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An Experimental Heat Transfer Analysis of Summer Cooling Requirements for Tractor Cabs in South Dakota

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AN EXPERIMENTAL HEAT TRANSFER ANALYSIS OF SUMMER COOLING REQUIREMENTS FOR TRACTOR CABS IN SOUTH DAKOTA

By

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CABS IN SOUTH DAKOTA

This thesis is approved as a creditable and independent investigation by a candidate for the degree, Master of Science, and is acceptable for meeting the thesis requirements for this degree. Acceptance of this thesis does not imply that the conclusions reached by the candidate are necessarily the conclusions of the major department.

Thesis Advisor Date

Head, Agricultural Engineering Department Date

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INTRODUCTION

The past decade has brought many changes to the farmers of the Great Plains. Fewer people are engaged in agricultural production, and the average farm size continues to grow. Today's farmers are buying bigger tractors, many of which have cabs to minimize effects of increased noise pollution, dust, and temperature extremes.

Since these cabs were designed to serve as protection from noise and the elements, they must be relatively tight. During hot summer weather the enclosure will become unbearably hot. For this reason an adequate air conditioning system is almost a necessity.

Modern tractor cab air conditioning allows the operator to control his summer work environment. Benefits from this controlled cab environment include increased efficiency and reduced accident and fatigue levels. A comfortable air conditioned cab during hot summer operation will also be more conducive to obtaining and keeping farm help.

A satisfactory air conditioning system must maintain a comfortable environment even under the most severe conditions. However, from an economical viewpoint the best air conditioning system requires the least amount of energy or tractor horsepower to operate. Increased recirculation rates and lower air flow rates should help to lessen power requirements. Therefore, a smaller air conditioning unit that provides sufficient capacity to maintain operator comfort will be more economical than a larger over-sized unit. To develop the optimum size air conditioning unit for a given set of conditions, heat loads or cooling requirements must be established.

OBJECTIVES

The cooling requirements for an air conditioned cab during a South Dakota summer were studied to accomplish the following objectives:

1. Determination of the pertinent variables involved in the cooling process.
2. Development of an experimental heat transfer equation for predicting cooling loads.
3. Evaluate the magnitudes of cooling loads for some typical ambient conditions.
4. Establish total theoretical savings in horsepower and fuel consumption made possible by recirculation of a portion of the air that has already been cooled.

REVIEW OF LITERATURE

With the increased emphasis on tractor operator comfort and safety, many improvements have been made in cab construction and design. Summer cooling of these enclosed spaces is now recognized as being necessary to maintain operator comfort during hot summer weather.

Operator comfort during hot weather is dependent on both the pressurization system and the air conditioning system. These systems can be operated jointly or independently to provide optimum comfort to the operator.

To provide a comfortable environment, Hosler (2) lists the criteria a cab pressurization system should meet:

1. Maintain a minimum cab pressure of 0.2 inches of water to keep the cab dust-free. This would eliminate dust infiltration in winds up to 32 km/hr (20 mph).
2. At this pressurization level, a minimum airflow rate should be 9.91 m^3/min (350 ft^3/min).
3. The cab air filter should have a service interval compatible with the engine air cleaner.
4. Air distribution should provide for both direct and indirect flow over the operator.

5. The sound level of the pressurizer should not contribute significantly to the noise level measured at the operator's ears.

The air conditioning system must have sufficient capacity to maintain operator comfort. According to Hosler (2), there are two severe conditions to consider:

1. An extremely hot and dry ambient [43.3°C (110°F) dry bulb, 20% relative humidity (RH)] which tests the condenser sizing.
2. A moderately hot, very humid ambient [35°C (95°F) dry bulb, 60% RH] which tests the evaporator capacity.

The total tractor cab heat load is due to four sources (2). They are: conduction, solar radiation, infiltration, and heat sources inside the cab.

Hosler (2) made heat load analyses on several cabs using the following assumptions:

1. Ambient conditions - 37.8°C (100°F) 40% RH
2. Inside cab conditions - 23.9°C (75°F) 50% RH
3. Heading into a 12.1 km/hr (7.5 mph) wind and 30 degree sun angle.
4. Located at a 40 degree N latitude.
5. 8.5 m^3/min (300 ft^3/min) of fresh air required to maintain 0.2 inches of water pressure.

The values in Table 1 reflect his findings. As Table 1 shows, better sealing to reduce the fresh air needed for pressurization would greatly reduce the total heat load.

TABLE 1

CAB HEAT LOAD

Heat Source	Approximate Heat Load Watts	Approximate Heat Load BTU/hr	Approximate % of Total
Conduction	732	2,500	11
Solar	1,464	5,000	20
Infiltration	4,740	16,200	65
Miscellaneous	246	850	4
TOTAL	7,182	24,550	100

DETERMINATION OF PERTINENT VARIABLES

The total heat load to be removed from a tractor cab is affected by four primary sources (2): conduction, infiltration, solar radiation, and heat sources inside the cab. The heat sources inside the cab were not included in this analysis, but rather were assumed to contribute a small constant amount of heat to the total heat load. The pertinent variables affecting the total cab heat load at steady state conditions are defined in Table 2.

Use of dimensional analysis and the Buckingham Pi Theorem (6), resulted in the set of 5 independent and dimensionless π terms shown in Table 3. The derivation of the pi terms is described in the following paragraphs.

In a convective heat transfer analysis, a coefficient of transfer by convection is frequently used. It is defined as "the quantity of heat transferred across an interface per unit of time, per degree temperature differential, per unit area of interface" (6). For each set of data used in this study an overall heat transfer coefficient for the test cab was calculated. This coefficient related the quantity of heat transferred through the cab walls by conduction and radiation. Multiplying the heat transfer coefficient by the length term, $(\text{volume})^{1/3}$, and dividing by the thermal conductivity resulted in a dimensionless ratio, π_1, which was the dependent variable. This dimensionless ratio is more commonly referred to as the Nusselt Number.

The Reynolds Number, a pi term frequently used to describe

TABLE 2

VARIABLES AFFECTING TOTAL CAB HEAT LOAD

Variable No.	Symbol	Description	Dimensional Symbol *
1.	V	Mean velocity of air in cab, ft/min	LT^{-1}
2.	ρ	Density of inside air, lb_m/ft^3	ML^{-3}
3.	μ	Dynamic viscosity of the inside air, lb_m/hr-ft	$ML^{-1}T^{-1}$
4.	l	Length term $(\text{volume})^{1/3}$, ft	L
5.	c	Specific heat of air, BTU/lb_m-°F	$HM^{-1}\theta^{-1}$
6.	k	Thermal conductivity of air, BTU/hr-ft-°F	$HT^{-1}L^{-1}\theta^{-1}$
7.	h	Heat transfer coefficient, BTU/hr-ft^2-°F	$HT^{-1}L^{-2}\theta^{-1}$
8.	ΔT	Temperature difference (Outside-Inside), °F	θ
9.	T_{RI}	Inside black globe temperature, °R	θ
10.	T_{RO}	Outside black globe temperature, °R	θ

* M, L, T, θ, and H are the basic dimensions of mass, length, time, temperature, and heat, respectively.

