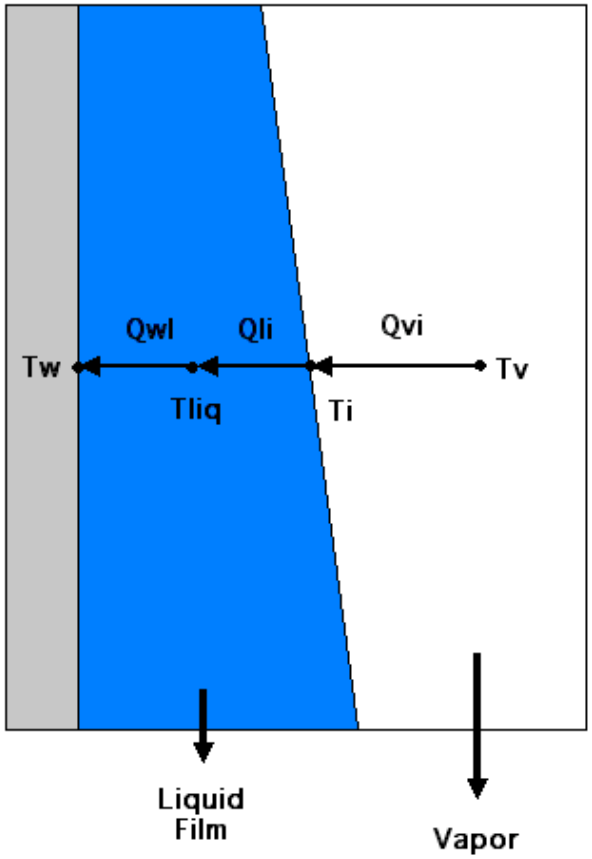
Model Description
Interfacial Friction

- **Film Thickness: Sheared Films**
  - TRACE Results for data of Andreussi & Zanelli



Film Condensation

- Model Requirements
  - Condensation with pure steam and steam-NC gas mixtures
  - Applicable to both falling and sheared films
- Models Needed
  - Film Thickness
    - Wall Friction
    - Interfacial Shear
  - Wall Heat Transfer
    - Wall-Liquid HTC
  - Interfacial Heat Transfer
    - Liquid-Interface HTC
    - Vapor-Interface HTC
    - Non-Condensible Gas Effect



TRACE

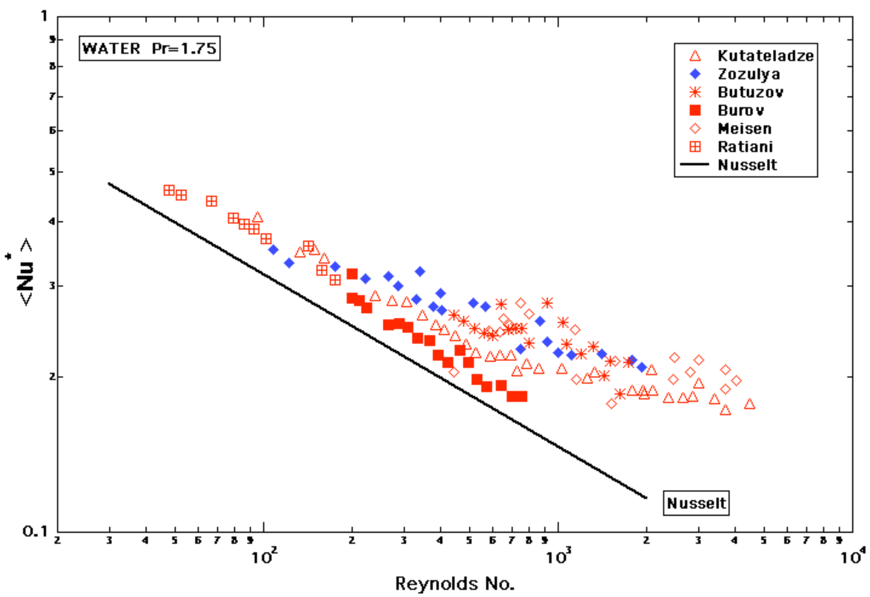
Model Description
Wall Heat Transfer

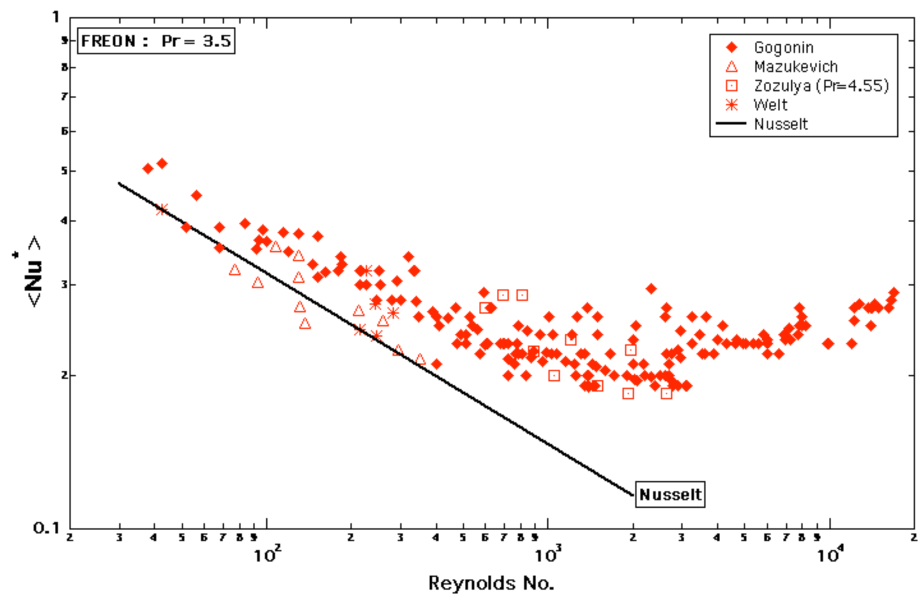
- **Modeling needs**
  - Wall-Film heat transfer coefficient
    - Laminar/smooth and laminar/wavy films
    - Turbulent films
- **Approach**
  - Laminar Films:
    - Use falling film condensation data to select suitable correlation.
    - Compare to pure-steam data of UCB-Kuhn.
    - Split heat transfer resistance between wall and interfacial.
  - Turbulent Films:
    - Use falling film heating data to select suitable correlation.
    - Interfacial heat transfer considered separately.

