# COMPRESSOR PERFORMANCE MODELLING AND MONITORING

by

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## ABSTRACT

One of Du Pont's plants operates a 20,000 hp synchronous motor driven process air compressor consisting of four compression sections and three intercoolers (Figure 1). A six percent capacity increase from refitting the first stage impeller led to the development of an analytical model of the compression
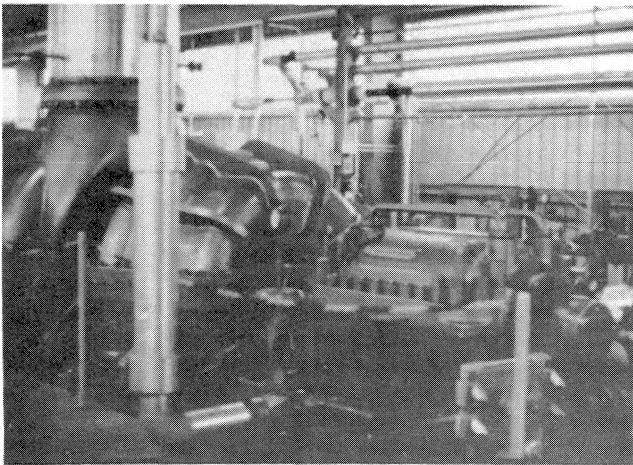
*Figure 1. Compressor Installation.*

train to study other ways to increase efficiency and capacity. Three computerized versions of this model are discussed. The POINT program calculates all system parameters for a single operating point. The MAP program calculates all outlet and power variables needed for a complete performance map. The COMPARISON program is used for on-line or off-line performance monitoring of each system component.

Results are presented from a thorough examination of the effects of speed, discharge pressure, inlet pressure, inlet temperature, cooling water temperature, inlet filter losses, and intercooler fouling. Guidelines are included for minimizing operating costs and meeting new capacity or range requirements. Although the numerical results apply to this particular machine, the general conclusions and guidelines are qualitatively applicable to other large compression trains.

## INTRODUCTION

Plant compressor requirements often change several times over the life of an installation and the future needs are frequently uncertain. Product demand swings may require minimum flow operation with venting to avoid surge during one time period and then a rapid change to absolute maximum flow requirements with a need for more capacity. Flow requirements for the compressor described herein have varied over a broad range since its installation. Efforts to meet these changing demands and to minimize operating costs are discussed.

Original refitting of the first stage impeller was so successful that a detailed study was made to determine what else could be done to meet capacity requirements at the lowest cost. A computer model of the compression train facilitated the study and provided accurate facts and numbers on which economic decisions could be based. The methods of analysis, the computer programs used, and the results of the analyses are presented.

One of the most valuable results of the study is that an on-line performance monitoring program has been produced that evaluates the condition of every compression section and intercooler on a daily basis. This monitor detects early deterioration of system components and eliminates frequent disassembly of components that are in perfect condition.

## IMPELLER REFITTING

Prior to December 1979, a poor mismatch between first compressor case diaphragm and impeller contours caused the compression train to operate at reduced maximum capacity. Measurements showed that when minimum impeller face clearances were set to prevent rubs, clearances at other locations were excessive. Clearances on both sides of the 30 in impeller were measured with feeler gages, and material was removed to obtain proper clearances (Figure 2). Desired clearances of 0.070 in at the outside diameter to 0.060 in at the eye were obtained by removing up to 0.020 in of material from vanes on both sides of the impeller. A tracer lathe was used with a contour pattern to remove the material so that the contours of
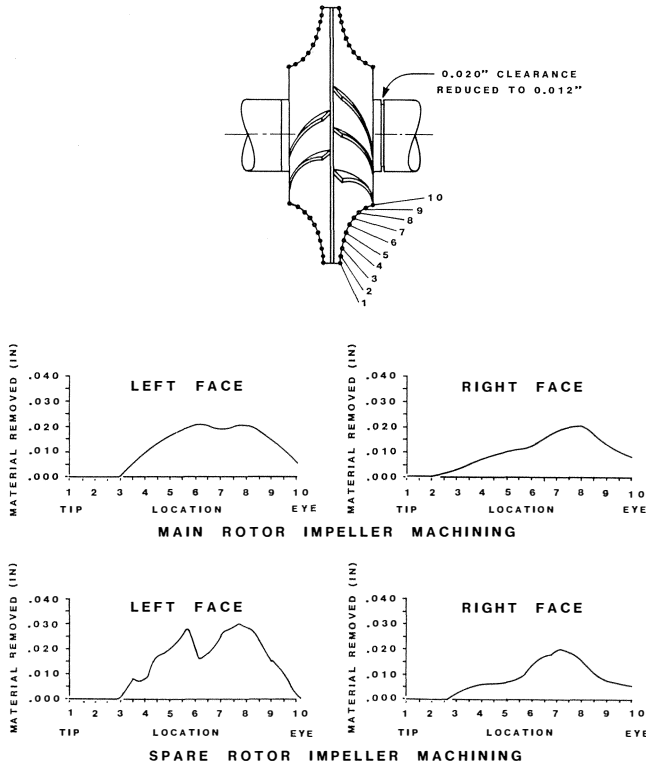
*Figure 2. Impeller Refitting.*

