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**USING THE EPRI TEST DATA TO VERIFY A MORE ACCURATE
METHOD OF PREDICTING COOLING TOWER PERFORMANCE**

by

RICHARD J. DESJARDINS

The studies and conclusions reported in this paper are the results of the author's own work. The paper has been presented before, and reviewed by the Cooling Tower Institute, and approved as a valuable contribution to cooling tower literature.

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INTRODUCTION

It has been known for years that the official CTI method of thermal performance analysis¹¹ is insufficiently accurate for large industrial and power projects where the cold water temperatures can be evaluated in the area of many thousands of dollars per degree. Many papers have mentioned the need for "hot water correction factors" ranging from 2 to 5 percent for each 10 degree increase in hot water temperature above 100 degrees F.

In the title of his 1984 CTI paper⁹ Marcel Lefevre asked if eliminating the Merkel Theory approximations could "Replace the Empirical Temperature Correction Factor". He did an excellent review of the Merkel¹ theory showing where changes are needed and the order of magnitude of the errors introduced by the simplifications. However, we now have data which confirms even greater improvements in the correlations are possible.

In 1989 EPRI¹⁰ published the results of their testing at Houston Lighting & Power Co., and the analysis of the tests can be used to show that while adjusting for the simplifying assumptions listed in Mr. Lefevre's paper are helpful, they alone cannot fully account for the scatter of the data points. Further changes are needed, and other authors have shown us how to do it, even though the industry has, up to now, generally ignored their work. It is time for a change, and this paper will use the EPRI raw test data to demonstrate the "why?" and "how?" of doing it. With the availability of modern computers there is no real reason not to correct the errors.

WHY SHOULD WE CARE?

The errors introduced by the simplified CTI methods almost always produce optimistic results, and some of them can be signifi-

cant. Cooling tower operating performance and price are almost always directly proportional to each other. i.e., a five percent reduction in performance is equivalent to a five percent reduction in price, and when you consider most companies are lucky to make five percent net profit on their sales and jobs are won or lost on price differences of less than one percent, it is obviously necessary to add a little accuracy to the calculations. It affects the manufacturer, the contractor and the user.

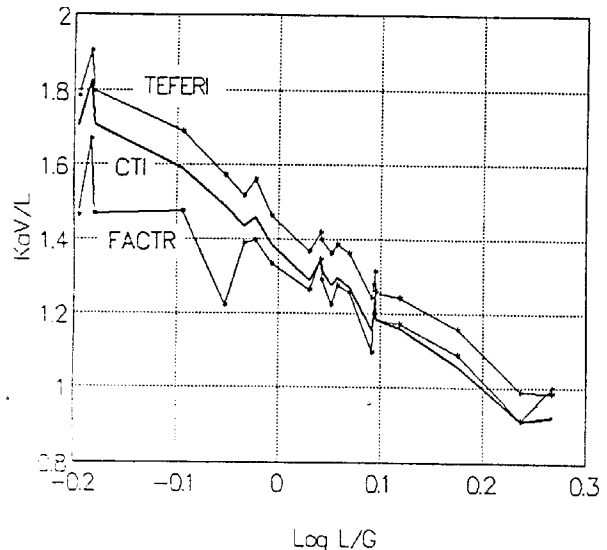


FIGURE 1. PLOT FROM EPRI REPORT RESULTS: MARLEY MC67 FILL, 3 FT. FILL HEIGHT

The data in the EPRI report can be used to graphically see the scatter produced by several methods of analysis. Figure 1. shows three plots of data; (1) CTI method of calculation, (2) TEFERI¹⁰ program calculations, and (3) FACTR¹⁰ program calculations. Please note the FACTR program actually increased the scatter of the data. The ideal theory would eliminate all scatter. Ideally it would be possible to draw a smooth curve through the plot of all test points. (This does not necessarily mean a straight line on log-log paper, just a smooth curve). With the scatter shown by these data plots, it would

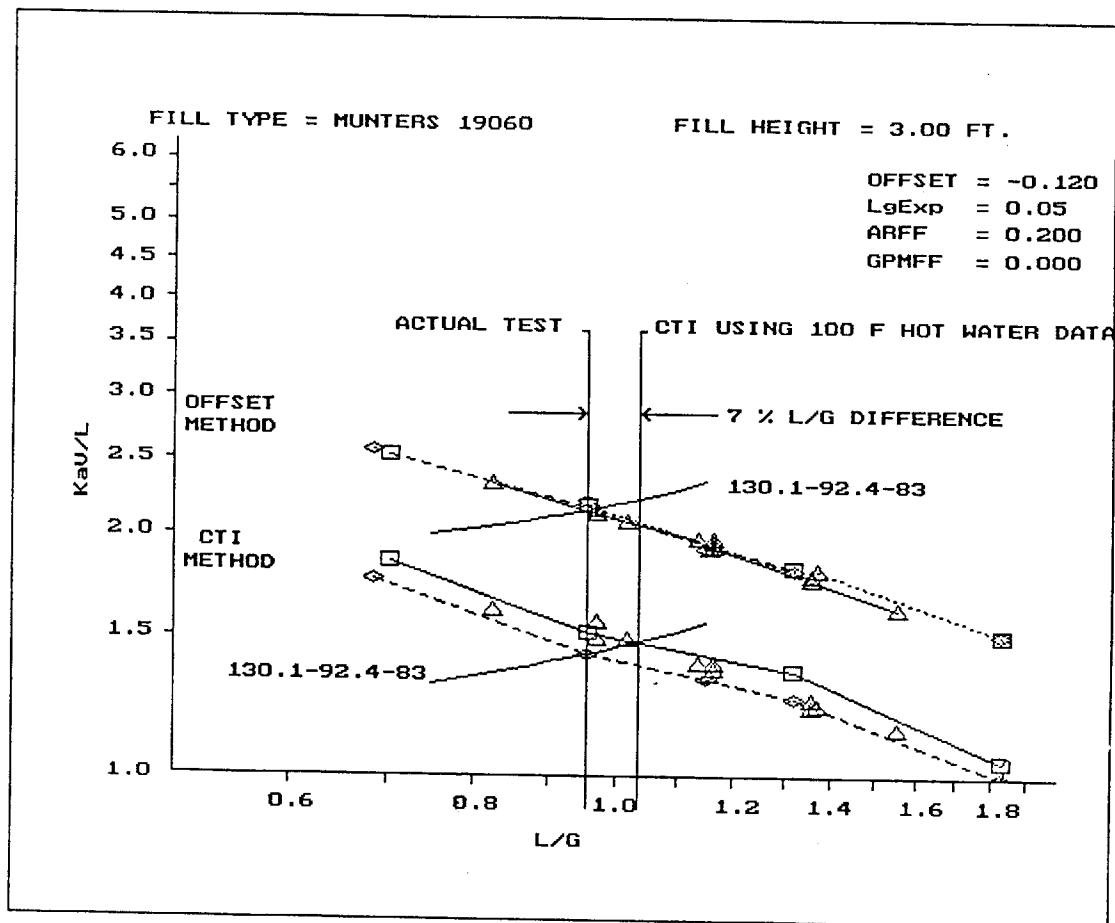


FIGURE 2. CTI SYSTEMS REQUIRES TEMPERATURE CORRECTION FACTOR

be easy to conclude there was considerable question of the accuracy of the EPRI tests. Quite to the contrary, the analysis to follow will show the data were likely very good.