TRACE

Model Description
Wall Heat Transfer

- **Film Condensation: Falling Films**
  - Data Base Example:
    - Heat transfer averaged over surface (no local values).
    - Significant enhancement over Nusselt due to waves.





Laminar Model Selection

- Film Condensation: Falling Films

  - Nusselt: $\langle Nu_0^* \rangle = 1.47 \cdot \mathrm{Re}_f^{-1/3} \quad \Rightarrow \quad Nu_\delta = 1$

  - Kutateladze: $\langle Nu_0^* \rangle = 1.23 \cdot \mathrm{Re}_f^{-1/4} \quad \Rightarrow \quad Nu_\delta = 0.895 \cdot \mathrm{Re}_f^{1/12}$

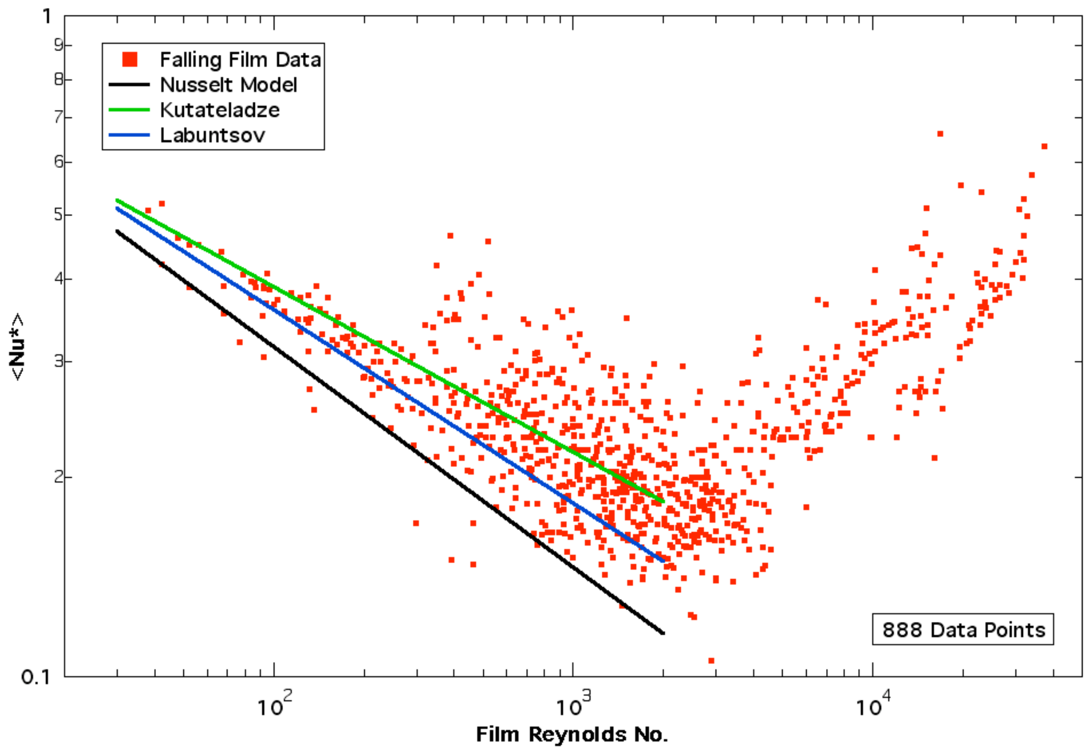
  - Nozhat: $\langle Nu_0^* \rangle = 1.28 \cdot \mathrm{Re}_f^{-0.263} \quad \Rightarrow \quad Nu_\delta = 0.921 \cdot \mathrm{Re}_f^{0.07}$

  - Zazuli: $\langle Nu_0^* \rangle = 0.955 \cdot \mathrm{Re}_f^{-0.22} \quad \Rightarrow \quad Nu_\delta = 0.8 \cdot \left(\mathrm{Re}_f/4\right)^{0.11}$

  - Labuntsov: $\langle Nu_0^* \rangle = 1.346 \cdot \mathrm{Re}_f^{-0.293} \quad \Rightarrow \quad Nu_\delta = \left(\mathrm{Re}_f/4\right)^{0.04}$

Wall Heat Transfer

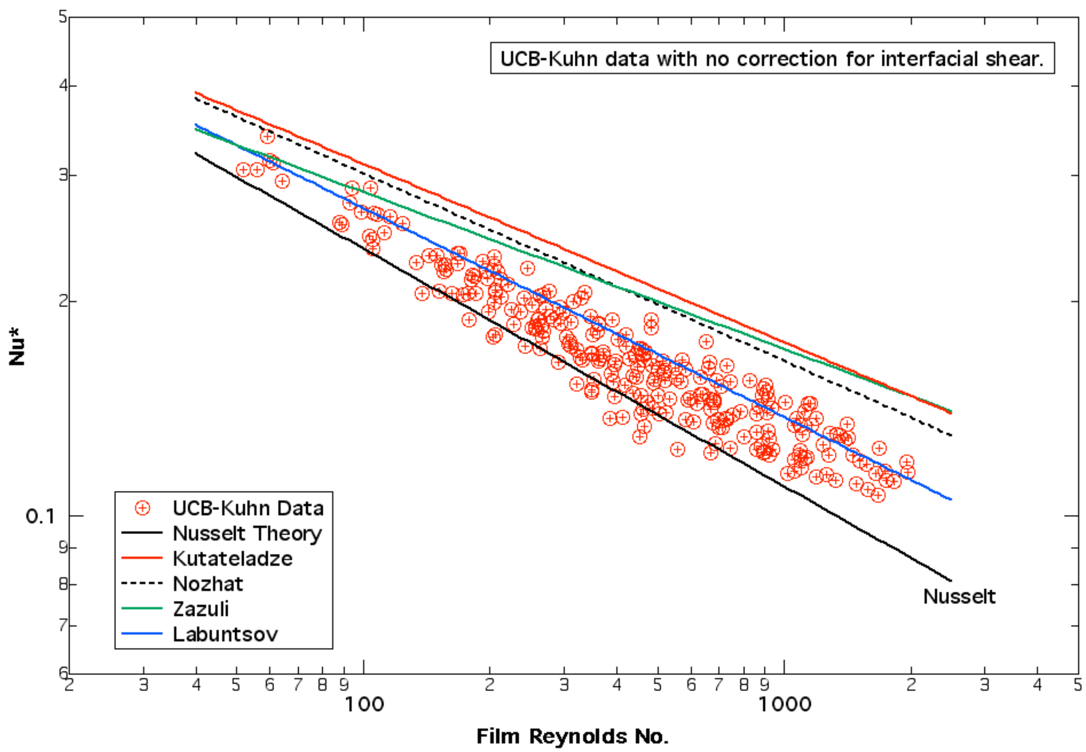
- **Laminar Model Selection**
  - Falling Film Condensation: surface average data