TABLE 3

LIST OF PI TERMS

Pi Term No.	Description
1. $\pi_1 = hl/k$	Dependent Pi term (Nusselt number)
2. $\pi_2 = \rho Vl/\mu$	Reynolds number
3. $\pi_3 = c\mu/k$	Prandtl number
4. $\pi_4 = T_{RO}/\Delta T$	Ratio of outside radiant temperature to the temperature differential
5. $\pi_5 = (T_{RO}-T_{RI})/\Delta T$	Ratio of radiant temperature differential to the temperature differential

fluid flow, was chosen as π_2, one of the independent variables. The only quantity in this dimensionless ratio which was not constant for this study was the mean cab velocity, V. Since the mean velocity of air inside the cab was found to be closely correlated with air flow rate in CFM, the Reynolds Number could be varied by varying the air flow rate.

The Prandtl Number, π_3, for this study was dimensionally equivalent to the ratio of the heat absorbed by the air to the heat transferred through the walls of the cab. However, this dimensionless ratio was not included in the data analysis due to the insignificant variation in dynamic viscosity over the air temperature ranges inside the cab. The variation in π_3 was less than 0.3 percent, varying from .707 to .709.

The remaining two independent pi terms would therefore necessarily include the temperature differential, the inside black globe temperature, and the outside black globe temperature. Black globe temperatures were used to measure radient temperature effects.

The ratio of outside black globe temperature divided by the temperature differential between outside and inside the cab became π_4. The author will also refer to this dimensionless ratio as the Raddiff Number.

A ratio of the differential between outside and inside black globe temperatures divided by the temperature differential between outside and inside the cab was used for π_5. The inside radiant temperature was measured at the operator's shoulder level and varied by only a few degrees from the inside cab temperature. As later tests revealed, this pi term was not significant for the data used in this study.

The dimensionless form of a possible general equation can be expressed as:

$$\frac{hl}{k} = F\left(\frac{\rho Vl}{\mu}, \frac{c\mu}{k}, \frac{T_{RO}}{\Delta T}, \frac{T_{RO}-T_{RI}}{\Delta T}\right) \qquad \text{Equation 1}$$

EXPERIMENTAL PROCEDURE

The Test Facility

The test facility shown in Figure 1 was a portable, self-contained tractor cab powered by external electrical power. All of the ducts were 8 inches in diameter and completely insulated. The air conditioned cab had a thermostat sensitivity of ± 1 F.

The tractor cab test facility was 60 inches in height, 50 inches in width, and 60 inches in length for a total volume of approximately 104 cubic feet. The exterior surface was painted red and had a total surface area of approximately 110 square feet. The four sides comprised 91.66 square feet of the total area with approximately 40% being tinted glass as shown in Table 4.

Air Distribution System Design

The air distribution system inside the cab was designed to afford an "air curtain" effect around the operator. Six 2 inch by 14 inch louvers were installed as shown in Figures 2 and 3. This pattern created higher velocities in the region immediately outside the operator's working area. Lower velocities still prevailed around the operator's head and body region to alleviate the problem of discomfort from high velocity air. The louvers were adjustable so air direction could be controlled by an operator.

Air Flow Calibration

A schematic of the test facility is shown in Figure 4. The

Figure 1. Test Facility Used in Summer Cooling Load Studies

TABLE 4

TOTAL AREA OF TEST FACILITY WALLS

Location	Total Surface Area (ft^2)	Window Area (ft^2)
Front Wall	20.83	10.50
Side Walls	50.00	12.40
Rear Wall	20.83	13.60
TOTAL	91.66	36.50