Most testing of tower fills has been at hot water temperatures of about 100° F, but many refineries, chemical plants and power plants operate with much higher hot water temperatures. Fortunately, the EPRI tests included a very good mix of temperatures, and the data plotted in Figure #2 are typical of the processed results. The CTI calculations show a wide separation of the tests points run at about 100° F and those run at about 130° F.

If testing was done only at approximately 100° F hot water, like the testing for CTI Paper TP-88-05,¹³ the test data would be plotted as shown by the square $\square$ points in the CTI METHOD plot of Figure #2. If a demand line is calculated for the test temperatures of 130.1° HW - 92.4° CW - 83° WB, the intersection results in a required L/G of 1.02. However, please note if line is plotted through the diamond $\diamond$ test points run at approximately 130° F hot water is used the

L/G is 0.95, and the tower designed using the lower temperature line would be deficient by seven percent (7%). If a properly fit curve is run through all the points, the error could be cut in half, but this is still an unacceptable deviation for most projects and contracts. If only low temperature tests are available, the error is not reduced.

WHAT ARE THE MOST RECOGNIZED ERRORS INTRODUCED BY USING THE SIMPLIFIED MERKEL THEORY?

In his 1984 CTI paper Marcel Lefevre listed the following problem areas:

1. Influence of water and air temperature, barometric pressure, and relative humidity on the driving force potential between the water and the air.
2. Influence of the Lewis Number.
3. Errors on heat balance and on integration due to neglecting evaporation and the combined heat balance and integration errors.
4. Errors due to the Tchebycheff 4-point integration method.

data to predict performance. Including the Lewis Number in the calculations did eliminate some of these corrections. Appendix Figures #1 and #2 show results with and without the Air Rate Fill Factor correction.

The plots shown in the Appendix are for both the CTI and "offset" methods with all other refinements included. It will be noted the lower temperature $\square$, medium temperature Δ , and the high temperature $\diamond$ test points are brought closely together with the use of the hot water "offset". This dramatic change cannot be obtained by adjustments to the Lewis Number as suggested by some of my colleagues on the CTI "Non-Merkel" Committee.

This is not a phenomenon of the EPRI test data. Baker and Shyrook demonstrated that it works for cross flow. Also, in 1955 the J. F. Pritchard Co. of Kansas City filed counter flow test data with the Library of Congress.⁸ The Pritchard data listed three fills with eight different hot water temperatures between 100° F and 140° F and six test points at 120° F. Appendix Curves #22, #23 and #24 show an analysis of the data using both the CTI and "offset" methods of analysis. Note the test points have been brought to a reasonable curve with the "offset" system. These tests were for counter flow, *splash type* fills. The "offset" values were closer to those obtained by Baker and Shyrook for *splash type*, cross flow fills. An "offset" value of -0.1 appears to be a best-fit. The Pritchard data do not contain enough test points to determine the variation of "offset" with L/G.

The best-fit "offsets" for the film type counter flow fills tested by EPRI were considerably higher. However, it is noted the 3' fill height best-fit "offset" value for Munters and Brentwood 19mm fills was considerably less than the best-fit values for taller fill heights. Also, Marley and Brentwood 1900 taller fill heights could use as much as -0.3 "offset". Minor variations in the "offset" value do not make significant fluctuations in the results, and an offset value of -0.193 appears to be a compromise best-fit for all film fills listed in the EPRI report. Actually, there is no real reason the "offset" needs to be a fixed value at all fill heights for each fill type. With use of a computer for analysis, the demand curve analysis could be adjusted for fill type and fill height equations based on varying "offset" values.

The "offset" consistently appears to vary as a function of $(L/G)^{0.5}$.

It should be noted the method of testing by EPRI resulted in absolute test data which might not realistically represent the operations of the various fills in actual application. The stated objectives "to develop reliable data on mass and heat transfer coefficients that would serve as input to predictive methods for evaluating cooling tower performance," and "to test these methods (and revise them as necessary) for their ability to use laboratory data (obtained under uniform inlet conditions) for predicting prototypic behavior under realistic (possibly nonuniform) operating conditions" led them to use only the Marley nozzle for all tests. When properly spaced the Marley nozzle can provide better water distribution than other less sophisticated nozzle designs. There is a real possibility the same fills with other nozzles will not perform as well due to less uniform water distribution. Likewise, the testing on the Marley MC67 film fill appears to have been negatively affected by testing with non-optimum distribution nozzle spacing, or there may have been excessive unaccounted wall water by-pass because Marley did not require end caps or sealing. Even with the Marley nozzle it appears there is some loss of performance due to light water loading as evidenced by the curvature of the characteristic curve. (Again, nobody ever said it *had* to be a straight line on log-log paper). It would be helpful if EPRI could run a few more tests with other nozzles.

Secondly, the rain zone below the fill was eliminated in the EPRI tests by their method of collecting and measuring the cold water temperature. Again, this was stated by EPRI to have been intentional.

Fill, spray system, and rain zone cooling go together just as certainly as fan, fill, spray system, eliminators, louvers, and plenum sizes. A cooling tower operates as a complete system. It is sufficient to correlate the data so it can be usefully repeated and applied. If this can be done, there is no need to isolate the mass transfer coefficient from the heat transfer coefficient.

CONCLUSIONS:

It is interesting to note there is little difference in the KaV/L per foot of fill height at a given L/G for all the different counter

flow, film fills tested. This, alone, does not mean there will not be differences in overall tower designs, because pressured drops and nozzles differences will affect the overall system operation, but it does indicate there is no magical new physics available to the entrepreneur.

The purpose of this paper is not to verify the absolute value of the data collected, but rather the method of analysis and use of the data. The high precision of the testing by EPRI has allowed this to be done.

The improvement in the data correlation using the "offset" system is evident and significant. R^2 statistical data analysis shows correlations to a third order equation best-fit curve of 0.985 or better for almost all of the EPRI counter flow test points, and for some data sets it is 0.995 or better. With this degree of accuracy available for theoretical thermal performance prediction it is now up to fan, fill, and nozzle manufacturers, to make practical application data available to the users. Clearly, other methods have not provided this quality of data analysis and certainty of prediction. Methods of analysis which increase the scatter of the data points serve no useful purpose. Use of the "offset" system reduces the scatter of the data points and eliminates the need for empirical hot water correction factors.

Performance predictions without an adjustment for the difference in temperature between the bulk water and the film surrounding the water drop almost always produce optimistic results for temperatures greater than 100° F. Performance predictions and check rating with curves or a data base developed by testing only at approximately 100° F hot water temperatures should be used with caution.

Although new parameters have been added, and empirical adjustments for specific fill and spray combinations may still be required, with the use of computers the minor modifications to standard Merkel analysis are simple, reasonable, and practical to implement. The "offset" system has been available for years. It was simply ignored. It is time to make the change.