Model Description
Wall Heat Transfer

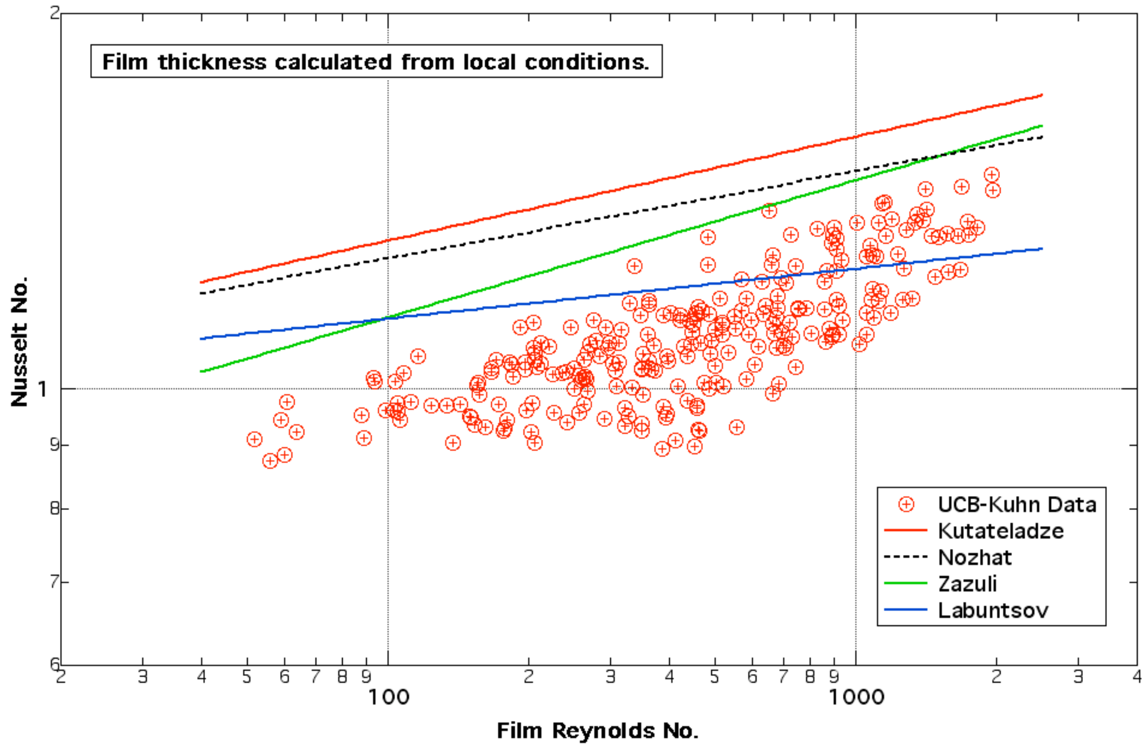
Laminar Model Selection

- Sheared Film: local data from UCB-Kuhn pure steam tests



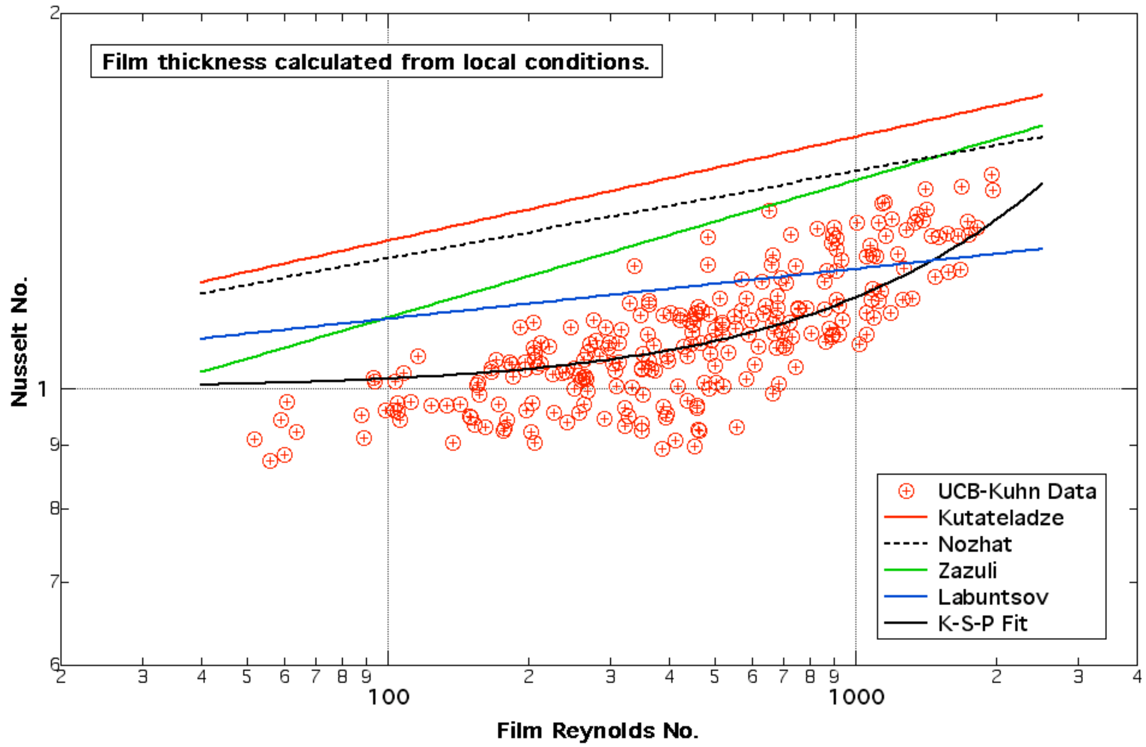
Model Description
Wall Heat Transfer

- **Laminar Model Selection**
  - Sheared Film: local data from UCB-Kuhn pure steam tests



