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THERMAL ANALYSIS FOR THE CRYOGENIC FLUID MANAGEMENT FLIGHT EXPERIMENT (CFMFE)

BY

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PRESENTED AT

CRYOGENIC FLUID MANAGEMENT TECHNOLOGY WORKSHOP

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THERMAL ANALYSIS FOR THE CRYOGENIC FLUID MANAGEMENT FLIGHT EXPERIMENT (CFMFE)

- CRYOGENIC FLUID THERMAL ANALYSES FOR SPACE WERE INITIATED IN THE LATE FIFTIES
- TODAY, CRYOGENIC FLUIDS ARE ROUTINELY USED (OR CONSIDERED FOR USE) IN MANY SPACE APPLICATIONS
  - LAUNCH VEHICLES
  - LONG TERM STORAGE FOR PROPULSION OR LIFE SUPPORT
  - SENSOR COOLING
  - REFRIGERATION
  - SUPPLY OR RESUPPLY OF PROPULSION STAGES, SATELLITES AND SPACE STATIONS
- CFMFE OBJECTIVE - PROVIDE DATA FOR THE DESIGN OF EFFICIENT, CRYOGENIC FLUID MANAGEMENT SYSTEMS IN A SPACE ENVIRONMENT
- PURPOSES OF THIS PAPER
  - IDENTIFY RECENT CFMFE THERMAL ANALYSIS EFFORTS
  - REVIEW A SMALL PART OF THIS ANALYSIS WORK - THE PREDICTION OF FLUID AND CONTAINER TEMPERATURE GRADIENTS DURING LOW GRAVITY STORAGE IN SPACE

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CFMFE THERMAL ANALYSES

- O EARLY REPORTS
  - O MARTIN MARIETTA
  - O GENERAL DYNAMICS
  - O BEECH
- O RECENT REPORTS
  - O MARTIN MARIETTA
    - CFMF FLUID AND THERMAL ANALYSIS REPORT
    - CRYOGENIC SYSTEMS ANALYSIS MODEL (CSAM) USER'S MANUAL
  - O NASA-LERC
    - THERMODYNAMIC MODELING OF RECEIVER TANK FILLING - AYDELOTT, CHATO AND DEFELICE
    - DUMPING OF TANKS ON ORBIT - CHATO
    - SUPPLY TANK VENT CALCULATIONS - LACOVIC
  - O ANALEX CORPORATION
    - CSAM - NAKANISHI
    - PRESSURE VESSEL THERMAL DESIGN - NAKANISHI
    - MULTILAYER INSULATION - NAKANISHI
    - FLUID AND THERMAL ANALYSIS - NAKANISHI
    - TWO DIMENSIONAL FINITE ELEMENT ANALYSIS METHOD FOR THERMAL CONDUCTION PROBLEMS - NAKANISHI
    - PRESSURE VESSEL AND FLUID TEMPERATURE PREDICTIONS USING SINDA THERMAL MODELS - SMOLAK

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PRESSURE VESSEL AND FLUID TEMPERATURE PREDICTIONS USING SINDA THERMAL MODELS

- O MODEL GEOMETRY - THREE DIMENSIONAL WEDGE
  - SMALL SPHERE - 22 CUBIC FEET VOLUME
  - LARGE SPHERE - 200 CUBIC FEET VOLUME
- O MODES OF HEAT TRANSFER
  - CONDUCTION
  - CONVECTION
- O ULLAGE SURFACE SHAPES
  - SPHERICAL
  - FLAT
- O HEATING/COOLING
  - WALL HEATING - UNIFORM DISTRIBUTION
  - COOLING WITH HEAT EXCHANGERS
    - INTERNAL
    - EXTERNAL