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ABOUT THE AUTHOR

Richard J. DesJardins is a Cooling Tower Consultant and Manufacturer's Representative for several heat transfer equipment companies. He received his BSME from the University of Colorado in 1959 and MBA from the University of Missouri at Kansas City in 1965.

Prior to formation of his own company in 1984 he worked twenty-five years with the Marley Co. and its subsidiaries.

APPENDIX A

LEGEND TO APPENDIX CURVES:

HOT WATER TEMPERATURE (F)

GREATER THAN 117° DAIMOND $\diamond$

105° to 117° TRIANGLE Δ

LESS THAN 105° SQUARE $\square$

AIR RATE LINES (UPPER CURVE)

GREATER THAN 45 LB. DRY AIR/MIN/SQ FT -----

40 TO 45 LB. DRY AIR/MIN/SQ FT _____

LESS THAN 40 LB. DRY AIR/MIN/SQ FT

FLOW RATE LINES (LOWER CURVE)

GREATER THAN 6.0 GPM/SQ. FT. -----

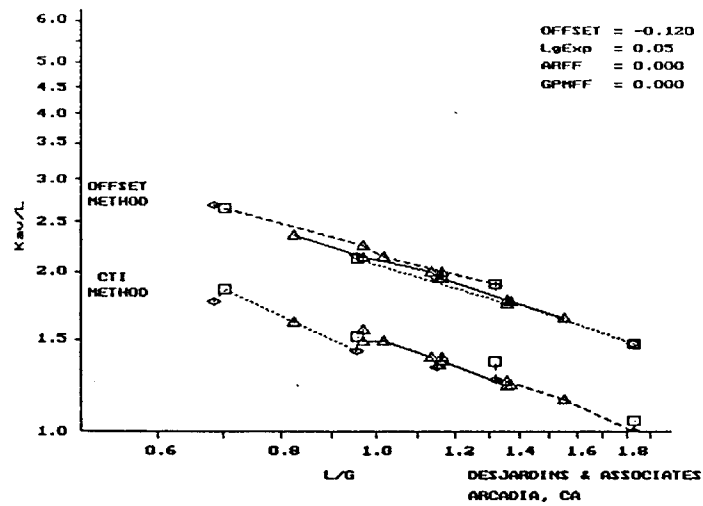
4.2 TO 6 GPM/SQ. FT. _____

LESS THAN 4.2 GPM/SQ FT

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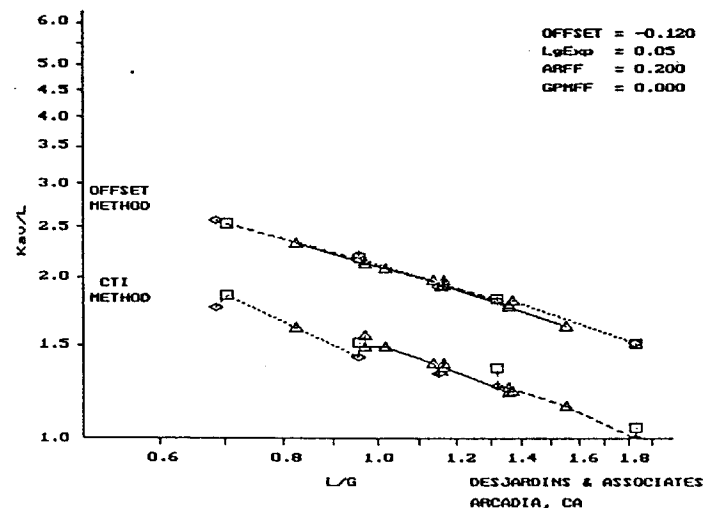
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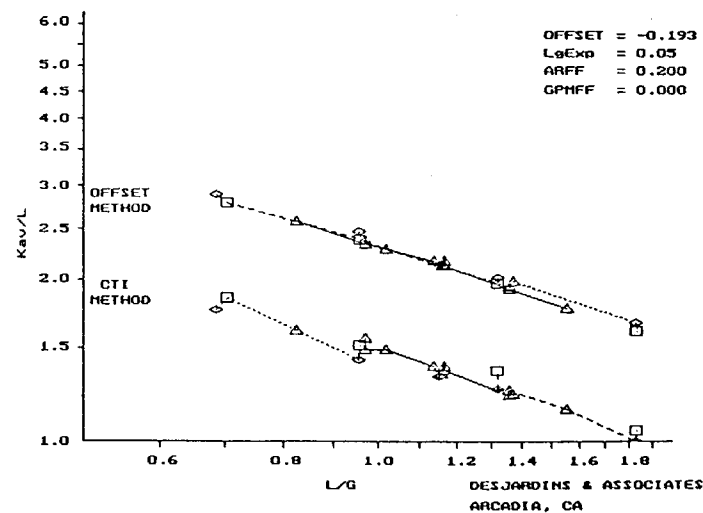
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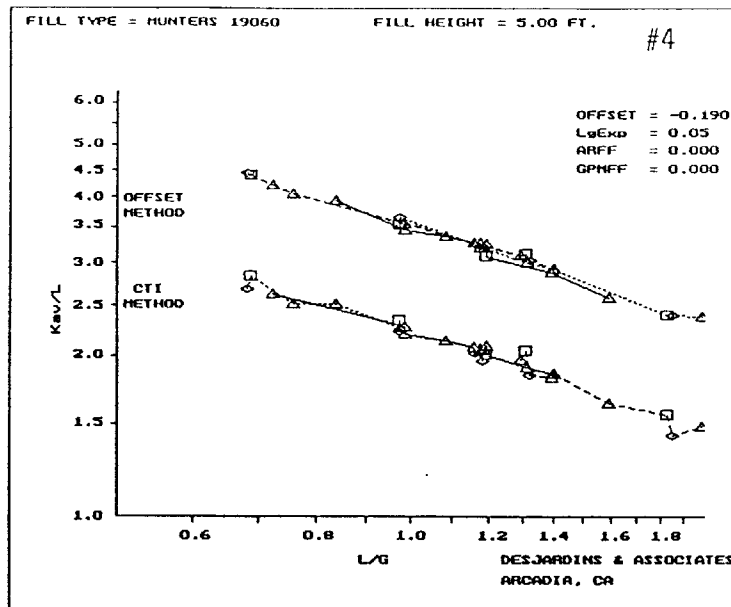
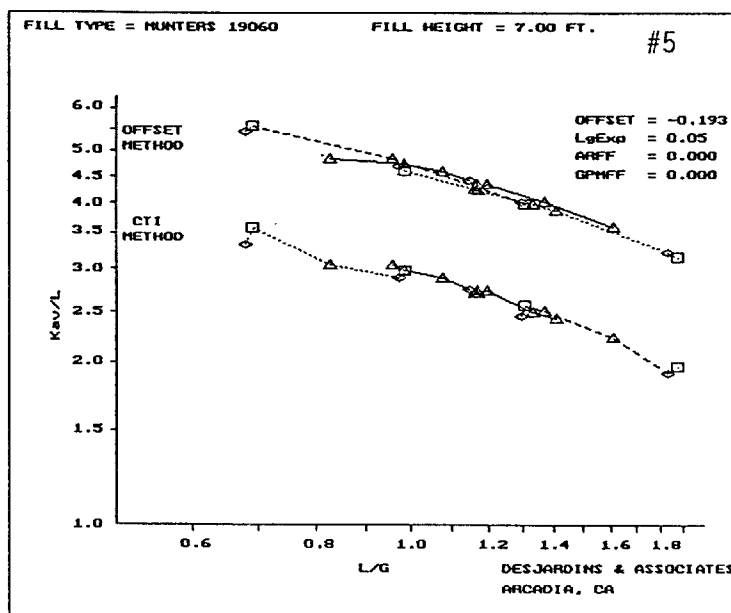
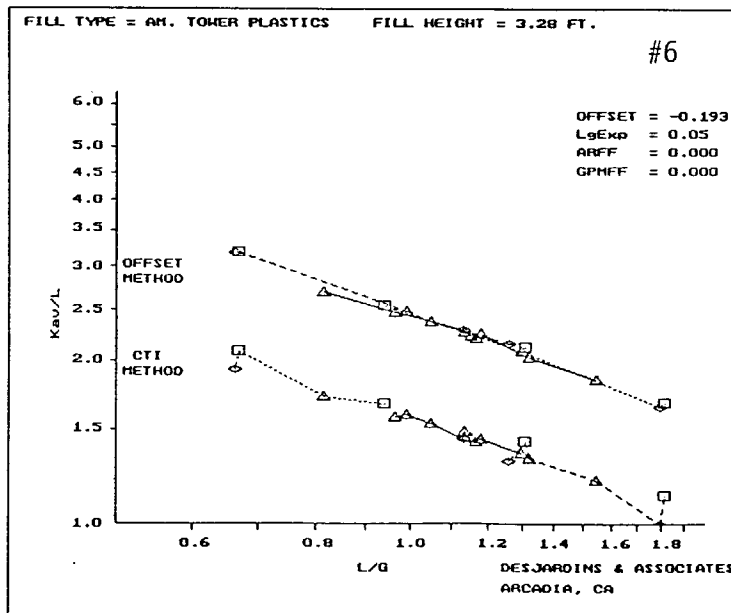