the impeller face and diaphragms would match. After recontouring the impeller and reducing the axial clearance between the impeller and its locating shoulders from 0.020 in to 0.012 in, we were able to move the diaphragms 0.060 in closer together. Relocation of the diaphragm was performed by machining and installing a new diaphragm locator key.

While the first case was disassembled, the spare rotor was also set in place and clearance variations measured. Its variations were as much as 0.030 in from the desired values. The spare rotor was also recontoured using the clearance variation measurements and the original tracer pattern as a guide. The spare rotor's impeller vane height was slightly less than that of the original vane. This will require a slightly modified diaphragm locator key to provide proper running clearances, if it is ever installed. Of course both rotors were rebalanced after the recontouring work.

After startup, the measured maximum flow rate was found to be about six percent above the previous maximum value at similar operating conditions. Many more data points were taken over the following months, and it was found that the maximum flow was consistently higher than that before recontouring by roughly six percent (Figure 3).

The increase in compressor capacity was so valuable to the plant that the engineers began to wonder what other changes could be made to increase capacity or efficiency. Without reliable calculations, improvements could only be discussed in uncertain generalities. Development tests to look for capacity increases were not desirable, because of risks of process upsets, and the need to run only maximum rates without downtime for changes. This led to the choice of an analytical model where the effect of any variable could be thoroughly examined without affecting the process in any way. An analytical model of the entire compressor system was developed using a combination of empirical data and standard theoretical calculations.
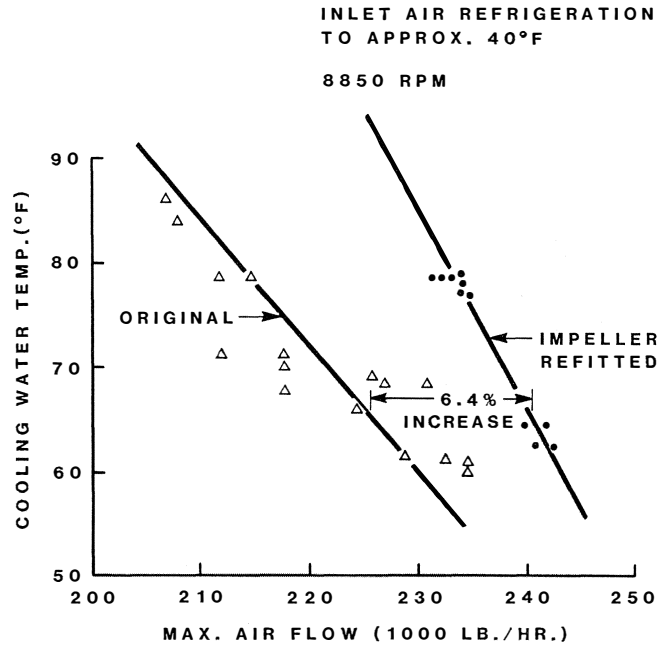