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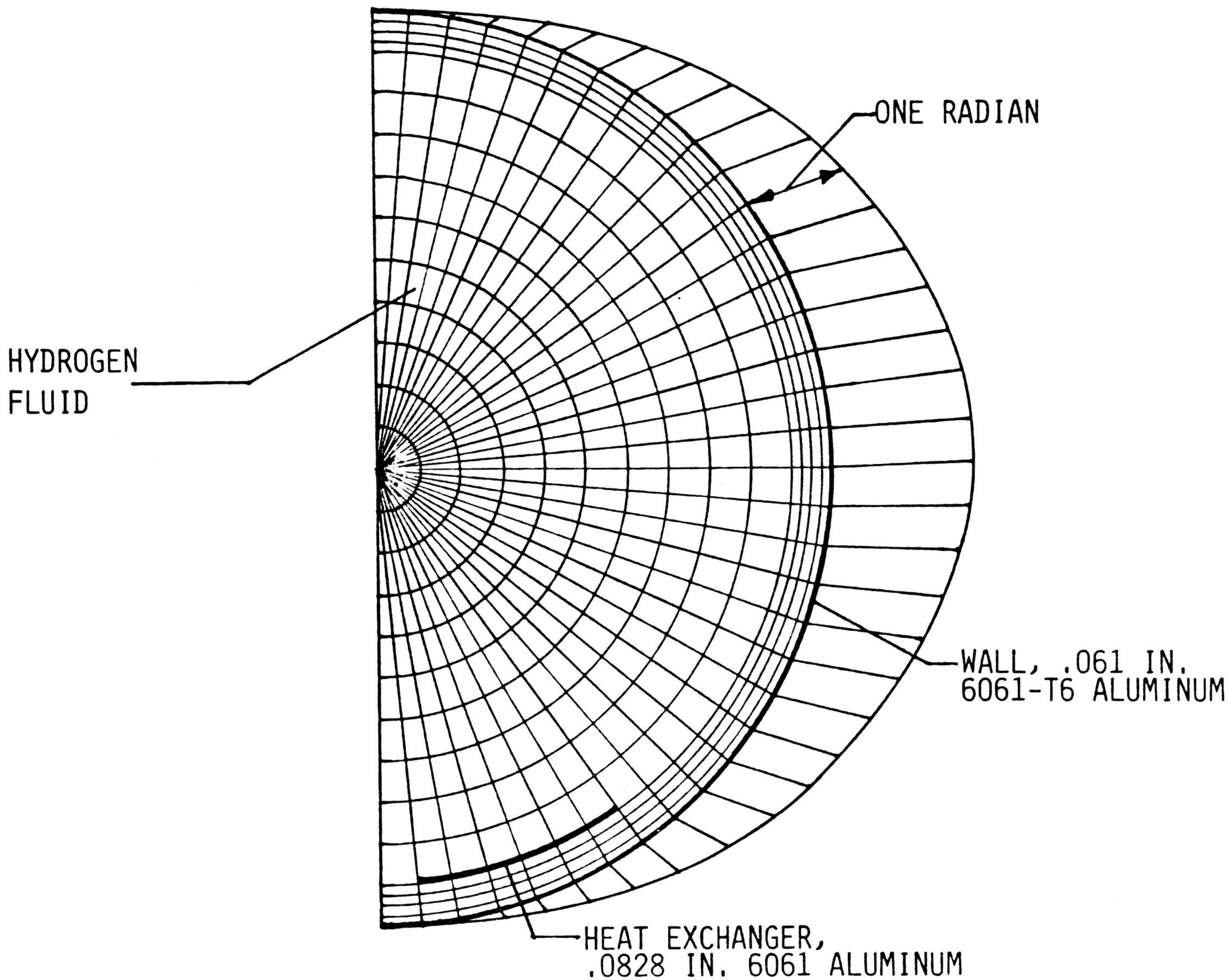
NODE LAYOUT FOR SINDA SPHERICAL

WEDGE MODEL OF CFMFE 22 CUBIC FOOT TANK



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SIMPLIFYING ASSUMPTIONS FOR SINDA WEDGE MODELS

- NO TEMPERATURE GRADIENTS NORMAL TO HEMISPHERICAL PLANE OF WEDGE
- CONDUCTION IN LIQUID, VAPOR AND WALL
- NO CONVECTION (EXCEPT AS NOTED LATER)
- NO RADIATION
- UNIFORM TEMPERATURE AT LIQUID/VAPOR INTERFACE
- THERMOPHYSICAL PROPERTIES OF FLUID AND SOLIDS TEMPERATURE DEPENDENT
- WEDGE THICKNESS - 1 RADIAN
- RADIAL HEIGHT OF NODES - 2 INCHES (EXCEPT 0.25 INCHES FOR NODES HAVING RADIUS EXCEEDING HEAT EXCHANGER RADIUS)

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STEADY STATE TEMPERATURE CONTOURS FOR A 22 CUBIC FOOT TANK

- A SPHERICAL VAPOR BUBBLE WAS ASSUMED FOR ZERO GRAVITY
- INTERNAL HEAT EXCHANGER (HEAT SINK) IS DOWNSTREAM OF LIQUID VENTED THROUGH AN EXPANSION VALVE
- UNIFORM HEATING OF WALL (0.325 BTU/HR-SQ FT)
- TEMPERATURE GRADIENT AROUND HEAT EXCHANGER IS STEEP