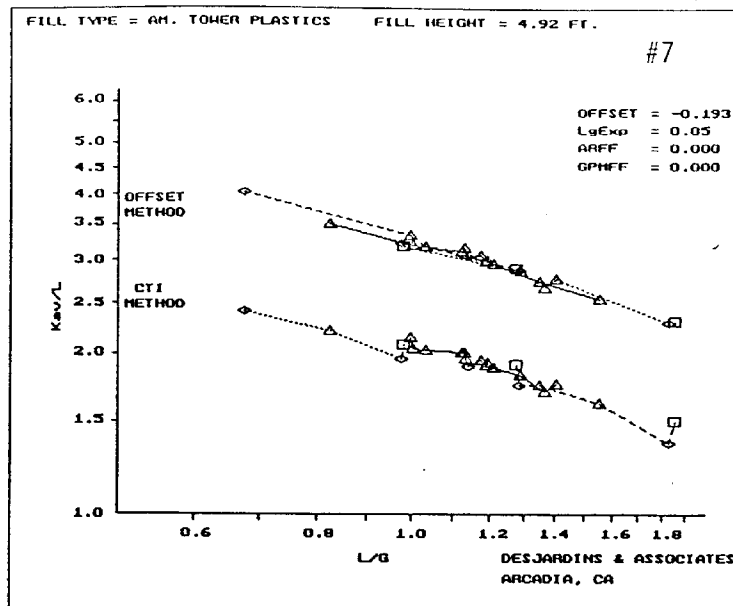
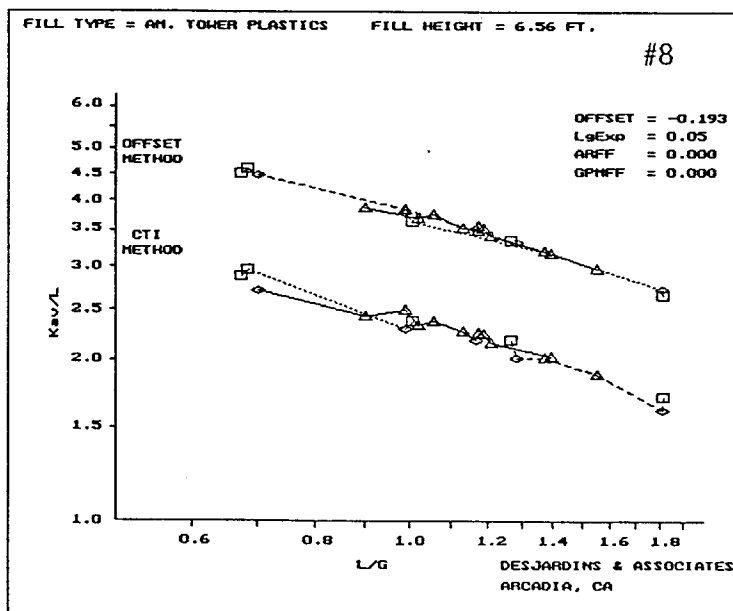
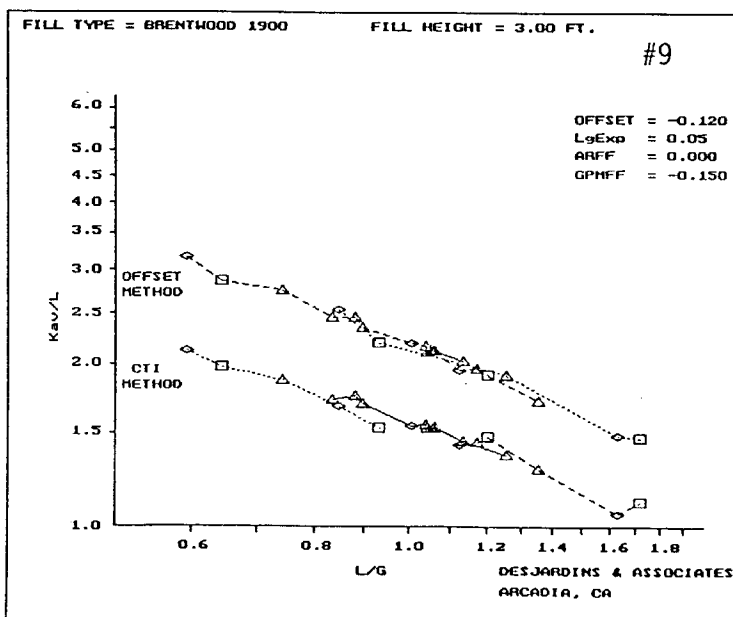
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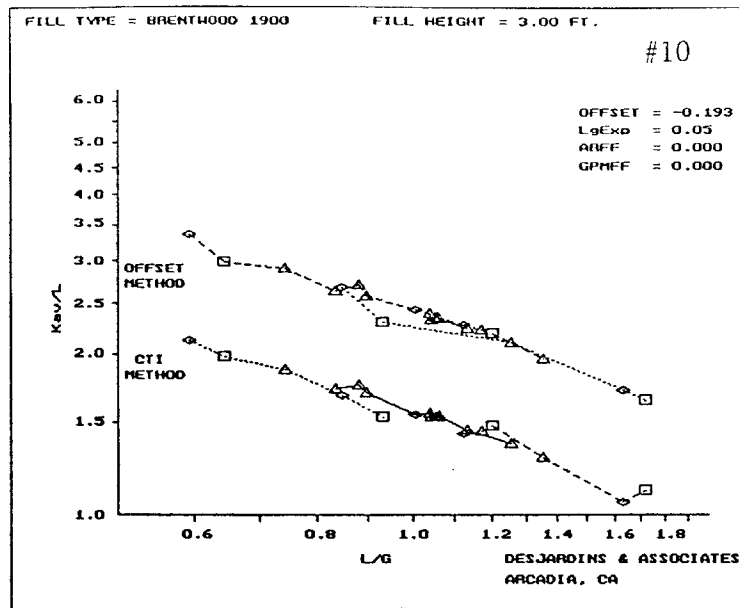
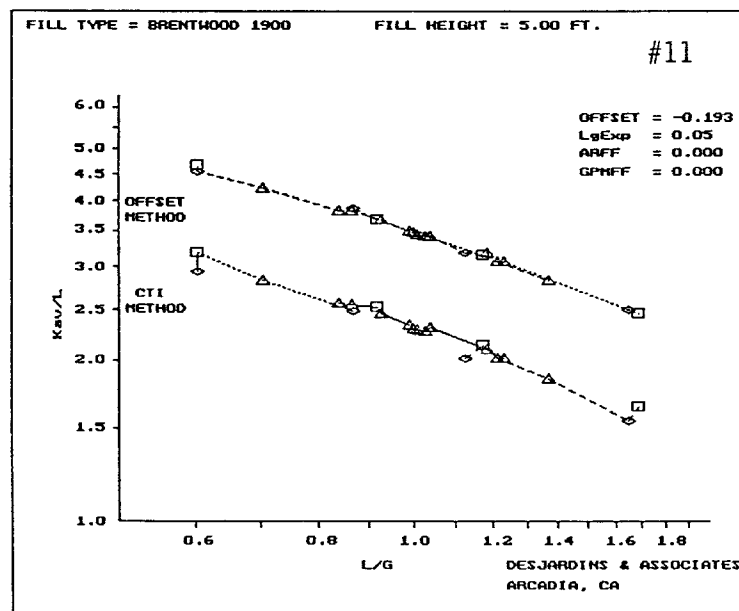
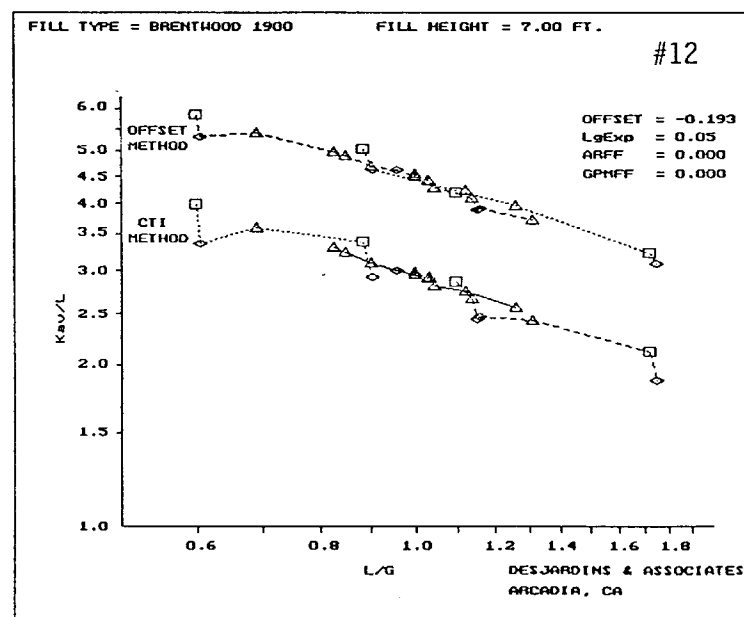