*Figure 3. Results of Refitting Impeller.*

## METHOD OF CALCULATION

### Description of Model and Variables

A schematic drawing of the model appears in Figure 4. The model includes an inlet booster blower and an aftercooler, with four main compression sections and three intercoolers. Inlet guide vanes are present at the inlet to the booster blower and first main compression stage. Information used by the model on polytropic efficiencies and polytropic head coefficients is provided in Figure 5 for varying volumetric flow at the operating speed of 8594 rpm. A description of the symbols used in the analysis is included in the NOMENCLATURE section.
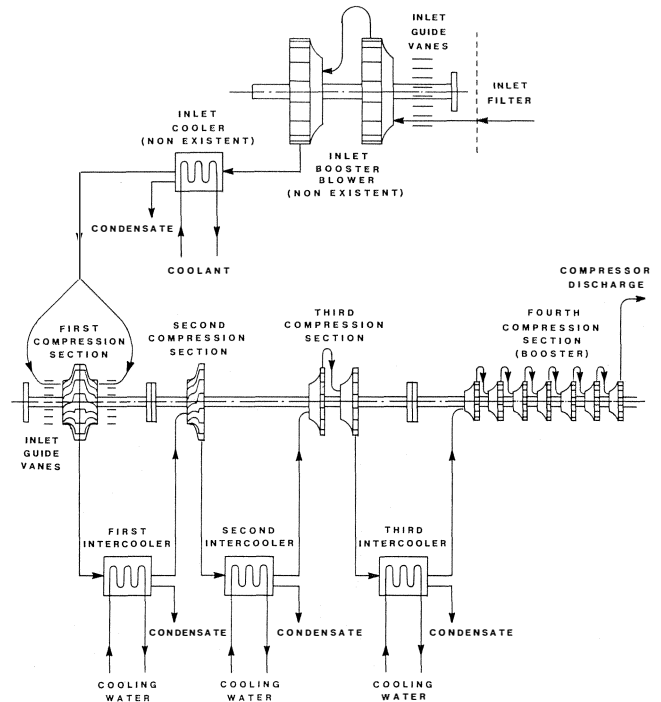


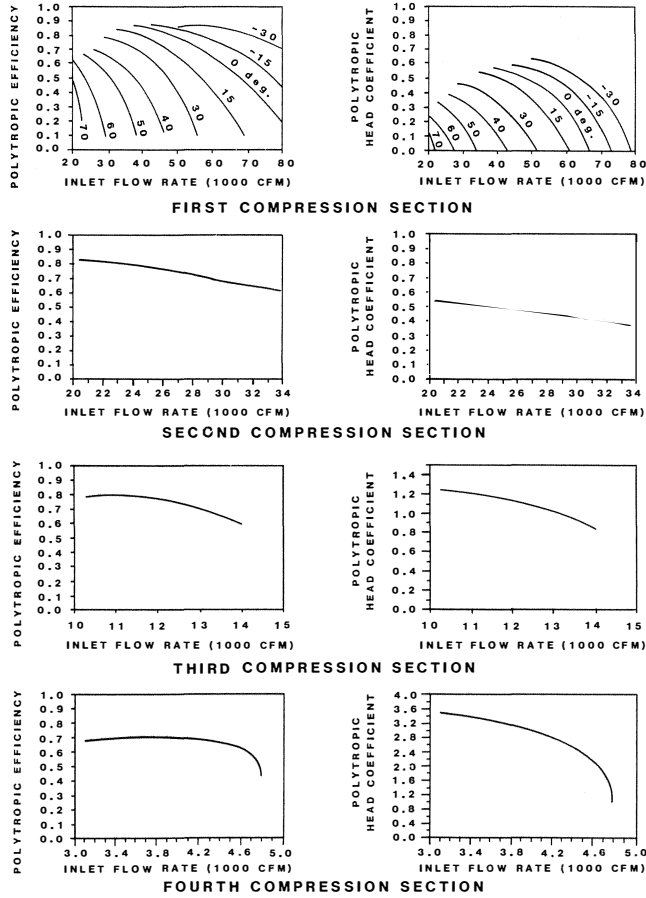*Figure 4. Schematic of Analytical Model.*

*Figure 5. Polytropic Efficiency and Head Coefficient Curves.*

## Compression Calculations

The polytropic head coefficient and polytropic efficiency versus volumetric flow rate curves are used with theoretical equations to calculate exit conditions at each compression section. The original curves furnished by the OEM have been modified, interpolated, and extrapolated by using extensive field measurements. Modified curves are shown in Figure 5 for a speed of 8594 rpm. These curves are used to develop information for other speeds with only small errors by converting the axis from volumetric flow rate to volumetric flow rate/rpm. Most of the data are reasonably accurate, especially for the second, third, and fourth compression sections. However, the first stage has a set of curves for each guide vane position and an infinite number of guide vane positions. Since plant operation did not permit operation at numerous guide vane positions or at surge points, extensive interpolation and extrapolation processes were necessary. Results from the program to date have shown that the calculations are generally very accurate and are certainly good for comparative studies.