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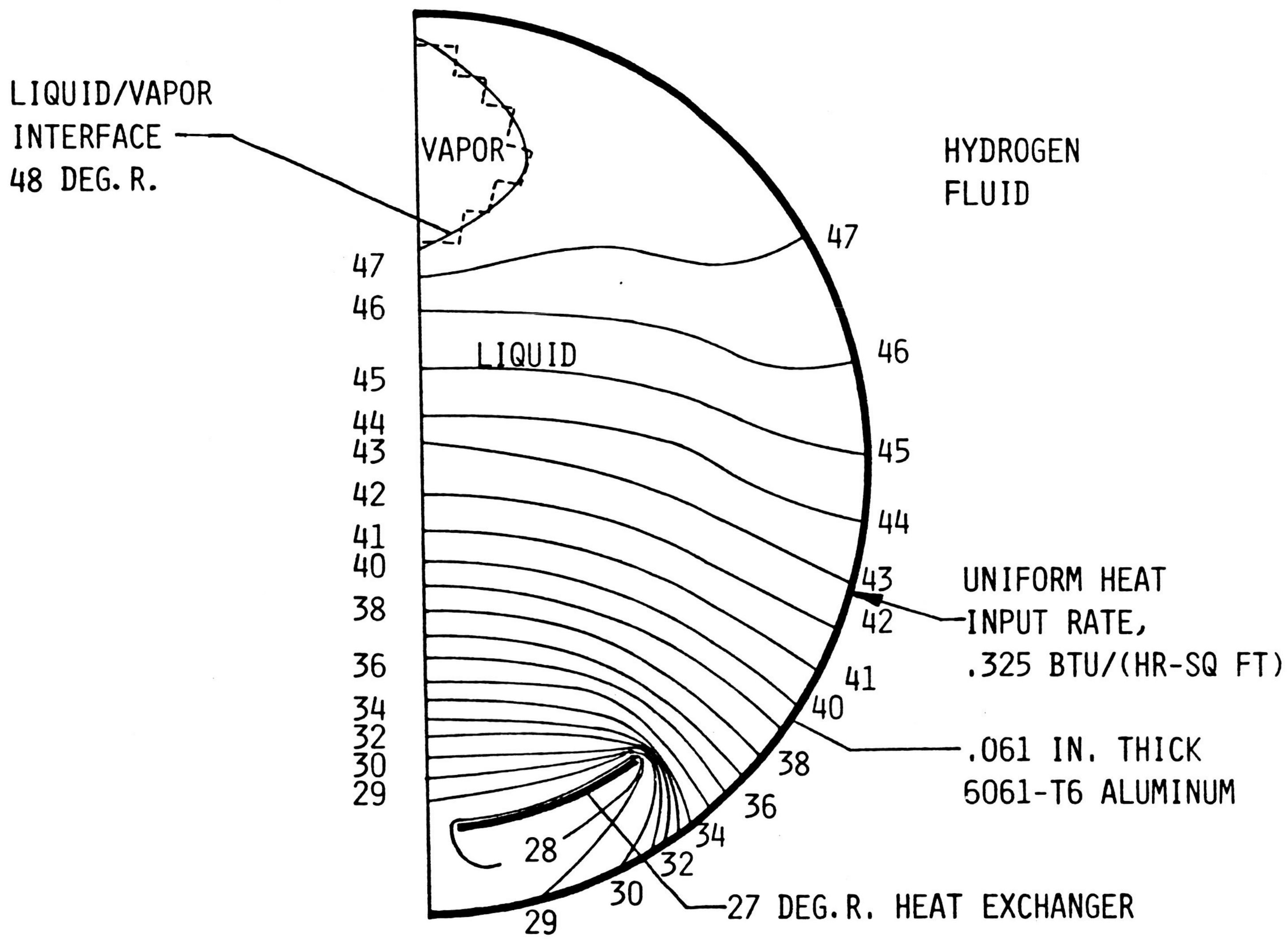
STEADY STATE TEMPERATURE
CONTOURS FOR 22 CUBIC FOOT TANK