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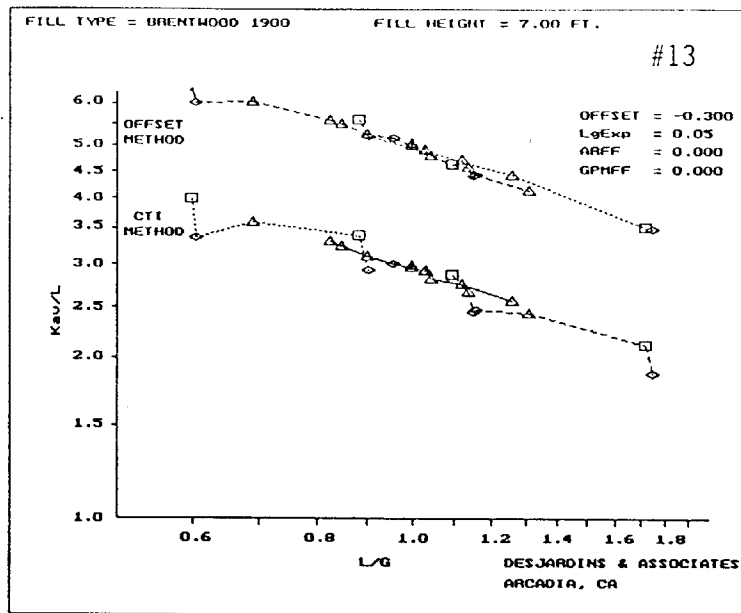
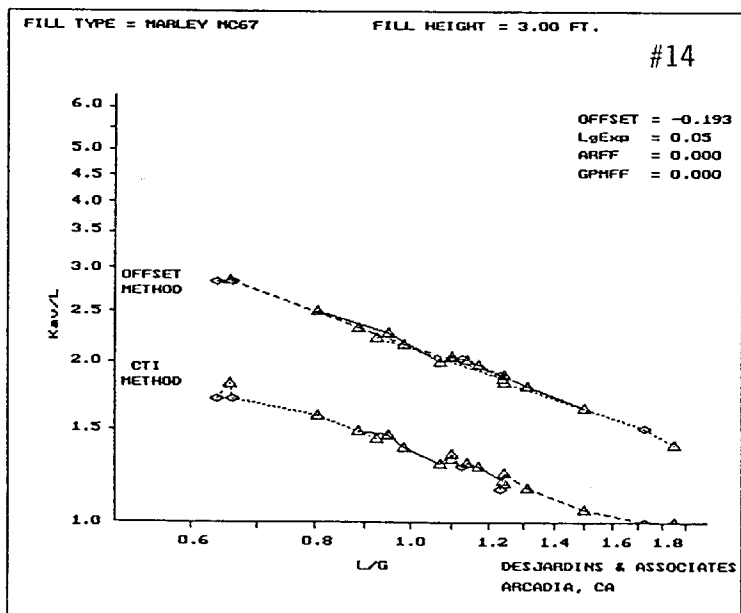
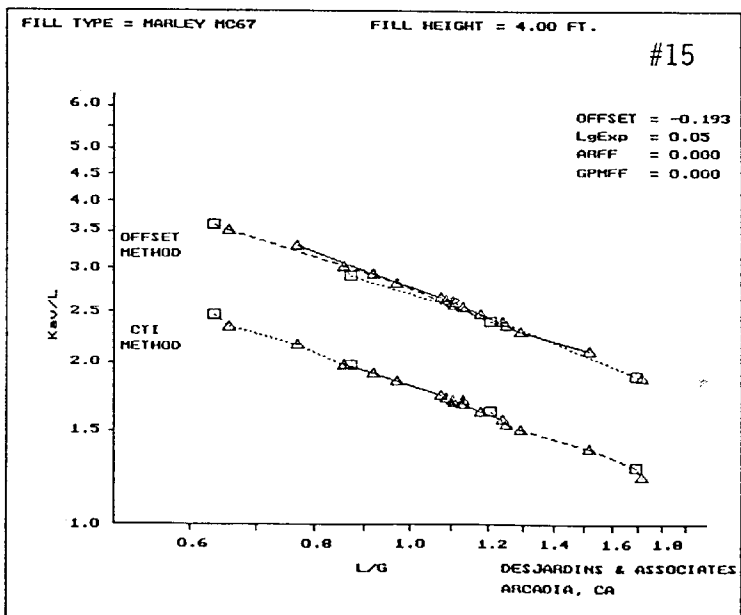
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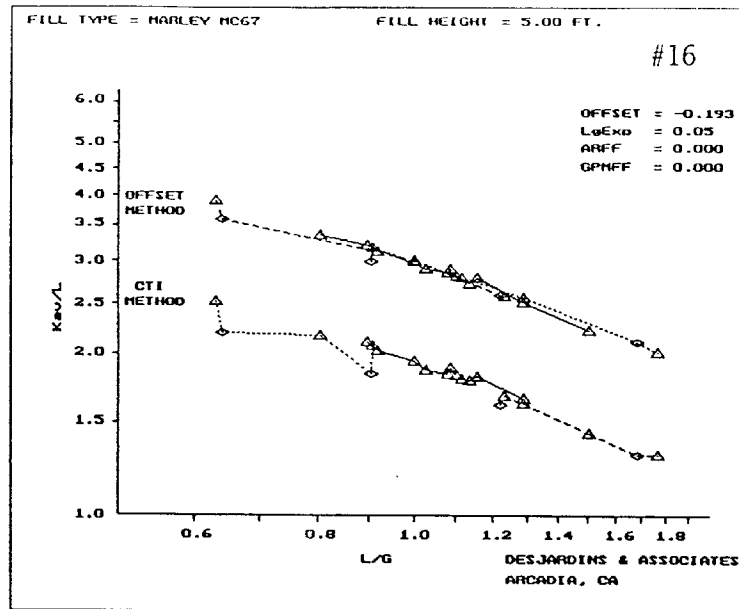
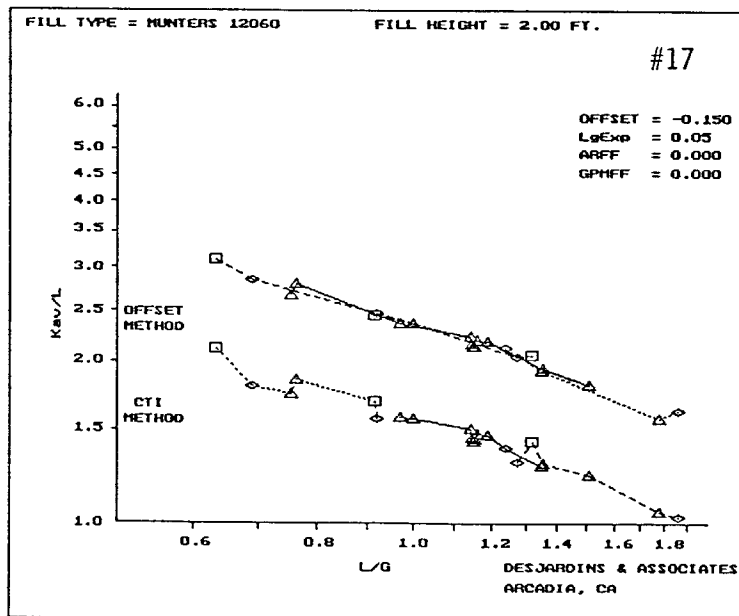
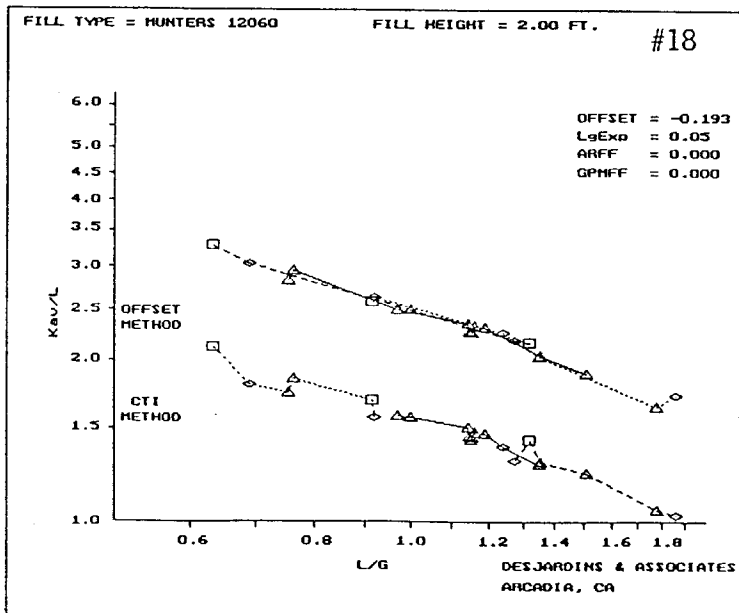