Laminar Model Selection

- Use Kuhn-Schrock-Peterson Fit: $Nu_\delta = \left(1 + 1.83\text{x}10^{-4} \cdot \text{Re}_f\right)$

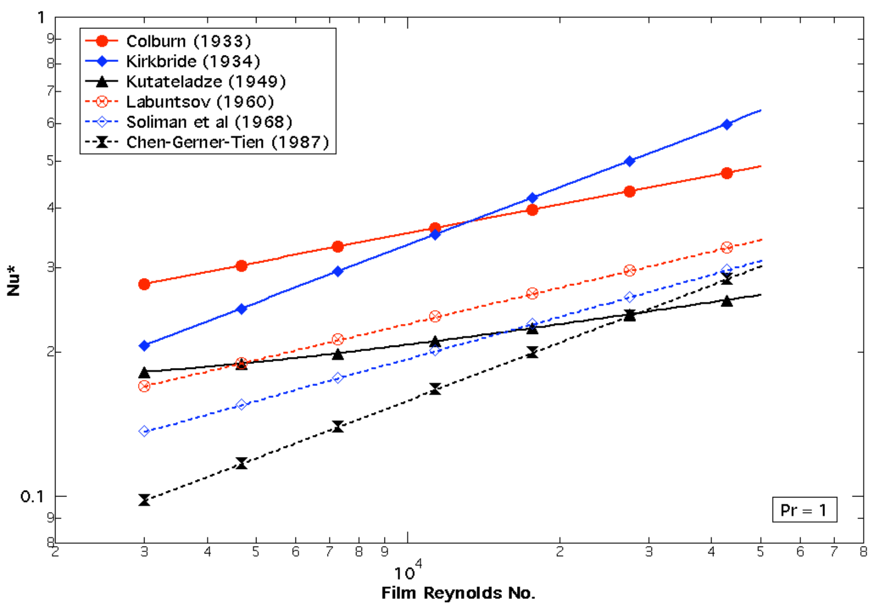


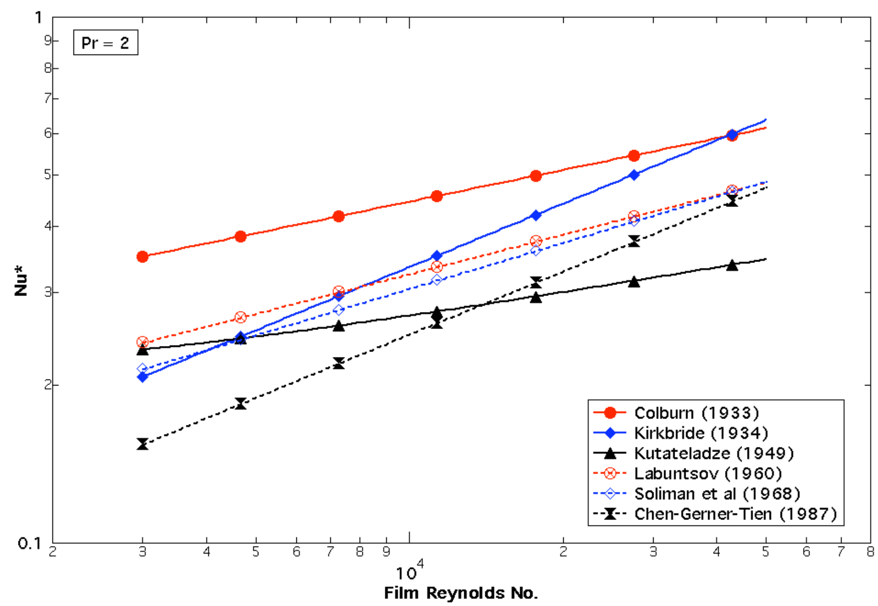
Turbulent Film Model

- Difficulty:
  - Falling film database does not have local heat transfer data, only values averaged over the entire heat transfer surface.
    - Data is integrated over both laminar and turbulent regions, and so
    - Cannot be used in a straightforward model selection process.
  - Sheared film data (e.g., NASA) have large uncertainties.
    - Data uncertainty & effect of interfacial shear on film thickness.
  - Correlations for turbulent condensation vary widely.
- Approach:
  - For the wall-liquid HTC, turbulent falling film heating data will be used for model selection.
  - For interfacial heat transfer, considered later, several models will be selected from the literature and compared to data.

Model Description
Wall Heat Transfer

- **Turbulent Falling Film Condensation:**
  - Large variation between condensation correlations for turbulent falling films.
    - Which correlation is “right” ?