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STEADY STATE TEMPERATURE CONTOURS FOR A 200 CUBIC FOOT TANK

- THESE ARE STEADY STATE, ZERO GRAVITY TEMPERATURE GRADIENTS FOR HYDROGEN IN A 200 CUBIC FOOT TANK WITH A WALL THICKNESS OF 0.128 INCHES

- IT APPEARS THAT IN THIS LARGER TANK CONDUCTION IS INADEQUATE RESULTING IN A LARGE TEMPERATURE RISE BETWEEN THE INTERNAL, COLD HEAT EXCHANGER AND THE LIQUID/VAPOR INTERFACE

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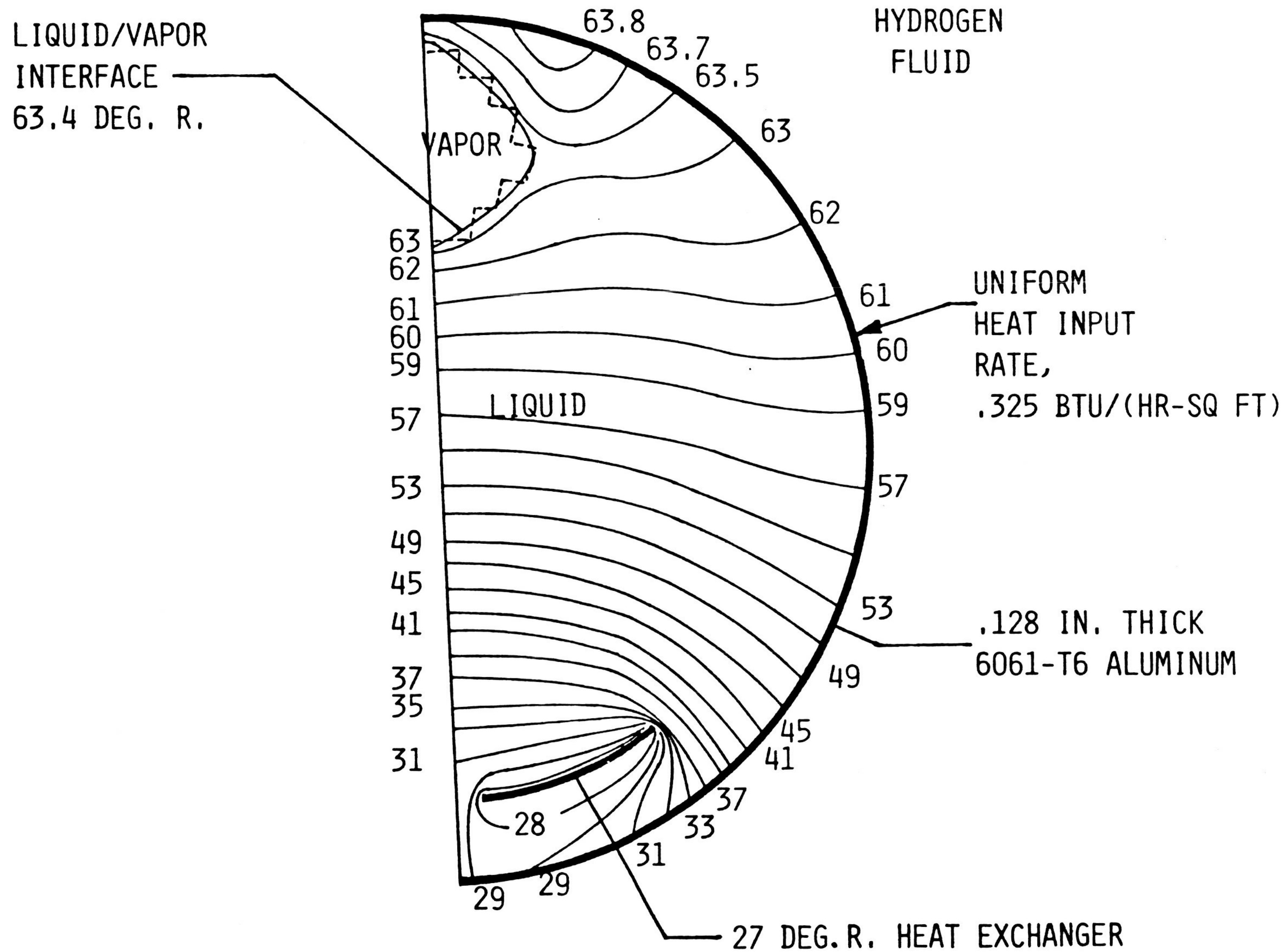
STEADY STATE TEMPERATURE
CONTOURS FOR 200 CUBIC FOOT TANK