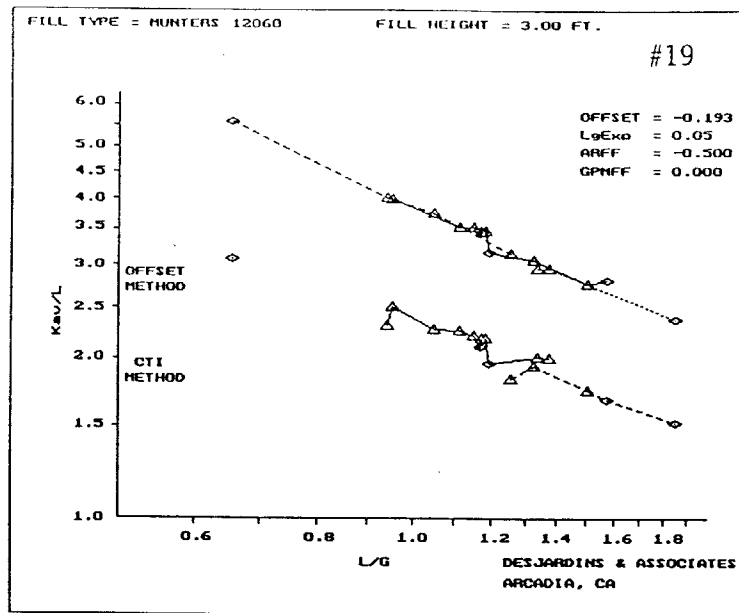
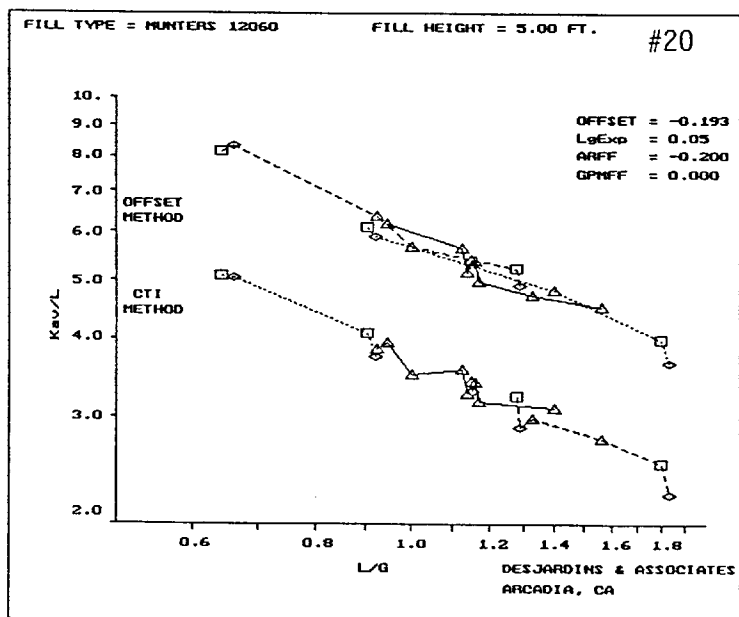
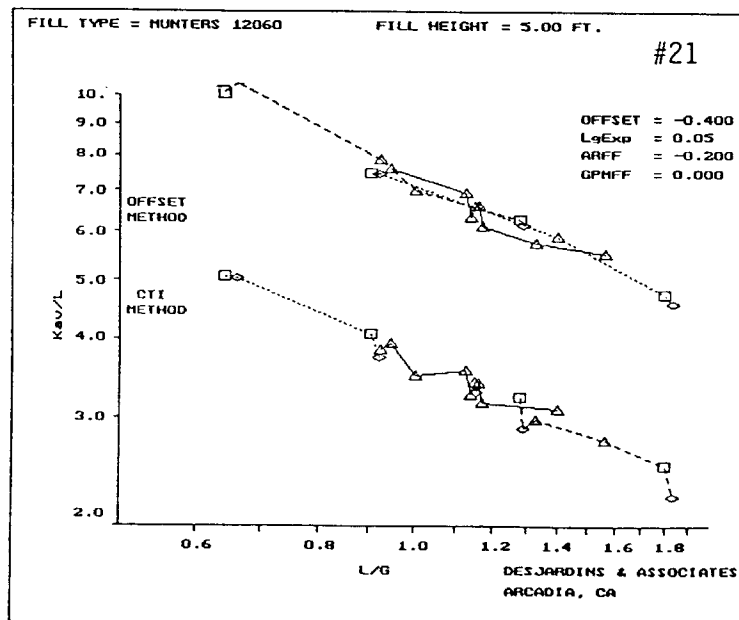