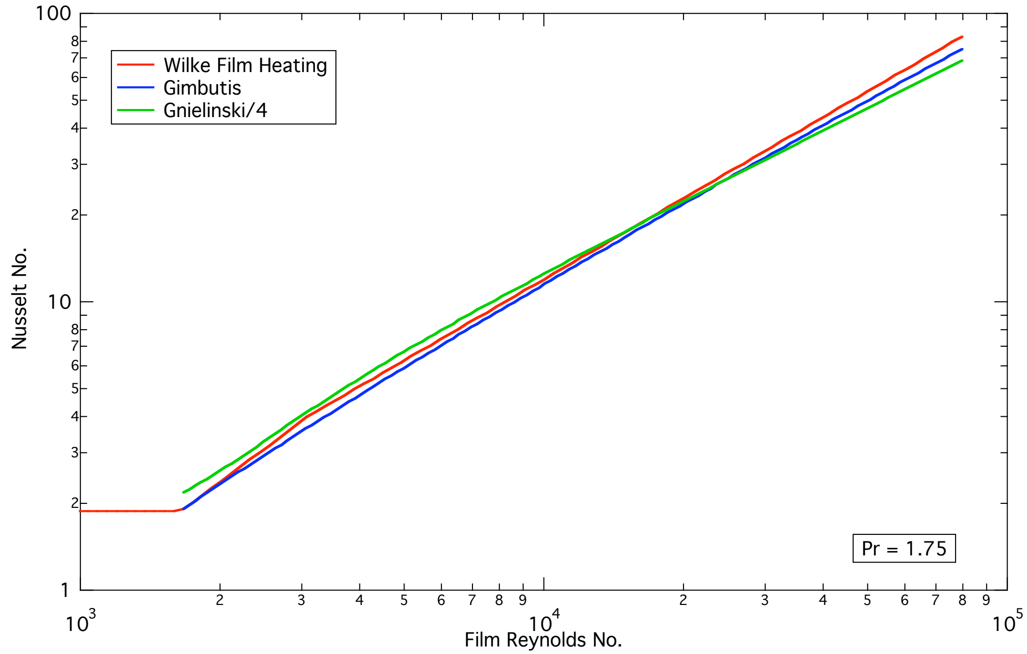




Model Description
Wall Heat Transfer

Turbulent Falling Films

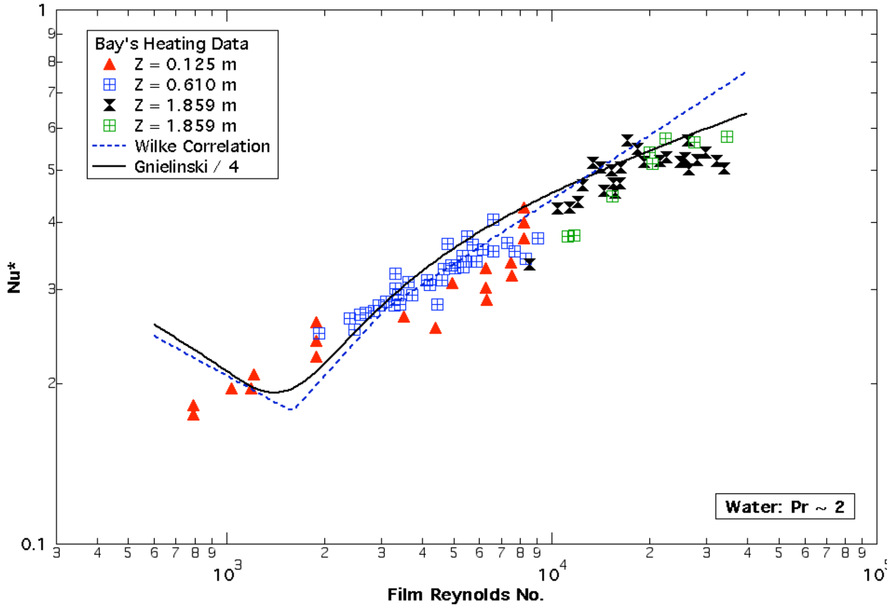
- Example of correlations for film heating:
  - Gnielinski is a modern correlation for single-phase forced convection modified for a film.

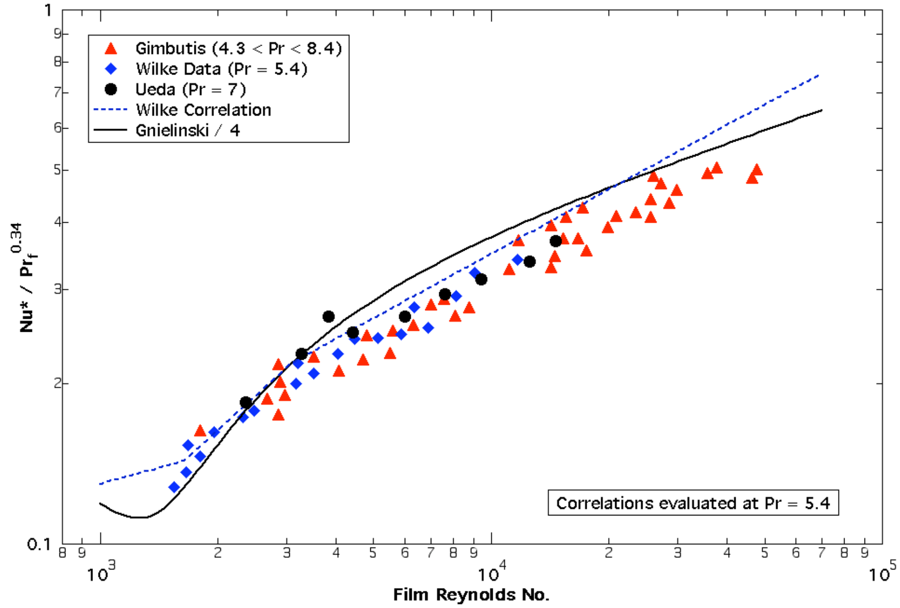


TRACE

Model Description
Wall Heat Transfer

- **Turbulent Falling Films**
  - Use correlation of Gnielinski modified for a film.





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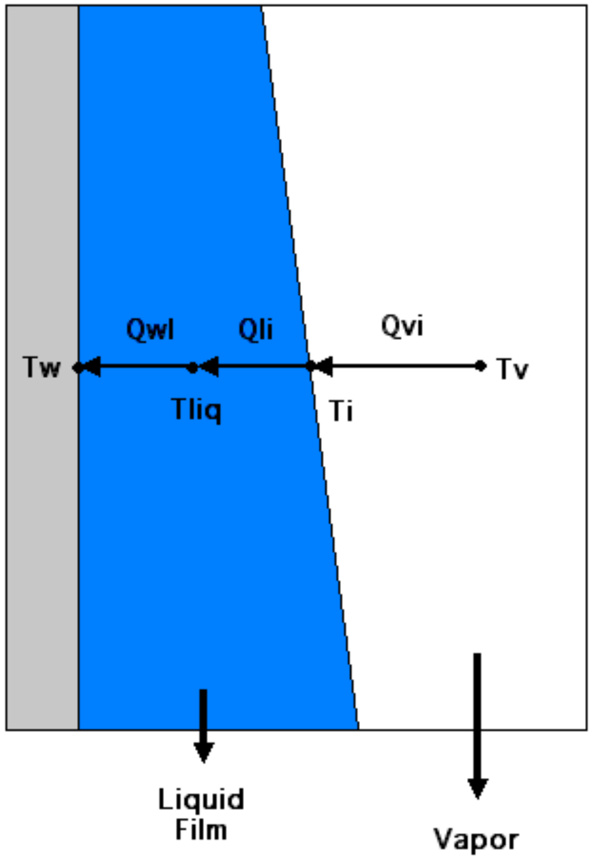
Model Description:
Wall Heat Transfer