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EFFECT OF CONVECTION ON LIQUID/VAPOR INTERFACE TEMPERATURE

- WHAT LEVEL OF CONVECTION IS NECESSARY TO HAVE A SIGNIFICANT EFFECT ON TEMPERATURE DISTRIBUTION?
- UNIFORM CONVECTION WAS ASSUMED THROUGHOUT THE FLUID AND BETWEEN THE LIQUID AND WALL
- THE CONVECTION HEAT TRANSFER COEFFICIENT BECOMES SIGNIFICANT IN THE RANGE OF 10E-04 TO 10E-03 BTU/(HR-SQ IN-DEG F) OR LARGER

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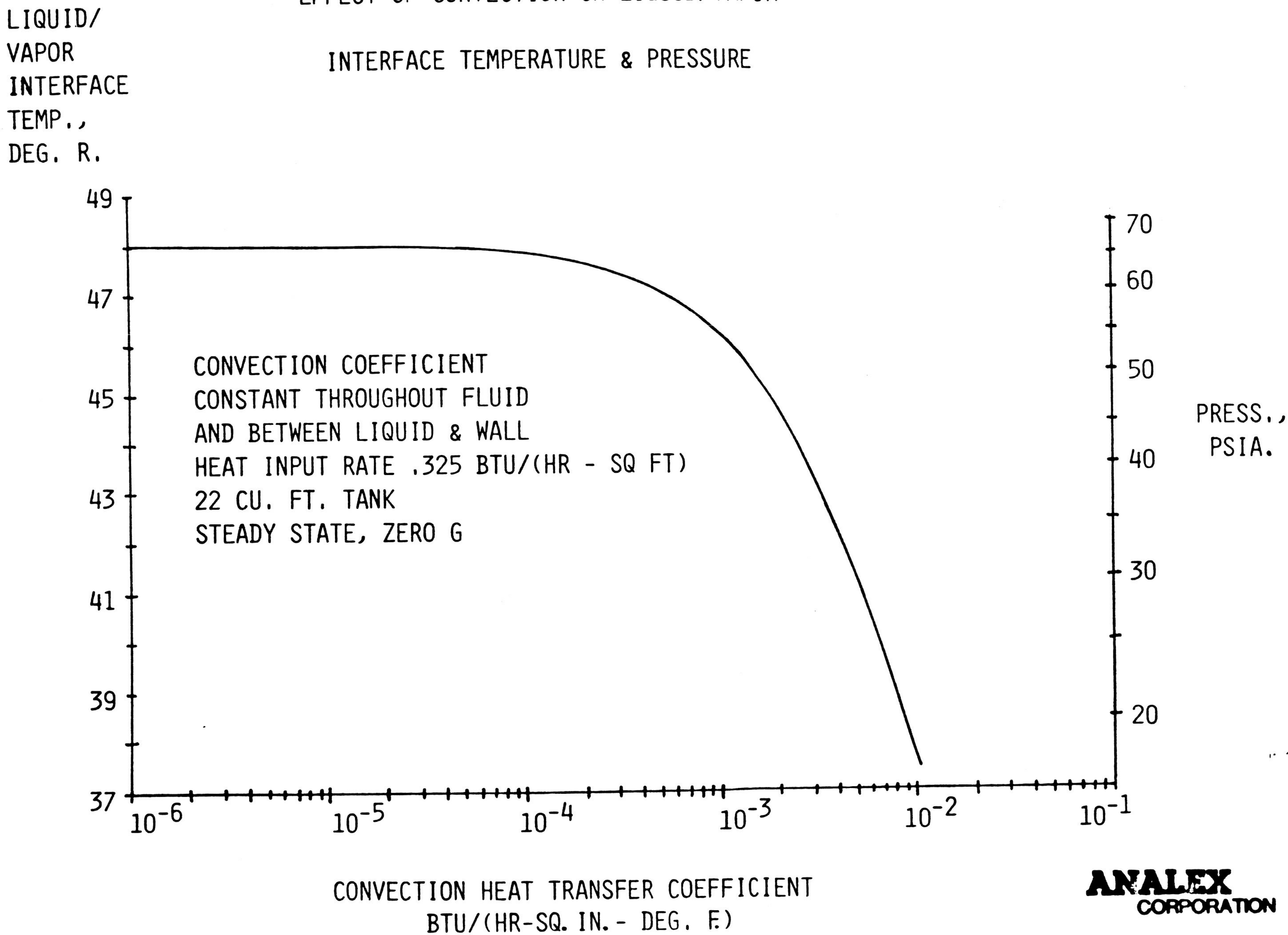
EFFECT OF CONVECTION ON LIQUID/VAPOR