For compression calculations, the volumetric inlet flow rate, $\dot{V}_i$, for the compression section is calculated from the following equation:

$$\dot{V}_i = \frac{Z_i \dot{m}_i \ R_i(T_i + 460.)}{144 \ P_i} \tag{1}$$

Using the calculated inlet flow rate and the specified inlet guide vane position (for first section only), the value for polytropic

head coefficient, $\mu_{P_{i,i+1}}$, and polytropic efficiency, $\eta_{P_{i,\ i+1}}$, is determined by interpolation in the curves. The polytropic coefficient, $n_{i,\ i+1}$, is then calculated from:

$$\eta_{P_{i,\ i+1}} = \frac{\dfrac{k-1}{k}}{\dfrac{n_{i,\ i+1}-1}{n_{i,\ i+1}}} \tag{2}$$

Next, the polytropic head, $H_{P_{i,\ i+1}}$, is calculated from:

$$\mu_{P_{i,\ i+1}} = \frac{g \ H_{P_{i,\ i+1}}}{v_i^2} \tag{3}$$

The discharge pressure, $P_{i+1}$, is calculated from:

$$\mu_{P_{i,\ i+1}} = \frac{gR_i(T_i + 460.)\left(\dfrac{n_{i,\ i+1}}{n_{i,\ i+1}-1}\right)\left[\left(\dfrac{P_{i+1}}{P_i}\right)^{\dfrac{n_{i,i+1}-1}{n_{i,\ i+1}}}-1\right]}{v_{i,\ i+1}^2} \tag{4}$$

The discharge volumetric flow rate, $\dot{V}_{i+1}$, is calculated from:

$$P_i \dot{V}_i^{n_{i,\ i+1}} = P_{i+1} \ \dot{V}_{i+1}^{n_{i,\ i+1}} \tag{5}$$

The discharge temperature, $T_{i+1}$, is next calculated from:

$$P_{i+1} \ \dot{V}_{i+1} = \frac{Z_{i+1} \ \dot{m}_i \ R_{i+1}(T_{i+1} + 460)}{144} \tag{6}$$

For situations where performance with a different number of booster impellers is needed, the previously calculated discharge pressure, $P_{i+1}$, is recalculated using this equation:

$$P_{\substack{i+1 \\ NEW}} = \frac{\text{number of impellers}}{7} \ (P_{i+1} - P_i) + P_i \tag{7}$$

Discharge volumetric flow rate and discharge temperature are then calculated as before using $P_{\substack{i+1 \\ NEW}}$ instead of $P_{i+1}$.

The actual head, $H_{i,\ i+1}$, or work per unit pound of air, is calculated by:

$$H_{i,\ i+1} = \frac{H_{P_{i,\ i+1}}}{\eta_{P_{i,\ i+1}}} \tag{8}$$

The stage gas horsepower, $GHP_{i,\ i+1}$, is calculated by:

$$GHP_{i,\ i+1} = \frac{\dot{m}_{i,\ i+1} \ H_{i,\ i+1}}{33000} \tag{9}$$

Adiabatic head, $H_{a_{i,\ i+1}}$, and adiabatic efficiency, $\eta_{a_{i,\ i+1}}$, are calculated by:

$$H_{\substack{a \\ i,\ i+1}} = R_i(T_i + 460.)\left(\frac{k}{k-1}\right)\left[\left(\frac{P_{i+1}}{P_i}\right)^{\frac{k-1}{k}}-1\right] \tag{10}$$

and

$$\eta_{\substack{a \\ i,\ i+1}} = \frac{H_{\substack{a \\ i,\ i+1}}}{H_{i,\ i+1}} \tag{11}$$

*Intercooler Calculations*

Air pressure loss, heat transfer rates, and condensate formation rates are calculated at each cooler. Pressure losses are assumed to be proportional to $\rho V^2$ and a separate empirical loss coefficient is used for each cooler. Heat transfer and condensate calculations are performed using an iterative procedure where the exit air temperature and condensate rates are obtained within a specified tolerance of convergence. An overall heat transfer coefficient is calculated based on the air flow rate, with provisions for a fouling resistance as input data, if desired. The log mean temperature difference method of heat transfer calculation is used.