- **Turbulent Falling Film Condensation:**
  - Colburn (1933) $$Nu^* = 0.056 \cdot \mathrm{Re}_f^{0.2} \cdot \mathrm{Pr}_f^{\frac{1}{3}}$$
  - Kirkbride (1934) $$Nu^* = 0.0084 \cdot \mathrm{Re}_f^{0.4}$$
  - Kutateladze (1949) $$Nu^* = \frac{0.0429 \cdot \mathrm{Re}_f \cdot \mathrm{Pr}_f^{0.4}}{\left(\mathrm{Re}_f^{\frac{5}{6}} - 149.2 + 66.7 \cdot \mathrm{Pr}_f^{0.4}\right)}$$
  - Labuntsov (1960) $$Nu^* = 0.023 \cdot \mathrm{Re}_f^{0.25} \cdot \mathrm{Pr}_f^{\frac{1}{2}}$$
  - Soliman et al (1968) $$Nu^* = 0.0132 \cdot \mathrm{Re}_f^{0.292} \cdot \mathrm{Pr}_f^{0.65}$$
  - Chen, Gerner & Tien (1987) $$Nu^* = 0.004 \cdot \mathrm{Re}_f^{0.4} \cdot \mathrm{Pr}_f^{0.65}$$

- **Film Condensation**
  - Model Requirements
    - Condensation with pure steam and steam-NC gas mixtures
    - Applicable to both falling and sheared films
  - Models Needed
    - Film Thickness
      - Wall Friction
      - Interfacial Shear
    - Wall Heat Transfer
      - Wall-Liquid HTC
    - Interfacial Heat Transfer
      - Liquid-Interface HTC
      - Vapor-Interface HTC
      - Non-Condensible Gas Effect



Model Description
Interfacial Heat Transfer

- **Modeling needs**
  - Liquid-Interface heat transfer coefficient
    - Laminar and laminar/wavy films
    - Turbulent films
- **Approach**
  - Laminar Films:
    - Use Kuhn-Schrock-Peterson correlation.
    - Split heat transfer resistance between wall and interfacial.
  - Turbulent Films:
    - Use NWU co-current flow condensation data to select correlation.
    - Look at other NWU data and UCB-Kuhn for the transition region.

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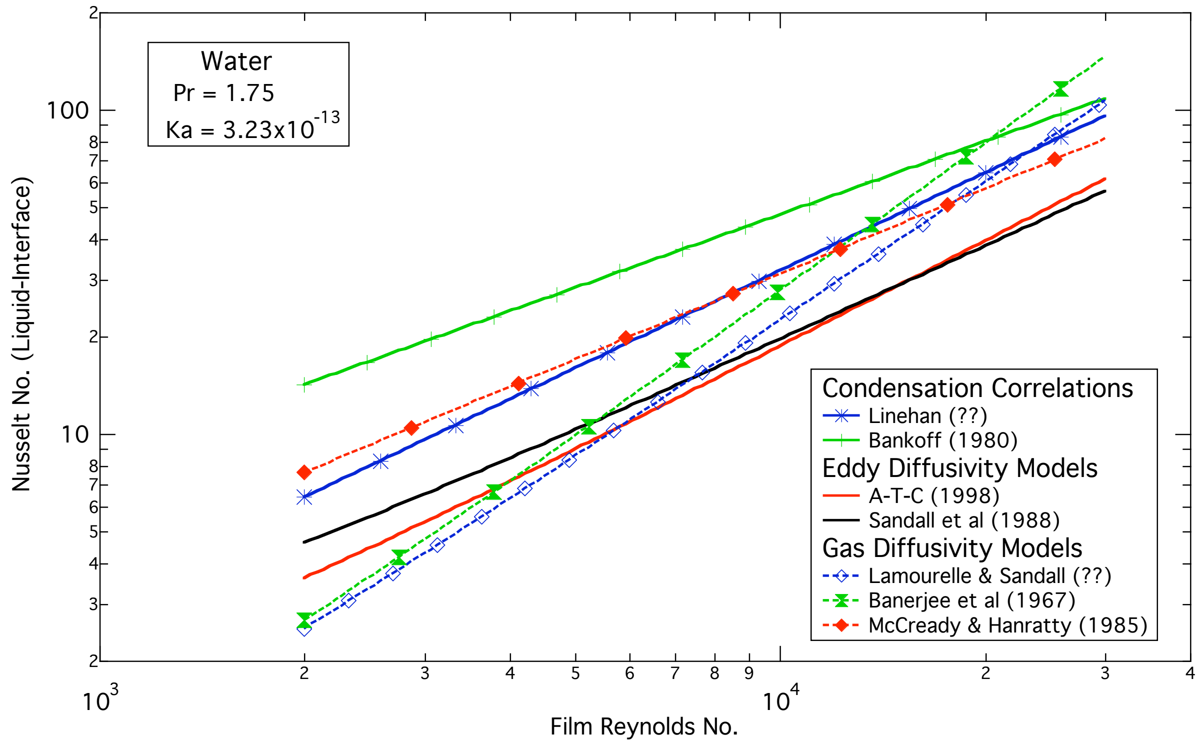
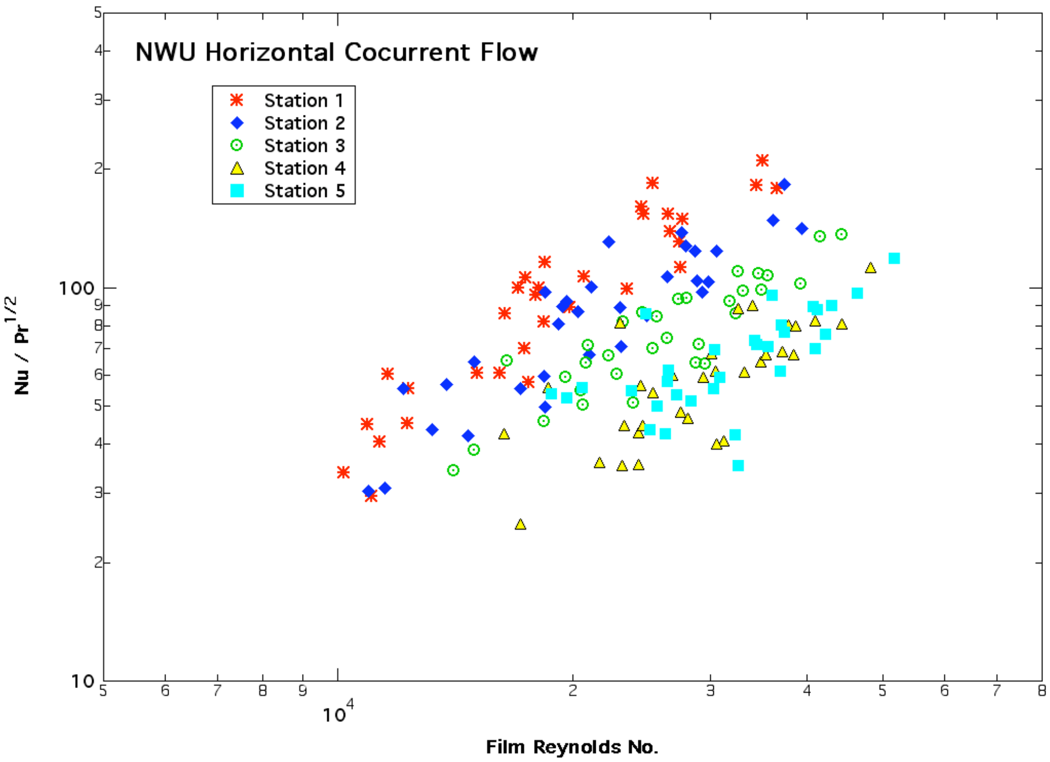
Model Description
Interfacial Heat Transfer