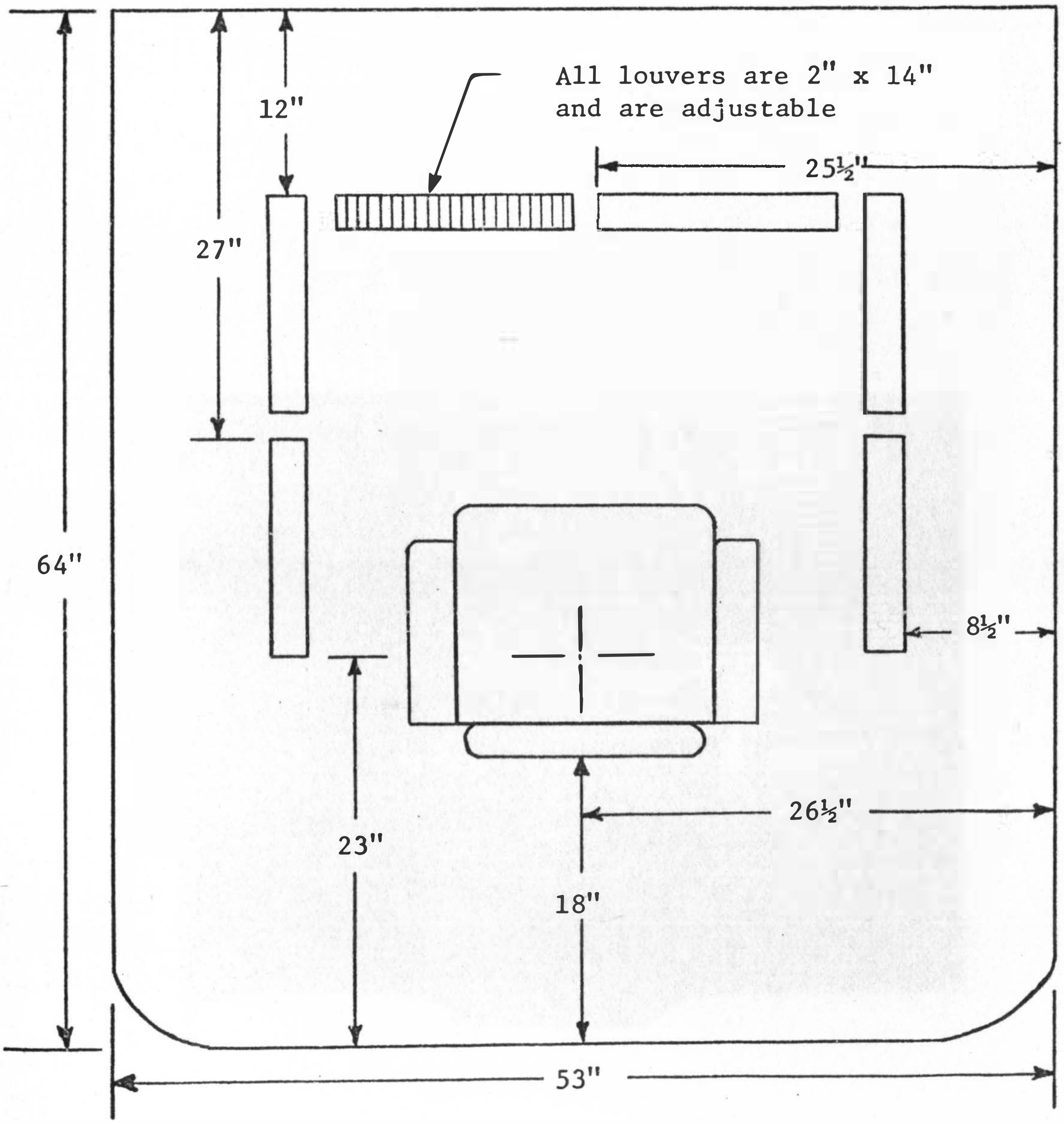

Figure 2. Top View of Cab Air Distribution System

Figure 3. Louvered Air Distribution System

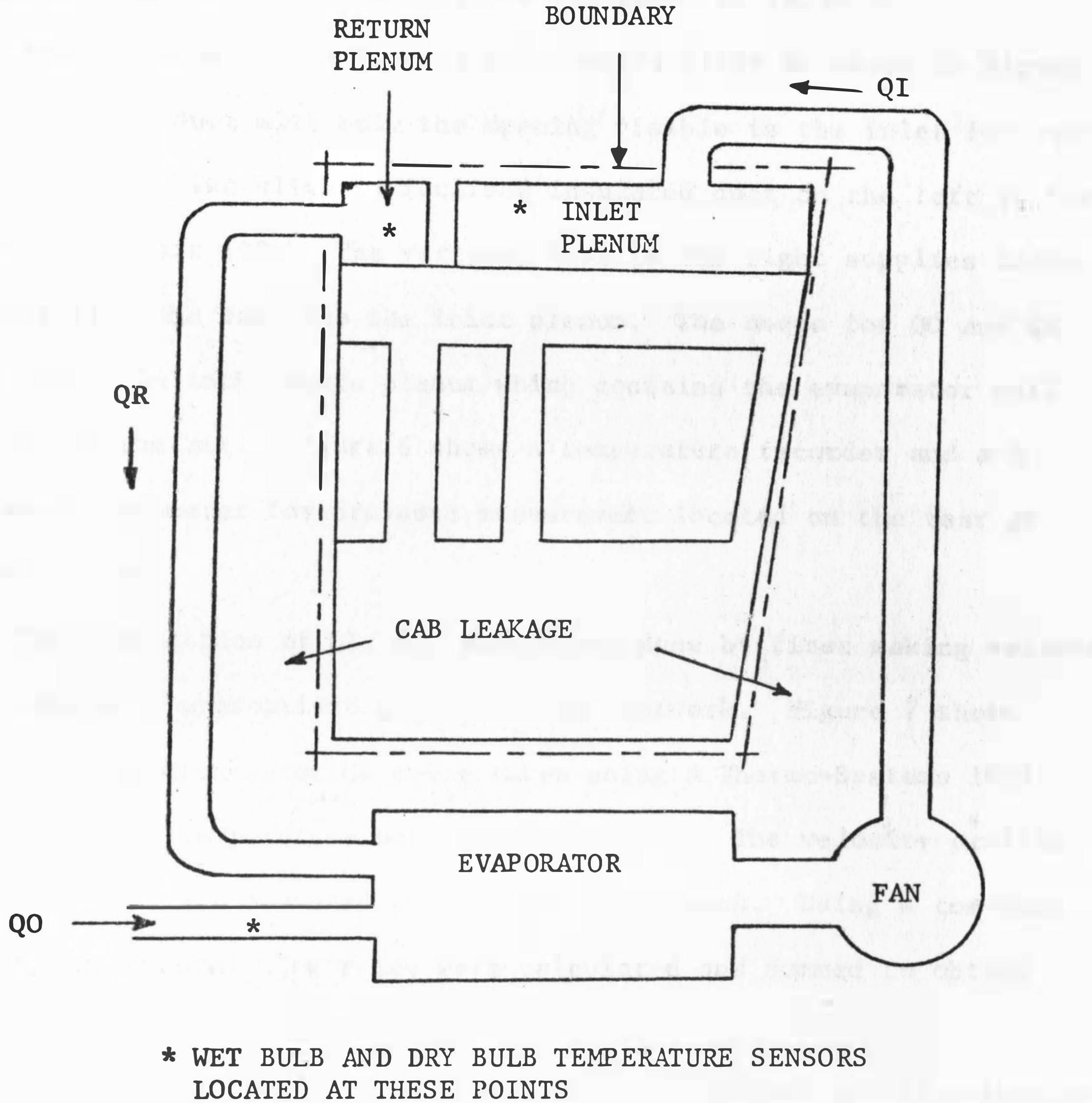

Figure 4. Schematic of Air Flow in the Test Facility

parameters used for air flow analysis are shown in Table 5.

The ductwork of the tractor cab test facility is shown in Figure 5. The bottom duct with only the opening visible is the inlet for outside air (QO). The slightly inclined insulated duct on the left is for recirculating air (QR). The vertical duct to the right supplies inlet air (QI) from the fan into the inlet plenum. The ducts for QO and QR merge into a 14 inch square plenum which contains the evaporator coil for cooling the air. Figure 6 shows a temperature recorder and a U tube water manometer for pressure measurement located on the rear of the main frame.

The calculation of QI, QR, and QO was done by first making velocity measurements at appropriate points in the ductwork. Figure 7 shows velocity measurements for QI being taken using a Thermo-Systems 1051 series constant temperature hot wire anemometer. The velocity profile across the duct was measured at 1/2 inch increments. Using a computer program, incremental flow rates were calculated and summed to obtain total flow rate.

The test facility was calibrated for the louvered air distribution pattern shown in Figure 3. Seven fan speeds were used ranging from 645-2080 RPM. Calibration of the system allowed selecting any desired air flow rate by setting fan RPM.

A linear regression analysis was made for flow rate in CFM versus fan speed in RPM for QI, QR, and QO. The results of the calibration are shown in Table 6. The recirculation rate was essentially constant

TABLE 5

PARAMETERS OF AIRFLOW IN TEST FACILITY

No.	Symbol	Description
1.	QI	Air flow input to cab, CFM
2.	QR	Air flow recirculated through return duct, CFM
3.	QO	Air flow through duct furnishing air from outside, CFM
4.	RPM	Fan speed, rev/min
5.	ΔP	Pressure differential between inside of cab and outside, inches of water

Figure 5. Tractor Cab Ductwork and Outside Black Globe

Figure 6. Temperature Recorder and U Tube Manometer

Figure 7. Taking Air Velocity Readings in Inlet Duct

TABLE 6

REGRESSION EQUATIONS FOR AIR FLOW WITH LOUVERED CEILING

Equation Number	Regression Equation * Dependent Variable, CFM	Variability Explained (R^2) (%)	Standard Deviation (CFM)
1.	QI = -43.69 + .394 (RPM)	99.88	7.86
2.	QR = 4.20 + .250 (RPM)	99.85	5.46
3.	QO = -1.80 + .086 (RPM)	99.74	2.51

* Each equation is based on 7 points. The RPM range was from 645 - 2,080.

throughout the study for all three air flow rates--varying from 74.3 to 76.2 percent on a mass air flow basis.

The three flow rates used in this study were 300, 450, and 600 CFM. The cab pressure at 300 CFM was the minimum necessary to keep the cab dust-free or 0.2 inches of water. The pressure inside the cab increased to 0.45 inches of water at 450 CFM, and at 600 CFM the cab pressure was 0.8 inches of water.

Velocity Measurement

In order to determine the value of Reynolds Number, π_2, a mean cab velocity was calculated for each of three flow rates. Two levels of measurements were taken as shown in Figure 8. The lower level was 28 inches above the floor or just above the operator's knees. The upper level was 46 inches above the floor at approximately shoulder height of the seated operator.