The general calculation procedure for an intercooler is as follows using subscripts $_j$ for inlet and $_{j+1}$ for exit.

Intercooler exit pressure, $P_{j+1}$, is calculated by:

$$P_{j+1} = P_j - k_{j,\,j+1}\,\dot{m}_j\,\dot{V}_j \tag{12}$$

The unit heat transfer conductance, $U_{j,\,j+1}$, is calculated by:

$$U_{j,\,j+1} = C_{ht_{j,\,j+1}}\,\dot{m}_j, \ (C_{ht_{23}}=0.027, C_{ht_{45}}=0.042, C_{ht_{67}}=0.040) \tag{13}$$

The heat transfer resistance with clean tubes is:

$$R_{j,\,j+1} = \frac{1}{U_{j,\,j+1}} \tag{14}$$

If a fouling resistance, $R_{f_{j,j+1}}$, is used, the combined resistance, $\Sigma R_{j,\,j+1}$, is given by:

$$\Sigma R_{j,\,j+1} = R_{j,\,j+1} + R_{f_{j,\,j+1}} \tag{15}$$

The overall conductance with fouling is:

$$UA_{eff,f_{j,j+1}} = \frac{A_{eff_{j,j+1}}}{\Sigma R_{j,\,j+1}} \tag{16}$$

For calculation of heat exchanger temperatures and condensates, an iterative procedure is used. Assumed condensate rate is zero, and the assumed exit water temperature is given by:

$$T_{w_{j+1}} = T_{w_j} + 10. \tag{17}$$

Using these assumed values and other known information, the value of $X_{j,\,j+1}$ is calculated where:

$$X_{j,j+1} = \frac{UA_{eff,f_{j,j+1}}\left(\dfrac{B_1}{B_2}\right)}{60} - \dot{m}_{w_{j,j+1}}C_{P_w}(T_{w_{j+1}} - T_{w_j})$$

where:

$$B_1 = (T_j - T_{w_{j+1}}) - [\dot{m}_{w_{j,j+1}}\,C_{P_w}\,(T_{w_{j+1}} - T_{w_j}) + \dot{m}_{COND_{j,j+1}}\,h_{fg}$$
$$+ \ \dot{m}_j C_{Pa}(T_j+460)] \ / \ \dot{m}_j C_{Pa} + (T_{w_j}+460)$$

and

$$B_2 = \ln(T_j - T_{w_{j+1}}) - \ln\{\,[-\dot{m}_j C_{P_w}(T_{w_{j+1}} - T_{w_j}) + \dot{m}_{COND}\,h_{fg}$$
$$+ \ \dot{m}_j C_{Pa}(T_j+460] \ / \ \dot{m}_j C_{Pa} - (T_{w_j} + 460)\,\} \tag{18}$$

If $X_{j,\,j+1}$ is not equal to 0., new values of $T_{w_{j+1}}$ are chosen and recalculated until it approaches 0. Once $T_{w_{j+1}}$ is determined, the exit air temperature, $T_{j+1}$, is calculated by:

$$T_{j+1} = \frac{-\dot{m}_{w_{j,j+1}}C_{P_w}(T_{w_{j+1}} - T_{w_j}) + \dot{m}_{COND_{j,j+1}}\,h_{fg} + \dot{m}_j C_{Pa}(T_j+460)}{\dot{m}_j\,C_{Pa}} - 460 \tag{19}$$

Using $T_{j+1}$, the intercooler volumetric air flow rate, $\dot{V}_{j+1}$ is calculated.

$$\dot{V}_{j+1} = \frac{Z_{j+1}\,\dot{m}_j\,R_j\,(T_{j+1}460)}{144P_{j+1}} \tag{20}$$

Using $T_{j+1}$, the saturation pressure, $P_{H_2O,\,SAT_{j+1}}$, is determined from a curve. Using P, the mass flow rate of water vapor in the air is calculated:

$$\dot{m}_{H_2O,\,SAT_{j+1}} = \frac{144\,P_{H_2O,\,SAT_{j+1}}\dot{V}_{j+1}}{Z^w_{j+1}\,R_{H_2O}\,(T_{j+1}+460)} \tag{21}$$

If $\dot{m}_{H_2O,\,SAT_{j+1}}$ is less than the incoming water vapor flow rate, $\dot{m}_{H_2O}$, the amount condensed is calculated by:

$$\dot{m}_{COND_{j+1}} = \dot{m}_{H_2O,\,SAT\atop(Previous)} - \dot{m}_{H_2O,\,SAT_{j+1}} \tag{22}$$

Otherwise, no condensate is formed.