INTERFACE TEMPERATURE & PRESSURE



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EFFECT OF WALL THICKNESS ON LIQUID/VAPOR INTERFACE TEMPERATURE

LARGE TANKS DEVELOP STEADY STATE TEMPERATURES AND PRESSURES MUCH HIGHER THAN SMALL TANKS FOR THE SAME WALL THICKNESS AND HEAT TRANSFER RATE THROUGH THE TANK WALL

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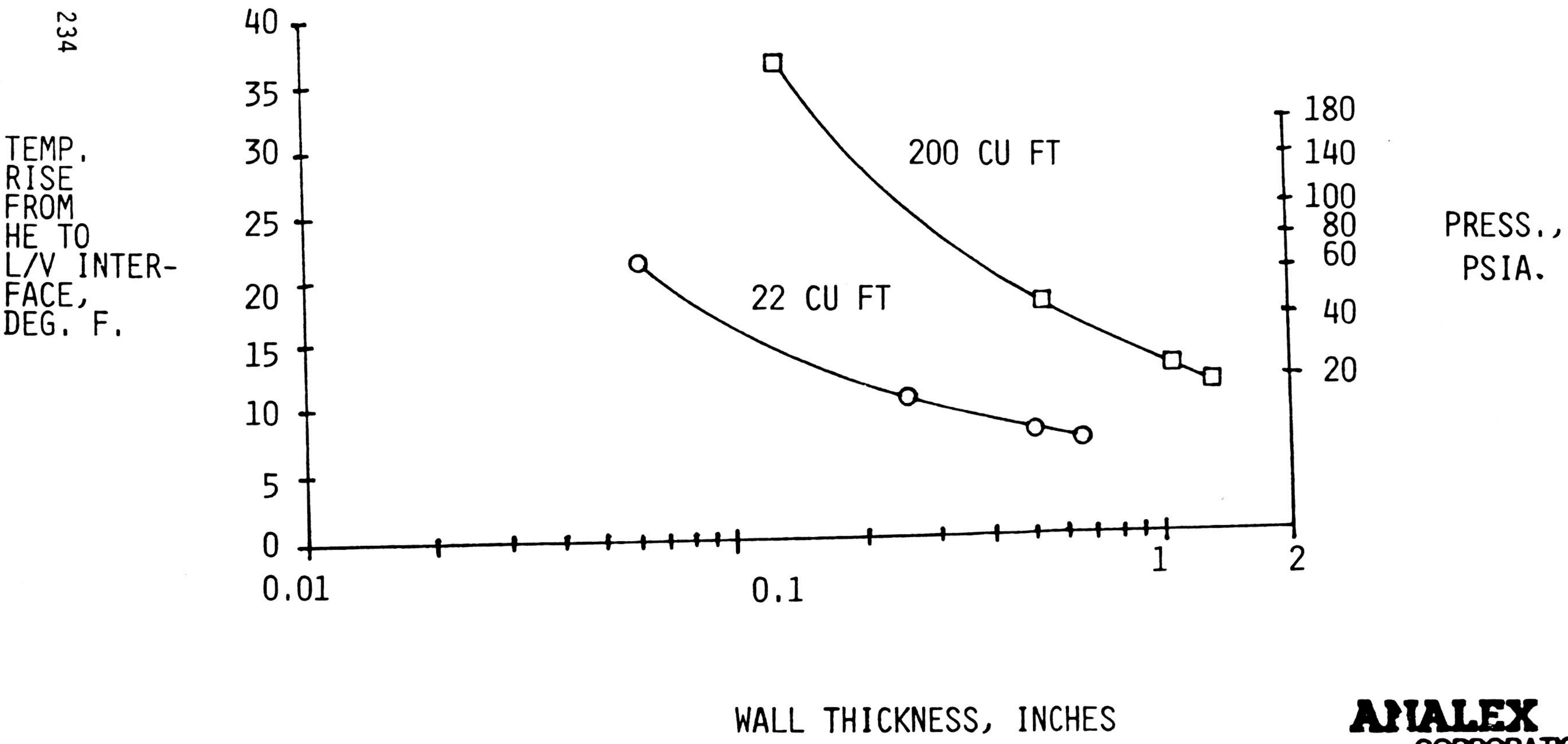
EFFECT OF WALL THICKNESS ON
LIQUID/VAPOR INTERFACE TEMPERATURE