Velocity measurements were made using a Thermo-Systems Model VT-161D omni-directional low velocity probe. Velocity readings within the cab were taken at 6 inch intervals from front to rear and from right to left in the following manner:

1. The voltage output of the Thermo-Systems Model VT-161D velocity probe was recorded using one channel of a two channel oscillograph (Offner Type RS). The probe and oscillograph are shown in Figure 9. The inking pen recorded dynamic data on curvilinear graph paper fed at 5 mm/sec with a sensitivity setting such that one

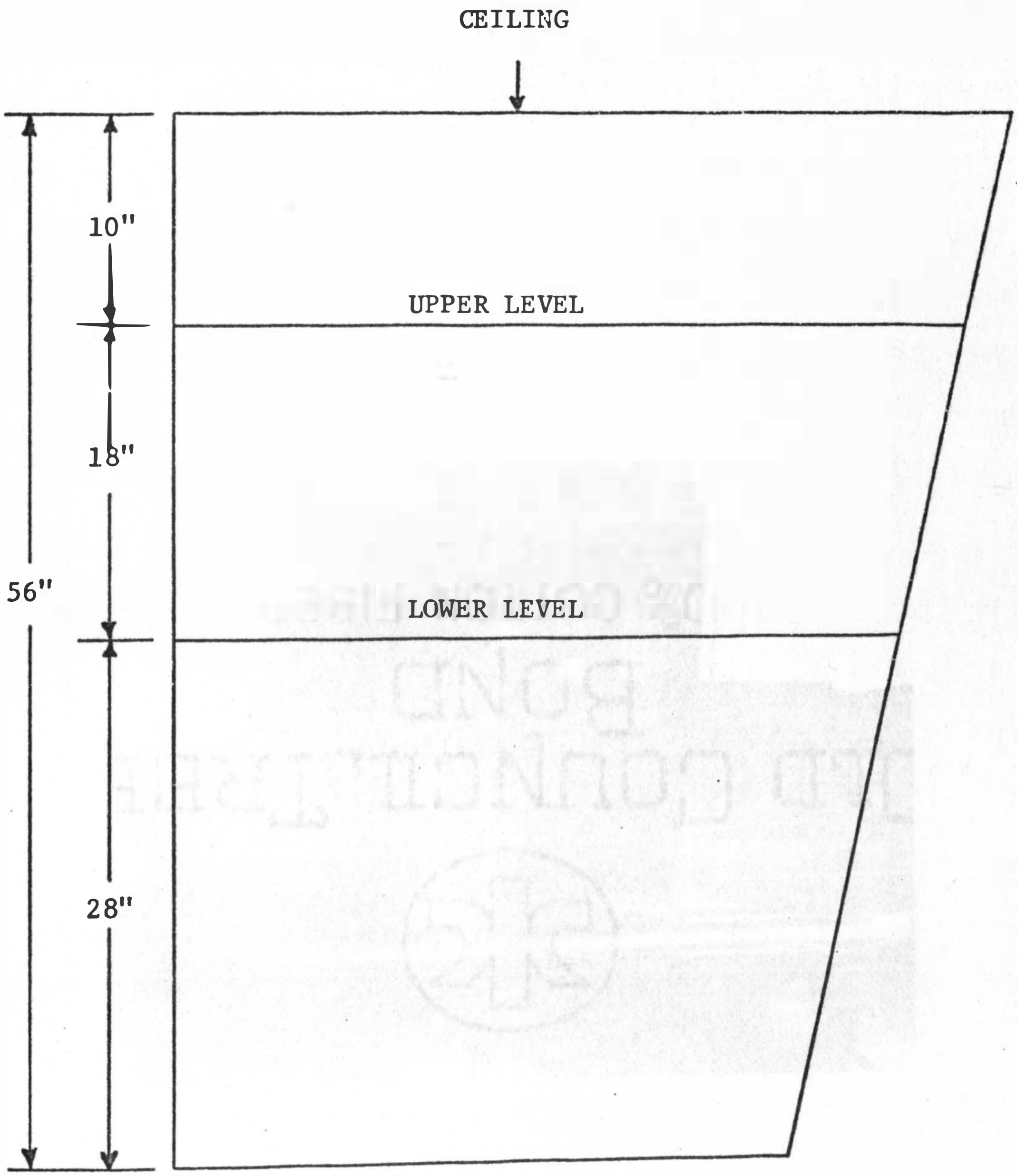

Figure 8. Levels of Air Velocity Measurement
Inside the Test Facility

Figure 9. Using Low Velocity Probe and Oscillograph

Figure 9. Using Low Velocity Probe and Oscillograph

line represented 0.2 volts.

2. All velocity readings were taken with the cab in the east wing of the Agricultural Engineering Building.

3. The velocity probe was hand held for all readings.

To obtain an average velocity reading for each data trace, the area between the zero velocity line and the data trace was measured using a compensated polar planimeter. This total area was then compared to a known voltage per known unit area to obtain the voltage reading for the total area. The calibration curve for this particular probe was then used to obtain a velocity for each voltage reading. Table 7 shows the velocities at each point. A mean cab velocity was also calculated for 300, 450, and 600 CFM. A linear regression equation of average cab velocity versus flow rate yielded the following regression equation for flow rates from 300 CFM to 600 CFM:

$V = 1.917 + 0.171\ (CFM)$

where V = mean cab velocity, feet per minute (fpm).

The coefficient of determination was 99.25 percent.

<u>Construction and Testing of Temperature-Humidity Devices</u>

Three temperature-humidity devices were constructed as shown in Figure 10 and 11. Air flow through the ducts provided the aspiration velocity. To insure that air flow in the ducts provided the necessary aspiration velocity for the three wet bulb measurements, a 15 minute calibration test was conducted. The dry bulb reading was 73 F, and

TABLE 7

AIR VELOCITIES INSIDE THE TRACTOR CAB

POINT *	300 CFM Lower Level (fpm)	300 CFM Upper Level (fpm)	450 CFM Lower Level (fpm)	450 CFM Upper Level (fpm)	600 CFM Lower Level (fpm)	600 CFM Upper Level (fpm)
1	12	26	28	30	44	64
2	47	43	59	49	49	66
3	50	215	98	260	90	330
4	70	93	158	45	188	78
5	70	33	90	49	130	115
6	25	92	50	91	55	130
7	25	58	50	63	99	136
8	20	20	25	40	60	70
9	17	43	26	22	50	110
10	60	98	140	130	135	190
11	65	100	100	140	91	170
12	15	20	55	40	80	45
13	50	80	63	140	98	182
14	52	30	70	99	90	64
15	51	35	50	35	91	70
16	30	41	49	50	70	63
17	45	13	70	58	80	75

TABLE 7 (continued)

POINT *	300 CFM		450 CFM		600 CFM	
	Lower Level (fpm)	Upper Level (fpm)	Lower Level (fpm)	Upper Level (fpm)	Lower Level (fpm)	Upper Level (fpm)
18	75	26	100	42	180	47
19	58	158	88	203	160	182
20	100	17	69	29	150	55
TOTALS	937	1,241	1,438	1,615	1,990	2,242
MEAN VELOCITY	54.5		76.3		105.8	

* Points 1 - 11 are from front to rear, and points 12 - 20 are from right to left (with respect to seated operator).