The condensate rate calculated is compared to the assumed condensate rate, and the entire set of calculations is repeated starting with the recalculation of $X_{j,\,j+1}$, using the new values for $T_{j+1}$ and $\dot{m}_{COND_{j+1}}$. The calculations are repeated until no appreciable difference is seen between the assumed and calculated values of $T_{j+1}$ and $\dot{m}_{COND_{j,j+1}}$.

*Power Calculations*

Based on motor nameplate information of 20,000 hp output with an input of 12,200 volts, 723 amps, and a power factor of 1.0, the motor's windage and friction losses are calculated to be 488 hp by using the following method:

$$\text{Input} = \frac{12,200 \times \sqrt{3} \times 723}{745.7} = 20,488\,\text{hp}$$

Output = 20,000 hp

Windage and friction losses = 20,488 − 20,000 = 488 hp

Friction losses in the gearbox and compressor bearings and air seal losses were calculated using measured motor voltage along with current and calculated gas horsepower values. The following equation was used:

Input Electrical Power = Gas Power Produced + Gear and Compressor Losses + Motor Power Losses

$$\frac{E\,I\,\sqrt{3}\,\,PF}{745.7} = \text{Gas Power + Gear and Compressor Losses}$$
$$+ \text{ Motor Losses}$$

For typical operation on 9/24/82:

$$\frac{12,200 \times 518 \times \sqrt{3} \times 1.0}{745.7} = 13,250 + \text{Gear and Compressor}$$
$$\text{Losses} + 488$$

Gear and Compressor Losses = 941 hp

Since the motor losses of 488 hp and the gear and compressor losses of 941 hp experience only slight changes at different flows and pressures, they are assumed to be constant. To calculate the total power required to the gearbox, the calculated gas horsepower is added to the losses of 941 hp.

To calculate the current to the motor, the motor losses are added to the gas horsepower along with gear and compressor losses to get the total power supplied to the motor. The current is then calculated by the following equation:

$$\text{Motor Current} = \frac{\text{Total Power Supplied} \times 745.7}{12,200 \times \sqrt{3} \times 1.0}$$

## COMPUTER PROGRAMS

### POINT Program

The POINT program makes all calculations described in the previous section for a single operating point. All system pressures, temperatures, condensate flow rates, power utilization, etc., are calculated for any selected operating condition consisting of speed, inlet guide vane position, atmospheric pressure, ambient temperature, cooling water temperature, and combination of cooling water flow rates.

The POINT program is used to make detailed calculations to provide a complete listing of all system variables from inlet to outlet of the compressor. This can be very helpful in the analysis of individual components in the system. Interstage pressure calculations determine the possible needs for relief valve protection of compressor cases or intercooler vessels. If performance of the compressor is deteriorated, the POINT program can be used to locate the source of the deficiency.

### MAP Program

The MAP program makes repeated runs of the basic POINT program for incremental changes of flow rate and guide vane position. For each incremented operating point, only the outlet pressure, temperature, power, flow rate, and guide vane position are printed. The MAP program is extremely useful for the following:

• Generating predicted performance curve data for any combination of conditions.

• Comparing effects of all variables on performance.

• Predicting minimum and maximum flow rates.

• Predicting the surge point.

• Predicting the discharge temperature for process or safety considerations.

• Conducting rerate studies considering removal of impellers, changing impeller diameters, using inlet refrigeration, using an inlet booster blower, or changing speed.