- **Turbulent Falling Films**
  - Example of Candidate Interfacial HTCs:



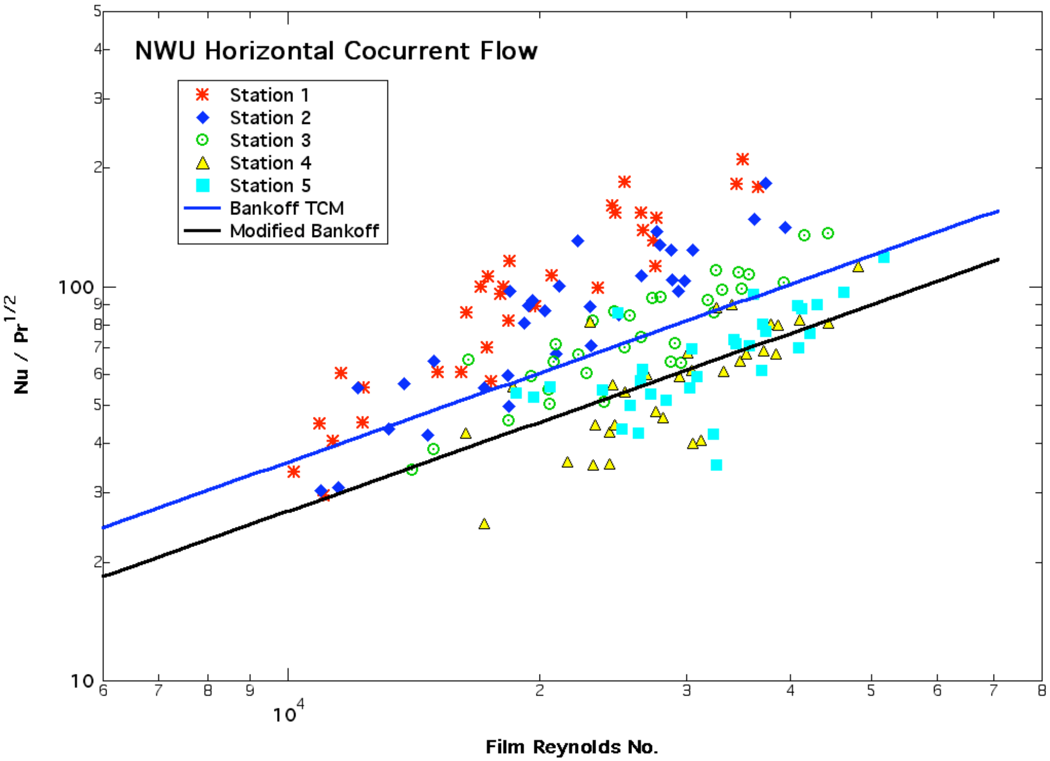
Model Description
Interfacial Heat Transfer

- **NWU Interfacial Condensation Data**
  - Exhibits significant entrance effect.



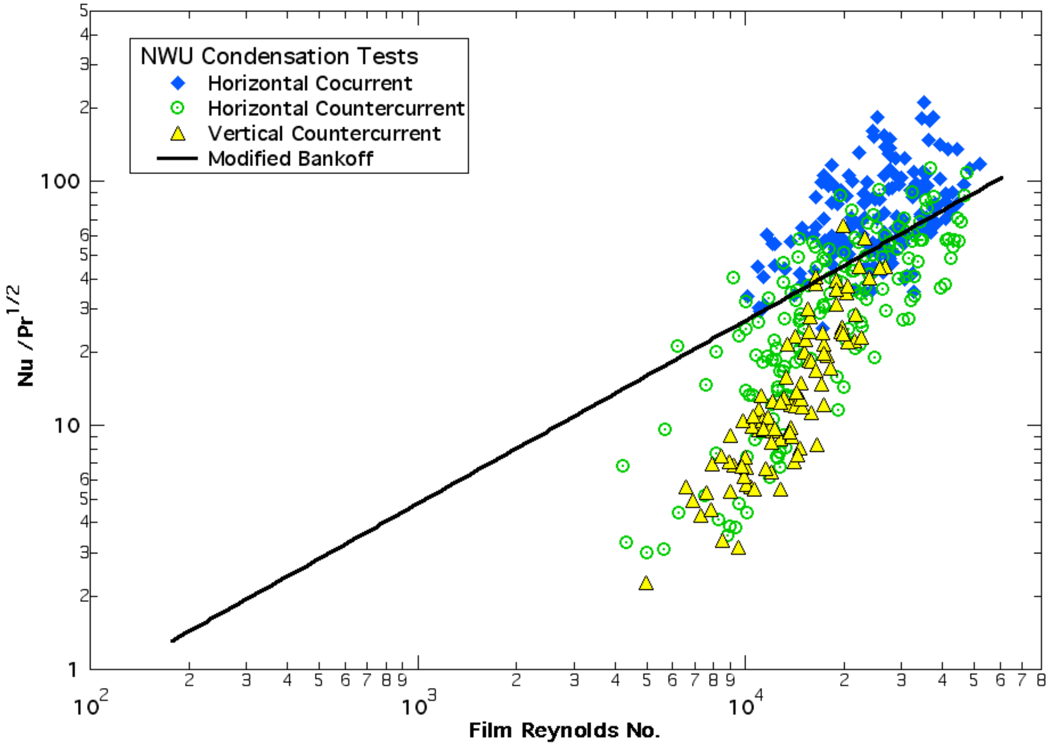
Model Description
Interfacial Heat Transfer