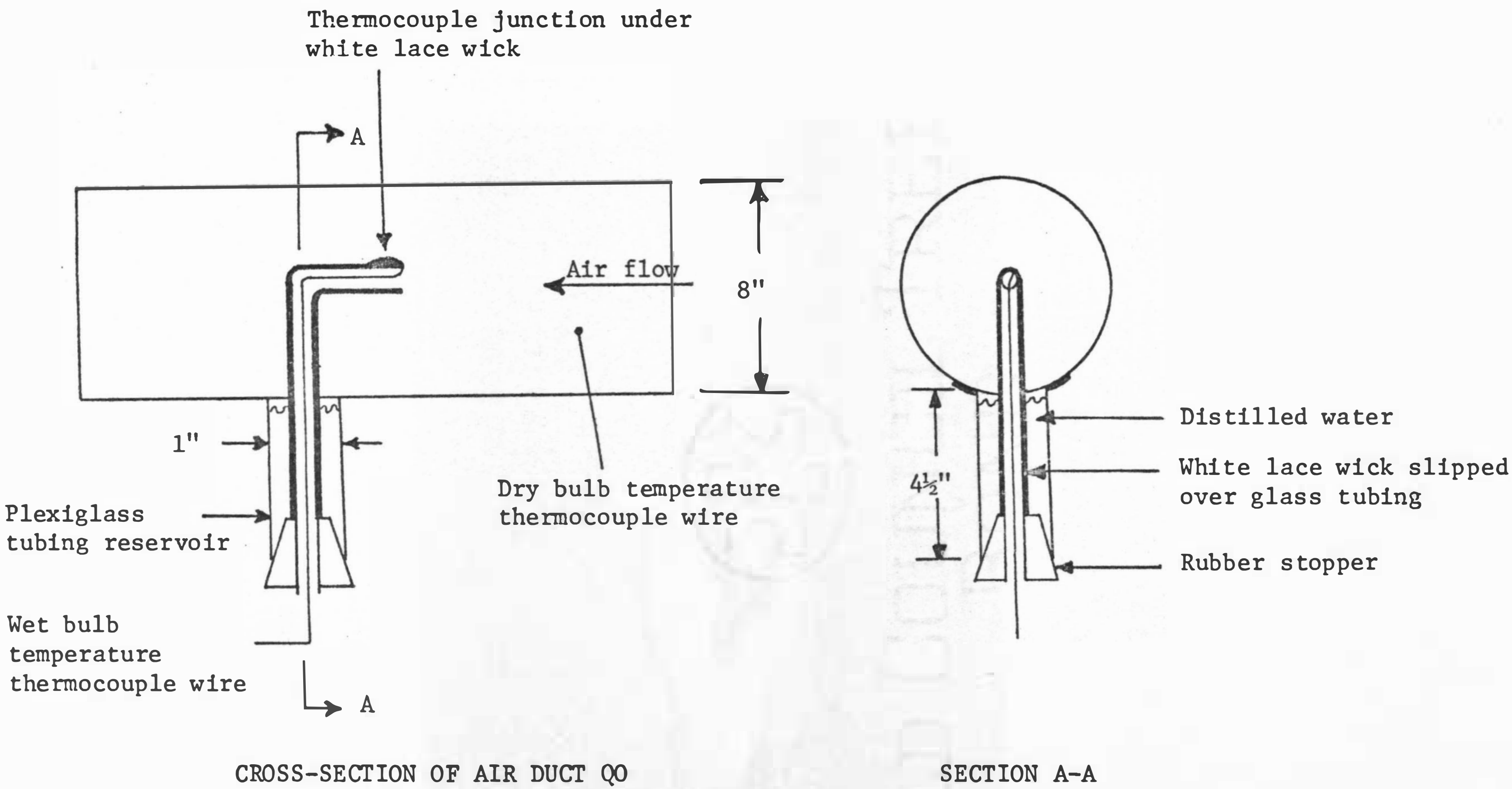

Figure 10. Details of Temperature-Humidity Device

Figure 11. Temperature-Humidity Device for Duct QO.

The wet bulb reading remained constant at 56 F for a velocity range of 5.17 to 28.2 feet per second. A battery powered psychrometer was used as a standard for comparison and agreed with the above wet bulb and dry bulb readings.

The duct velocities at each temperature-humidity device location were measured. Table 8 shows that all the velocities are within the tested aspiration velocity range.

Temperature Measurement

Temperature readings were made with 26 gauge copper-constantan thermocouples and recorded on a Honeywell 24 point, strip chart, recording potentiometer. All outside tests were conducted on the east side of the Agricultural Engineering Building with the test facility facing south. On several occasions the test facility was left inside the east wing of the Agricultural Engineering Building to simulate a very cloudy day with reduced radiant energy. The tests were conducted during June, July, and August of 1974, from 9:00 a.m. to 5:00 p.m. Central Daylight Savings Time. Twenty thermocouples were mounted on and within the cab as defined in Table 9.

Two 3 inch black globes with a thermocouple inserted in the center were used to measure the effects of radiant energy. The inside black globe was located at the approximate location of the operator's head and shoulder region, Figure 12. The outside black globe was located approximately 10 feet from the cab and 5 feet above the pavement as shown in Figure 5.

TABLE 8

ASPIRATION VELOCITIES FOR WET BULB MEASUREMENT

Point	300 CFM	450 CFM	600 CFM
QI	7.28*	13.00	18.70
QR	12.35	18.70	23.95
QO	13.44	20.30	28.20

* Aspiration velocities for all points are in feet per second.

TABLE 9

THERMOCOUPLE LOCATION IN TEST FACILITY

Thermocouple No. *	Location **
1.	Inside black globe
2.	Outside black globe
3.	QO-outside ambient in duct
4.	QO-outside ambient wet bulb in duct
5.	QI-inlet air plenum
6.	QI-inlet air plenum wet bulb
7.	QR-return air plenum
8.	QR-return air plenum wet bulb
9.	Inside-at cab leakage point
10.	Inside-wet bulb at cab leakage point
11.	Left wall-outside ambient
12.	Right wall-outside ambient
13.	Rear wall-outside ambient
14.	Average temperature at shoulder height around operator. Points 14, 15, and 16 were hooked in parallel.
17.	Waist high-right inside wall of cab
18.	Waist high-rear inside wall of cab
19.	Waist high-left inside wall of cab
20.	Waist high-front inside wall of cab

* Number corresponds to channel number of temperature recorder.

** Right is defined with respect to seated operator.

Figure 12. Inside Black Globe

PRELIMINARY INVESTIGATION

Temperature data was collected to determine if interactions of independent pi terms were important in an analysis of variance. Two levels of three independent pi terms were used. The value of the dependent variable, π_1, was determined by plotting the intersection of the wet bulb and dry bulb temperature lines at QO, QI, and QR on the psychrometric chart and using the procedure shown in Appendix A.

Three replications of three factors at two levels were analyzed. Table 10 lists the values for both levels used in the analysis of variance. The value of π_1 was calculated for each observation and the data shown in Appendix B was analyzed by digital computer. The analysis of variance is shown in Table 11.

An F test revealed that all the interactions were non-significant. This indicates that the response surfaces or curves do not intersect which meets the requirements for the function to be a sum according to Murphy (6). On the basis of the analysis of variance of the preliminary data used in this study, an additive model was assumed to best describe the system.