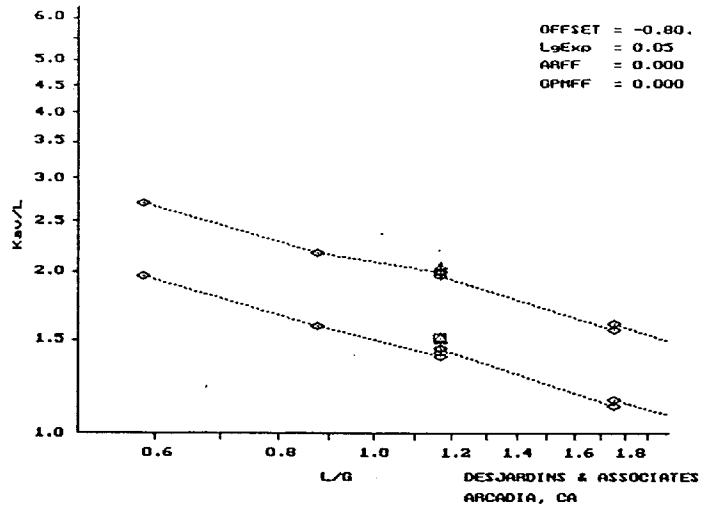




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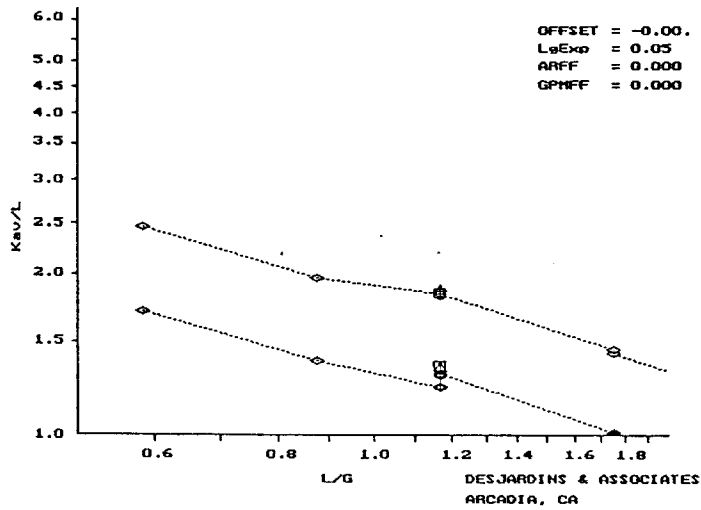
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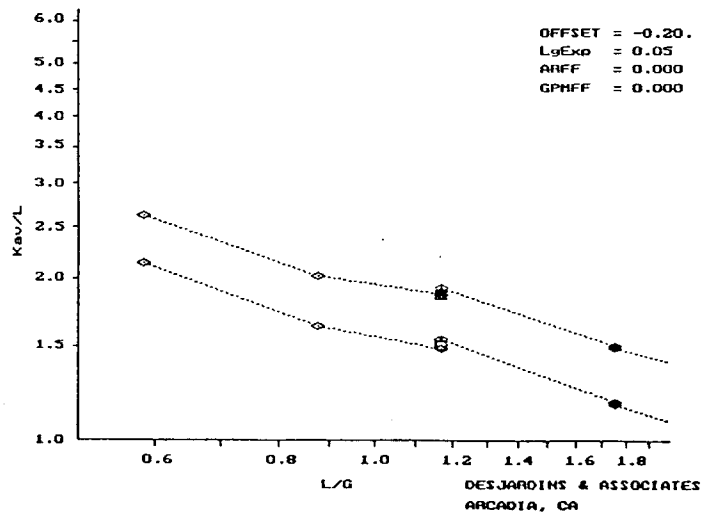
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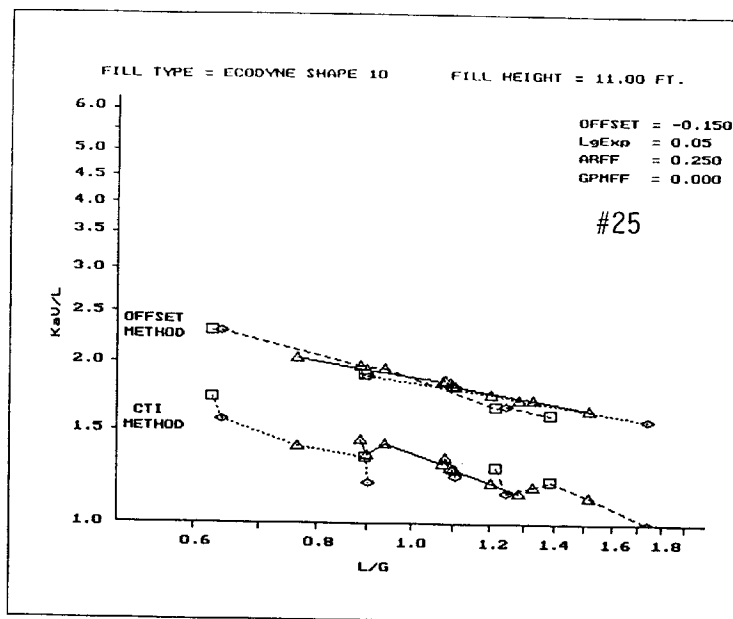
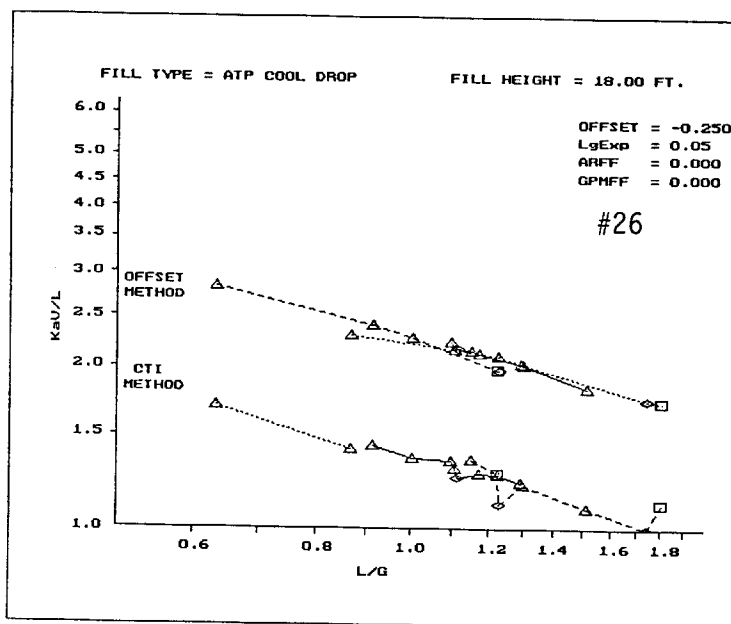
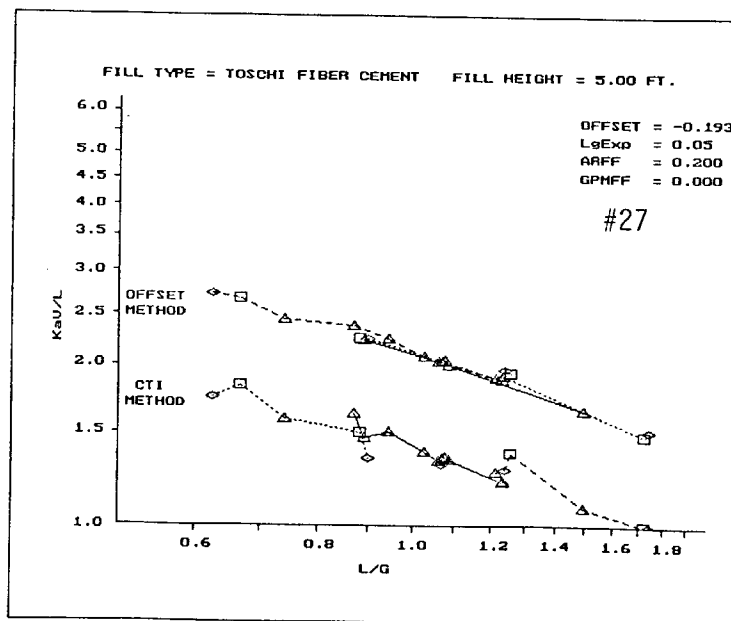


FILL TYPE = PRITCHARD C DECK

FILL HEIGHT = 20.00 FT.

#24





APPENDIX B

The key equations used to compute the data points are as follows:

The "offset" Ratio:

$$\text{OffsetAmount} = (T_{\text{film}} - T_{\text{water}}) / (\text{Enthalpy}_{\text{film}} - \text{Enthalpy}_{\text{WBT}})$$

(Note: The "OffsetAmount" is a negative number. At each point in the integration routine, where the bulk water temperature is known, search for T_{film} and its corresponding $\text{Enthalpy}_{\text{film}}$ that agrees with the chosen "offset". Then, $(\text{Enthalpy}_{\text{film}} - \text{Enthalpy}_{\text{WBT}})$ becomes the enthalpy driving force).

The "offset" Amount at any given L/G:

$$\text{OffsetAmount} = \text{offset} * (L_{\text{in}}/G)^{-0.5}$$

The Lewis Ratio (Le):