STEADY STATE, ZERO G
NO CONVECTION
HEAT INPUT RATE .325 BTU/(HR-SQ FT)



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LIQUID/VAPOR INTERFACE TEMPERATURE VERSUS TIME

- BOTH THE SMALL AND LARGE TANKS REQUIRE HUNDREDS OF HOURS TO REACH EVEN 60 PSIA
- IN ABOUT 1000 HOURS THE SMALL TANK IS CLOSE TO EQUILIBRIUM
- THE LARGE TANK REQUIRES MANY THOUSANDS OF HOURS TO REACH EQUILIBRIUM

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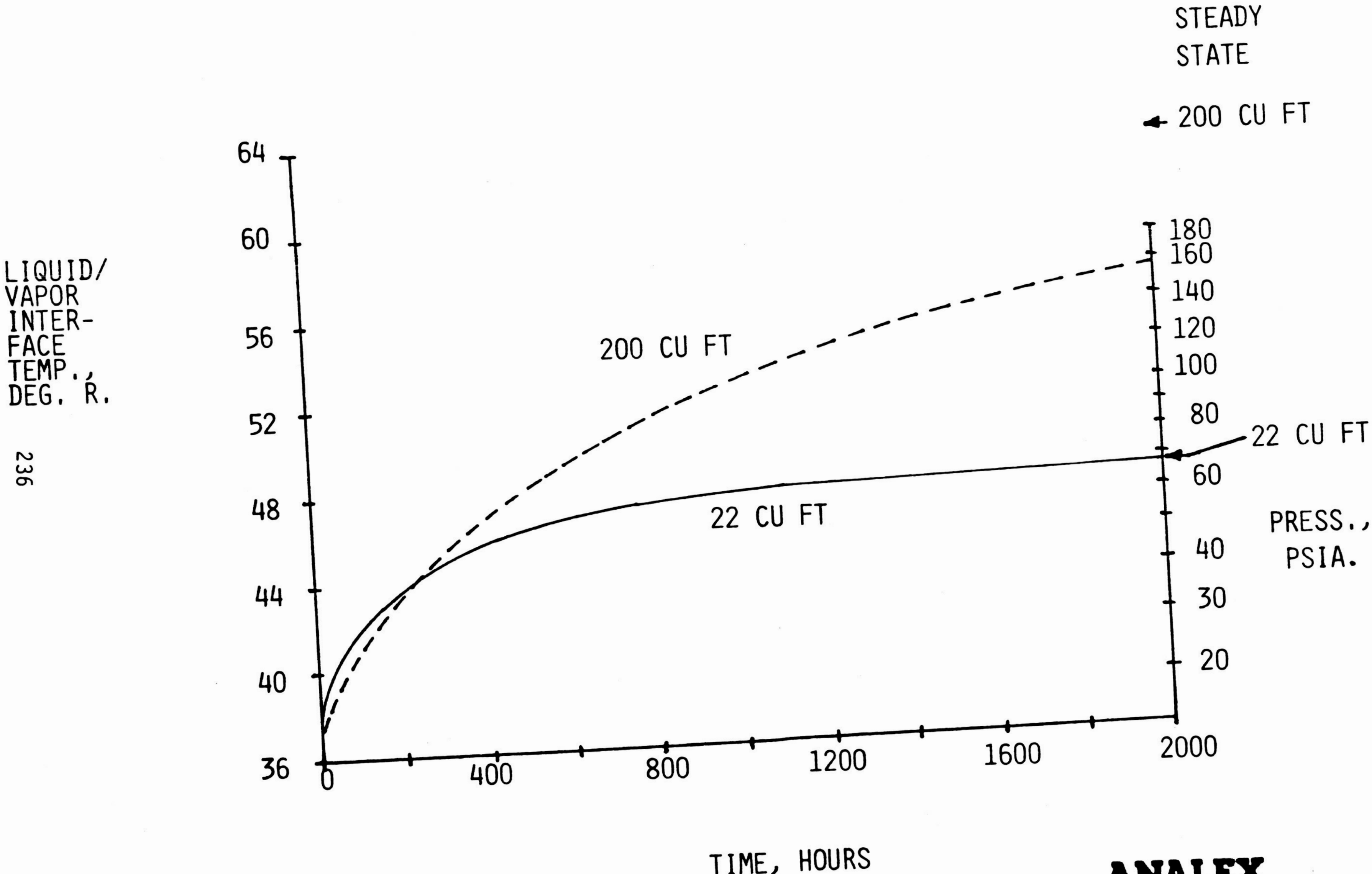
LIQUID/VAPOR INTERFACE
TEMPERATURE VERSUS TIME