TABLE 10

NUMERICAL VALUES OF LEVELS OF FACTORS

Level	Factor		
	π_2	π_4	π_5
1	35,945.00	47.92	1.33
2	49,842.40	28.50	1.10

TABLE 11

ANALYSIS OF VARIANCE

Source	SS	df	MS	F
π_2	$1.996(10)^7$	1	$1.996(10)^7$	35.01*
π_4	$4.049(10)^8$	1	$4.049(10)^8$	52.69*
π_5	$1.106(10)^5$	1	$1.106(10)^5$	<1
Reps	$6.033(10)^6$	2	$3.017(10)^6$	---
$\pi_2\pi_4$	$1.498(10)^6$	1	$1.498(10)^6$	1.30
$\pi_2\pi_5$	$4.118(10)^2$	1	$4.118(10)^2$	<<1
$\pi_4\pi_5$	$2.862(10)^6$	1	$2.862(10)^6$	<1
$\pi_2\pi_4\pi_5$	$2.179(10)^6$	1	$2.179(10)^6$	1.73
π_2 Reps	$1.141(10)^6$	2	$5.703(10)^5$	---
π_4 Reps	$1.537(10)^7$	2	$7.684(10)^6$	---
π_5 Reps	$7.149(10)^6$	2	$3.574(10)^6$	---
$\pi_2\pi_4$ Reps	$2.297(10)^6$	2	$1.149(10)^6$	---
$\pi_2\pi_5$ Reps	$1.248(10)^7$	2	$6.241(10)^6$	---
$\pi_4\pi_5$ Reps	$1.602(10)^7$	2	$8.008(10)^6$	---
Error	$2.515(10)^6$	2	$1.258(10)^6$	---
TOTALS	$4.945(10)^8$	23	---	---

* Significant at the 5% level.

RESULTS AND DISCUSSION

Development of a Prediction Equation

Upon completion of the preliminary investigation, further data analysis was undertaken. A total of 65 temperature data sets (temperature recording points 1-20) were analyzed and values for the dependent and three independent pi terms ($\pi_1, \pi_2, \pi_4, \pi_5$) were calculated according to Table 3, and are listed in Appendix C.

The air flow in the test cab had a velocity profile sufficiently low to assume laminar flow. Also, the air flow along the cab walls could be considered laminar flow over flat plates. In order to calculate the Nusselt Number for a plate heated over its entire length, Holman (4) notes that Reynolds Number is raised to the 0.5 power. Therefore, the Reynolds Number pi term was assumed to be raised to the 0.5 power for this study.

The temperature ratios for π_4 and π_5 were originally assumed to be $T_{RO}^4/\Delta T^4$ and $(T_{RO}^4 - T_{RI}^4)/T_{RI}^4$, respectively, since radiant energy is related to absolute temperature to the fourth power. The discussion in the following paragraphs explains the relationship that was developed using this assumption.

A step-wise multiple linear regression analysis was done using the digital computer. The resulting equation was not valid for the 300 CFM flow rate as it indicated a decrease in total heat load with an increase in temperature differential. A further analysis of each temperature data set produced the following results:

1. The change in the outside black globe temperature per unit of time was determined.
2. Pi term calculations for each data set were rechecked to insure accuracy.
3. The heat transfer analysis was based on steady state conditions, and transient effects had unpredictable results. Therefore, any temperature data sets where the cab interior was not at an equilibrium condition or where the outside black globe temperature changed more than .25 F per minute between sets were excluded from the study. A total of 34 temperature data sets were included in the remaining analysis.

The resulting multiple regression equation had a coefficient of determination of 79.1 percent. However, the equation did not attribute enough significance to the radiant energy heat load. Using a temperature differential of twenty degrees, the equation attributed less than 1.0 percent of the total heat load to radiant energy. A comparison of predicted cooling loads versus observed cooling loads indicated the equation was not satisfactory.

The temperature ratios for π_4 and π_5 were changed to $T_{RO}/\Delta T$ and $(T_{RO} - T_{RI})/\Delta T$, respectively, in an attempt to improve the predictive capability of the model. A third step-wise multiple linear regression analysis resulted in a much improved model. Several test runs also revealed that the $\sqrt{\pi_2}$ was a more significant variable than π_2 as had been thought. An F Test indicated that π_5 was not significant at the .05 level. Therefore, it was excluded from the final regression equation. The step-wise multiple linear regression equation, Equation 2, and its limiting values are listed below:

$$\pi_1 = 90.29\sqrt{\pi_2} + 182.41\pi_4 - 14,045.86 \qquad \text{Equation 2}$$

$R^2 = 93.7\%$

where π_1 (minimum) = 5,417.00

π_1 (maximum) = 19,837.80

$\sqrt{\pi_2}$ (minimum) = 160.23

$\sqrt{\pi_2}$ (maximum) = 223.25

π_4 (minimum) = 25.00

π_4 (maximum) = 109.80

By substitution of the variables for each pi term and simplification, the prediction equation becomes:

$$Q = (138.13\sqrt{V} - 990.23)\Delta T + 12.86T_{RO} \qquad \text{Equation 3}$$

where Q = BTU/hr

V = mean cab velocity, fpm

ΔT = outside temperature minus inside cab temperature, F

T_{RO} = outside 3 inch black globe temperature, R

Twenty observations taken during various weather conditions in August, 1974 were used to test the predictive capability of the model. A plot of predicted cooling load values versus observed values using the step-wise multiple regression equation is shown in Figure 13. The average percent error for the 600 CFM flow rate was approximately 7 percent. The prediction equation tended to predict slightly high