$$\text{Thermal Conductivity} = 0.0000224 * \text{TempAir} + 0.0137332$$

$$\text{Diffusivity} = 0.00215 / \text{psia} * (\text{TempR})^{2.5} / (\text{TempR} + 441)$$

$$\text{Cp}_{\text{vapor}} = 0.24 + (0.45 * (W_{s1} + W_{s2}) / 2)$$

$$\text{Le} = (\text{ThermalConductivity} / \text{Density} / \text{Cp}_{\text{vapor}} / \text{Diffusivity})^{2/3}$$

Actual Air Enthalpy : (Accounts for evaporation)

$$\text{ActualRise}_i = (L_{\text{in}}/G * (\text{Range})) + ((t_2 - 32) * (\text{Cp}_{\text{water}}) * (W_{s2} - W_{s1}))$$

$$\text{ActualEnthAir}_i = \text{InletAirEnth} + \text{ActualRise}_i$$

(Note: Values of Range and ActualEnthAir at various points in the integration routine are weighted depending on method of integration)

"offset" Delta Enthalpy:

$$\text{OffsetDeltaEnth} = ((\text{Enth@offsetTemp} - \text{ActualEnthAir}) * \text{Le}) + (W_{s_{\text{filmtemp}}} - W_{s_{\text{airtemp}}}) * \text{Cp}_{\text{vapor}} * T_{\text{air}}$$

Total Enthalpy Rise:

$$\text{EnthRise} = (\text{ActualEnthAirOut} - \text{InletAirEnth}) / (L_{\text{in}}/G)$$

KaV/L:

$$\text{KaV/L} = \text{SUM OF } (\text{IntegrationWeightingFactor}_i * \text{EnthRise} / \text{OffsetDeltaEnth}_i)$$