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CONCLUSIONS

- CONDUCTION PROCESSES ARE VERY SLOW IN THE FLUID
- FOR SMALL TANKS, AN INTERNAL HEAT EXCHANGER IS EFFECTIVE IN CONTROLLING MAXIMUM TANK TEMPERATURE/PRESSURE
- THE FLUID CONVECTION COEFFICIENT MUST BE OF THE ORDER OF 10E-04 BTU/HR-SQ IN-DEG F OR LARGER TO CAUSE A DECREASE IN THE LIQUID/VAPOR INTERFACE TEMPERATURE
- INTERNAL CONVECTION (WITH A MIXER, FOR EXAMPLE) IS NECESSARY TO KEEP TANK PRESSURE LOW, ESPECIALLY IN LARGE VOLUME TANKS

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SPEAKER: GEORGE R. SMOLAK/ANALEX CORP. (LEWIS RESEARCH CENTER)

John Schuster/General Dynamics Space Systems:

The considerations on thermal mixing that you've done have been primarily limited to tanks nearly filled with liquid with a relatively small vapor ullage. Have you studied the opposite situation where you have mainly vapor in the tank and you are concerned about how that vapor will equilibrate with the liquid, and what the evaporation rate might be out of the liquid?

Smolak:

No, I didn't go into that. The model assumes that there is a constant temperature at the liquid vapor interface. The model at this point does not include evaporation at the liquid vapor interface, that part of the thermodynamics is not in there. The conduction processes and convection processes in the vapor were included when I did include convection. That part of it is rigorous, but that is as far as the thermodynamics went. It is not truly a complete thermodynamic model by any means.

A. A. Sonin/Massachusetts Institute of Technology:

I wonder if you could elaborate on how you applied the convective heat transfer coefficient, and what the boundary conditions were on your analysis when you did that?

Smolak:

I said that I had a heat transfer coefficient of the values that were stated. The area over which that convective coefficient was active was the area between adjacent nodes in the SINDA model and the appropriate temperature difference was the temperature difference between the adjacent nodes. In other words, it was put in as a HA-Delta-T type effect. It was not done with any directionality or preforentially; in other words, it was done between all adjacent surfaces. If there is better information on how convection acts in reduced gravity, I would be happy to put it into the model, but I couldn't find anything better in the literature.

Robert Rudlin/Martin Marietta Denver Aerospace:

I want to ask a question, but I observed one thing in your analysis which I think was very good. First, you do have to know what the convection coefficient or the Nusselt number is. I didn't pick up what the Nusselt number was, but I did pick up that your spacial dimensions from the free surface down to the heat exchanger at 27 degrees, which I guess is about 3 feet, is probably your biggest resistor, or the primary conductor. If I draw these conclusions correctly, it seems to me what your saying is when you doubled everything you basically doubled that 3 feet to 6 feet, and, sure enough, your temperature difference went from 30 degrees to 60 degrees. That's pretty much like you'd expect with a resistor/conductor model. Am I drawing the right conclusions in saying that what you're really telling me is that you have to have your heat exchanger on the wall and uniformly distributed through the fluid in order to keep that basic dimension significantly less than 3 feet if you want to have a fairly well mixed fluid? In other words, don't stir it up; just get your resistance path as small as possible, if you really want to control the temperature.

Smolak:

That's a long question. I would say yes, your supposition is pretty much valid. The scaling was done from the small tank to the larger tank and the results were not entirely unexpected. We did some more work that I didn't mention that had to do with placing a heat exchanger on what amounted to the north pole in the model, which was up near the vapor bubble. That was rather effective in helping to destroy these temperature gradients in the fluid. Yes, you are on the right track. We have not explored really extensively the optimum positions for those heat exchangers. What we had in mind was, since most of the tank penetration was toward the top of the tank, that we'd intercept the heat coming through those penetrations with a low temperature fluid from the visco jet or thermodynamic vent system and that seemed to be rather effective.