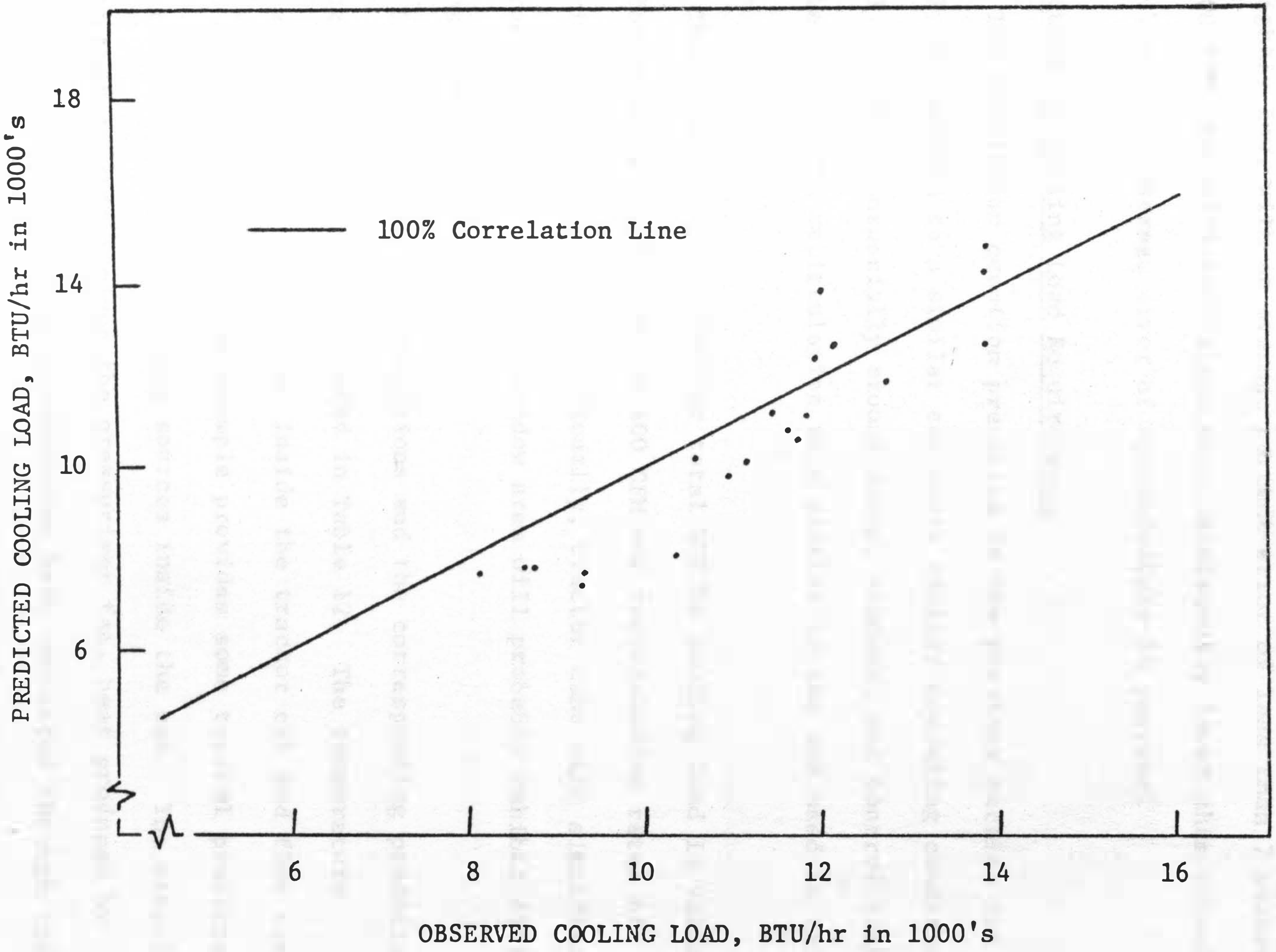

Figure 13. Predicted Versus Observed Cooling Loads.

for the 600 CFM flow rate. At 450 CFM the predicted values tended to be slightly low, with an average percent error of less than 7 percent. For 300 CFM the predicted values were consistently lower than actual values, with an average error of approximately 14 percent.

Prediction of Cooling Load Requirements

The prediction equation presented in the previous section can readily be applied to a similar cab under similar operating conditions. A well sealed cab especially around doors, windows, and control linkages is essential for a recirculation rate similar to the one used in this study.

The prediction equation for total BTU/hr cooling load is valid only for flow rates from 300 to 600 CFM and recirculation rates of approximately 75 percent. Additionally, tractor cabs with significantly different dimensions or total window area will probably exhibit different cooling requirements.

Some typical ambient conditions and the corresponding predicted cooling load magnitudes are listed in Table 12. The temperature differential between outside and inside the tractor cab and flow rate in CFM are also listed. This example provides some typical predicted cooling loads excluding the heat sources inside the cab. The miscellaneous heat sources include the pressurizer fan, heat produced by the operator and engine, and transmission heat conducted through the cab walls. In a well insulated cab, these miscellaneous heat sources contribute very little to the total heat load (3). A constant value

TABLE 12

COOLING LOADS FOR VARIOUS AMBIENT CONDITIONS

Ambient Conditions	ΔT	CFM	Predicted Cooling Load (BTU/hr)
Bright, Sunny, 95 F (T_{RO} = 575 R)	20	600*	16,006.
Bright, Sunny, 95 F (T_{RO} = 575 R)	20	450**	11,721.
Bright, Sunny, 95 F (T_{RO} = 575 R)	20	300***	7,985.
Bright, Sunny, 86 F (T_{RO} = 572 R)	12	600	12,523.
Bright, Sunny, 86 F (T_{RO} = 572 R)	12	450	9,952.
Bright, Sunny, 86 F (T_{RO} = 572 R)	12	300	7,710.
Partly Cloudy, 83 F (T_{RO} = 560 R)	9	600	11,077.
Partly Cloudy, 83 F (T_{RO} = 560 R)	9	450	8,408.
Partly Cloudy, 83 F (T_{RO} = 560 R)	9	300	7,467.

* V = 105.8 fpm

** V = 76.3 fpm

*** V = 54.5 fpm

of 850 BTU/hr was approximated for this value by Hosler (2). This constant amount should be added to the prediction equation to correct for the additional cooling load requirement.

Theoretical Horsepower and Fuel Consumption Savings

Recirculation of a portion of the air that has already been cooled results in a considerable reduction of total cooling load required. This is especially true on very hot, sunny days with high air flow rates. An example of the theoretical horsepower and fuel consumption requirements for cooling the test cab on a moderately hot day (97 F dry bulb, 50 percent relative humidity) are shown in Table 13.

From this example, it can be shown that the maximum dollar savings possible using 75 percent recirculation versus 0 percent recirculation would be $0.23 per hour. This is assuming diesel fuel costs to be $0.40 per gallon. Total theoretical horsepower requirements were reduced by approximately 55 percent.

The results of this study indicate that an air conditioning unit that removes 18,000 BTU/hr should be adequate to cool the test cab during South Dakota summers. Since most tractor cabs manufactured today are designed to serve as protection from noise and the elements and are built relatively tight, an 18,000 BTU/hr air conditioning unit should provide adequate cooling for practically all late model cabs operating in South Dakota.

TABLE 13

THEORETICAL HORSEPOWER AND FUEL CONSUMPTION SAVINGS AT VARIOUS RECIRCULATION RATES*

Percent Recirculation	Total Q (BTU/hr)	Total Theoretical HP 100% Efficiency (Horsepower)	Total Fuel Consumption** (Gal/hr)
75%	15,500	6.09	.49
50%	21,300	8.37	.67
25%	27,400	10.77	.86
0%	34,000	13.36	1.07

* The temperature data for this example was taken from Appendix A. The air flow rate was 600 CFM.

** Assuming a diesel tractor with a specific fuel consumption of 12.5 HP-HR/GAL.

CONCLUSIONS

The following conclusions were reached in this study:

1. The pertinent variables involved in the cooling process can be described by one dependent pi term (Nusselt Number), and two independent pi terms (Reynolds Number and Raddiff Number). The Prandtl Number was essentially constant for the temperature range inside the cab.

2. A highly significant relationship developed between π_1, π_2 and π_4 predicted 93.7 percent of the variation in Nusselt Number:

   $$\pi_1 = 90.29 \sqrt{\pi_2} + 182.41\pi_4 - 14,045.86$$

   The resulting prediction equation for cooling requirements is:

   $$Q = (138.13\sqrt{V} - 990.23)\Delta T + 12.86\ T_{RO}$$

   where: Q = BTU/hr

   V = Velocity, feet per minute

   ΔT = Temperature differential

   T_{RO} = Outside black globe temperature, degrees Rankine

3. For a typical 90 F sunny day with an air flow rate of 450 CFM and a ΔT = 20 F, an average value for predicted cooling load would be approximately 12,500 BTU/hr.

4. At a flow rate of 600 CFM, an increase in the recirculation

rate from 0 to 75 percent for the conditions given in Appendix A will reduce the theoretical horsepower and fuel consumption required to cool the test facility by approximately 55 percent.

5. An 18,000 BTU/hr air conditioning unit should provide adequate summer cooling for well designed tractor cabs operating in South Dakota.

SUMMARY

The increased use of tractor cab air conditioning has resulted in increased productivity and reduced fatigue levels. However, adequate design criteria for cooling load requirements in South Dakota is not available. Therefore, a study of summer cooling requirements in South Dakota was conducted.

Using the principles of similitude, five dimensionless groups (pi terms) were established, describing fluid properties, temperature, and tractor cab geometry. Observations on the effects of air flow rate, black globe temperature, and difference between inside and outside air temperature on the cooling load requirements were conducted under actual weather conditions during the summer of 1974. Statistical analyses of the basic data and pi terms were performed using linear regression, analysis of variance, and step-wise multiple linear regression techniques.

Results indicated that air flow rate, recirculation rate, temperature differential, and outside black globe temperature had significant effects on the cooling load requirements. On the basis of an analysis of variance, an additive model was assumed to best describe the system.