- **NWU Interfacial Condensation Data**
  - Use Bankoff Turbulence Centered Model
    - Modify coefficient to match fully developed data.



Model Description
Interfacial Heat Transfer

- **NWU Interfacial Condensation Data**
  - What about transition region?
    - Look at other NWU data.



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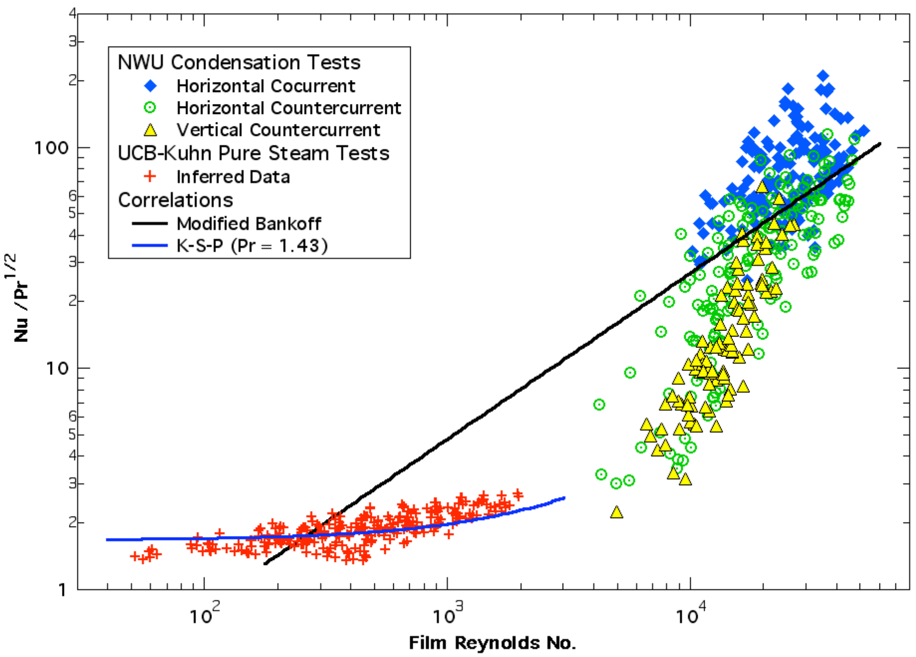
Model Description
Interfacial Heat Transfer

- **NWU Interfacial Condensation Data**
  - What about transition region?
    - Bankoff TCM would significantly over-predict.



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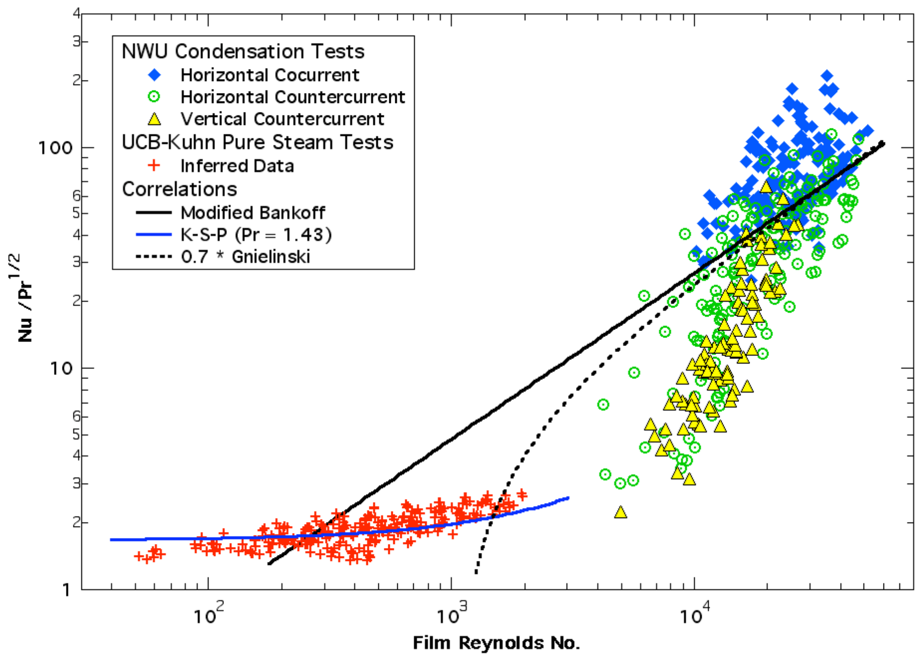
Model Description
Interfacial Heat Transfer

- **NWU Interfacial Condensation Data**
  - What about transition region?
    - Better correlation is needed.
      - Use Gnielinski modified to match Bankoff (for now).



TRACE

Model Description
Interfacial Heat Transfer

- **NWU Interfacial Condensation Data**
  - What about effect of interfacial shear on Nusselt no.?
    - Negligible effect for horizontal co-current (not shown) and for vertical counter-current flow.
    - Possible effect for horizontal counter-current flow.