The model for prediction of tractor cab cooling loads developed by using step-wise multiple linear regression analysis was: $Q = (138.13\sqrt{V} - 990.23)\Delta T + 12.86\ T_{RO}$. The coefficient of determination was 93.7 percent.

The results of this equation indicate that substantially smaller cooling loads are possible when higher recirculation rates are used.

Lower air flow rates also reduce the heat load that must be removed to maintain a comfortable environment.

Results of this study indicate that an air conditioning unit which removes 18,000 BTU/hr should be adequate to cool well designed tractor cabs operating in South Dakota.

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APPENDICES

APPENDIX A

CALCULATION OF π_1

CALCULATION OF π_1

Presented is a sample calculation of π_1 from temperature data taken on the afternoon of June 20, 1974.

Wet and dry bulb temperatures:

QO 97.0 F dry bulb and 80.5 F wet bulb
QI 80.0 F dry bulb and 70.0 F wet bulb
QR 83.5 F dry bulb and 70.0 F wet bulb

Corresponding specific volumes are:

QO 14.45 ft^3/lb
QI 13.89 ft^3/lb
QR 13.96 ft^3/lb

Mass flow rate:

QO = 138.7 CFM/14.45 ft^3/lb = 9.60 lb/min
QR = 412.6 CFM/13.96 ft^3/lb = 29.56 lb/min
Total m = 39.16 lb/min

H_1-H_2 = enthalpy change per pound of dry air

H_1-H_2 = 6.25 BTU/lb

$Q = m(\Delta H) = (39.16\ lb/min)(6.25\ BTU/lb)(60\ min/hr) = 14,685.0\ BTU/hr$

h = overall heat transfer coefficient

$h = Q/l^2(\Delta T) = 14,685.0/(4.7)^2(12) = 55.37\ BTU/ft^2\text{-hr-F}$

$\pi_1 = hl^*/k = (55.37\ BTU/ft^2\text{-hr-F})(4.7\ ft)/.0150\ BTU/hr\text{-ft-F} = 17,350.3$

$^*l$ is the length term or $(\text{volume})^{1/3}$. $l^2 = (\text{volume})^{2/3}$.

APPENDIX B

DATA FOR ANALYSIS OF VARIANCE

DATA FOR ANALYSIS OF VARIANCE

Level π_2	π_4	π_5	Replications	Corresponding Value of π_1
1	1	1	1	15,735.6
1	1	2	1	14,483.8
1	2	1	1	6,914.0
1	2	2	1	8,544.9
2	1	1	1	18,717.6
2	1	2	1	14,750.5
2	2	1	1	9,111.2
2	2	2	1	8,674.8
1	1	1	2	13,609.1
1	1	2	2	15,317.6
1	2	1	2	7,944.9
1	2	2	2	7,552.5
2	1	1	2	16,256.2
2	1	2	2	17,350.3
2	2	1	2	11,853.1
2	2	2	2	8,661.2
1	1	1	3	16,843.4
1	1	2	3	17,082.9
1	2	1	3	8,447.8
1	2	2	3	7,377.4
2	1	1	3	16,412.1
2	1	2	3	23,537.6
2	2	1	3	8,148.3
2	2	2	3	8,279.0

APPENDIX C

EXPERIMENTAL DATA

EXPERIMENTAL DATA

π_1	Dimensionless Ratios $\sqrt{\pi_2}$*	π_4	π_5	Rate of change of T_{RO} degrees/min
16,246.2	223.25	46.16	1.400	0.2
16,412.1	223.25	48.00	1.458	0.0
14,750.5	223.25	53.33	2.476	0.6
8,279.0	223.25	29.89	1.053	0.7
8,148.3	223.25	28.15	0.750	1.0
11,853.1	223.25	32.82	0.765	0.2
9,111.2	223.25	29.61	0.816	1.0
8,661.2	223.25	28.60	1.200	0.6
8,674.0	223.25	33.47	1.412	1.4
22,299.4	223.25	61.83	2.056	1.0
23,537.6	223.25	55.70	1.500	0.6
17,350.3	223.25	47.83	1.333	0.2
11,903.5	223.25	36.00	1.437	0.2
21,086.3	223.25	57.50	2.700	0.5
15,238.8	223.25	51.90	0.000	0.0
15,648.6	223.25	54.50	0.000	0.0
14,979.4	223.25	54.50	0.000	0.0
10,605.0	223.25	24.44	0.000	0.0
7,871.0	223.25	26.19	0.000	0.0
15,995.2	223.25	44.38	1.308	0.2
15,928.0	223.25	50.26	1.652	0.0
15,735.6	189.60	47.92	1.375	0.3
13,609.1	189.60	47.10	1.469	0.2
6,914.9	189.60	38.53	1.600	2.0
8,447.8	189.60	31.11	0.750	0.2
7,377.4	189.60	27.14	1.024	0.5
7,944.9	189.60	29.08	1.026	0.7
7,552.5	189.60	27.85	1.098	0.2
8,544.9	189.60	28.50	1.100	0.2
14,483.8	189.60	55.70	1.900	1.3
16,843.4	189.60	68.30	1.500	2.1
15,317.6	189.60	55.65	1.950	1.0
17,082.9	189.60	53.33	1.905	0.4
10,732.2	189.60	42.69	1.077	0.8
19,224.2	189.60	65.65	2.294	0.4
10,315.6	189.60	41.88	0.000	0.1
9,898.2	189.60	41.92	0.000	0.0

EXPERIMENTAL DATA (continued)

	Dimensionless Ratios			Rate of change of T_{RO} degrees/min
π_1	$\sqrt{\pi_2}$*	π_4	π_5	
9,547.0	189.60	40.37	0.000	0.0
5,417.0	189.60	26.19	0.000	0.0
5,922.0	189.60	26.19	0.000	0.0
22,217.7	189.60	67.65	1.765	0.4
17,650.1	189.60	57.30	1.350	0.4
12,511.0	189.60	38.33	1.000	0.8
6,714.3	160.23	27.52	0.714	0.4
10,977.0	160.23	58.00	1.263	0.2
8,815.7	160.23	46.08	2.240	0.2
9,642.2	160.23	66.94	2.706	0.4
11,393.8	160.23	71.25	3.000	0.2
8,991.7	160.23	45.36	1.360	0.6
13,787.4	160.23	57.00	2.400	0.6
6,892.9	160.23	45.44	1.640	2.4
9,278.0	160.23	45.72	1.920	0.3
8,519.2	160.23	45.67	0.500	0.8
9,208.5	160.23	38.21	0.690	0.2
11,296.5	160.23	61.39	1.278	0.2
11,479.8	160.23	50.27	0.818	0.3
6,077.0	160.23	37.55	0.000	0.0
6,621.0	160.23	36.33	0.000	0.0
7,893.0	160.23	38.93	0.000	0.0
19,837.8	160.23	109.80	0.000	0.0
12,706.1	160.23	64.65	0.000	0.0
7,241.2	160.23	27.50	0.000	0.0
5,640.0	160.23	25.00	0.000	0.0
8,463.6	160.23	30.60	1.026	1.2
8,034.7	160.23	29.59	0.590	0.2

* This term is dependent on the flow rate. The pi term values and corresponding flow rates are listed below:

$\sqrt{\pi_2}$	Flow Rate (CFM)
223.25	600
189.23	450
160.23	300