Typical performance maps generated by the MAP program are provided for a typical hot and cold day (Figures 6 and 7).
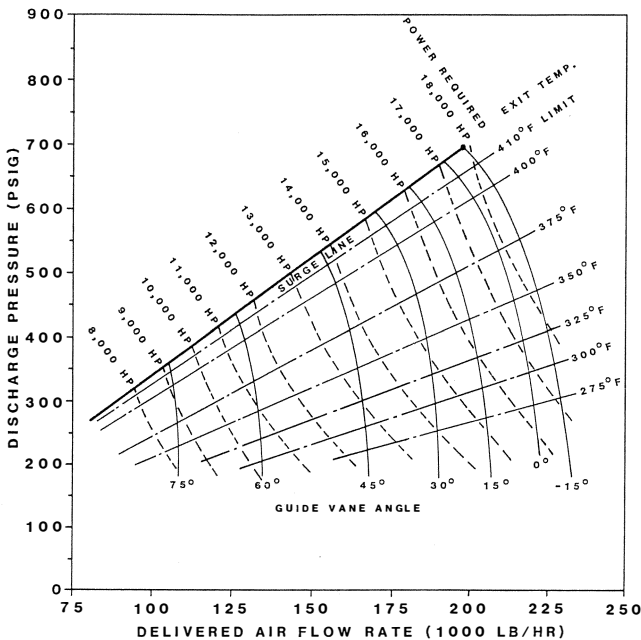


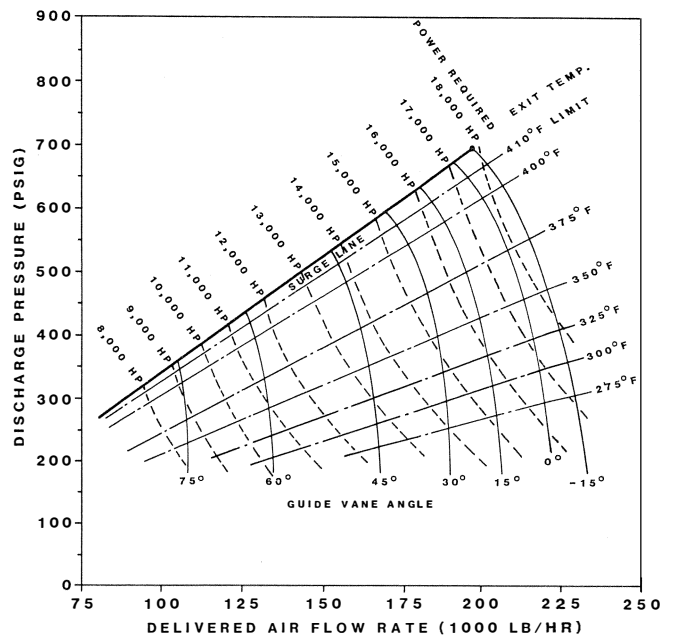Figure 6. Hot Day Performance Map.



Figure 7. Cold Day Performance Map.

Results of a comprehensive parametric study using MAP to evaluate the effects of all variables on performance on this machine are also presented in this paper.

### COMPARISON Program

A COMPARISON program is used to compare measured performance of each system component to its "good condition" performance. Deteriorated performance is detected at the earliest possible time; and information concerning the deteriorated component, the degree of deterioration, and possible causes for the deterioration are pointed out for the user. This artificial intelligence program is used in both online and offline versions.

The calculations for COMPARISON are basically the same as for POINT and MAP. Measured data (Figure 8) are used for the inlet conditions of each system component. Based on the measured inlet conditions, outlet conditions are predicted for the component based on "good condition" performance. Measured outlet conditions are compared to predicted outlet conditions, and significant differences are flagged. When deteriorated performance is noted in a compression section, decreased head coefficient and polytropic efficiency are printed along with a message suggesting possible measurement errors, excessive seal clearances, excessive impeller clearances, or dirty impellers. When deteriorated intercooler performance is noted, appropriate heat transfer parameters are printed along with a message suggesting measurement errors, low water flow rate, fouled tube side surfaces, or fouled shell side surfaces. After the initial performance evaluation, a more detailed summary of all measured and calculated variables and differences is printed for reference.

The COMPARISON program can be run frequently as an online program or occasionally as an offline program. Either way it is very useful in the following ways:

• It detects problems early. This can prevent long term inefficient operation or unexplained lost capacity. Many developing problems can be detected long before large capacity or efficiency losses are noticed.

• It can be used to monitor developing problems. If an intercooler starts to foul, for instance, the fouling rate can be reduced by changing the cooling water additives. For other developing
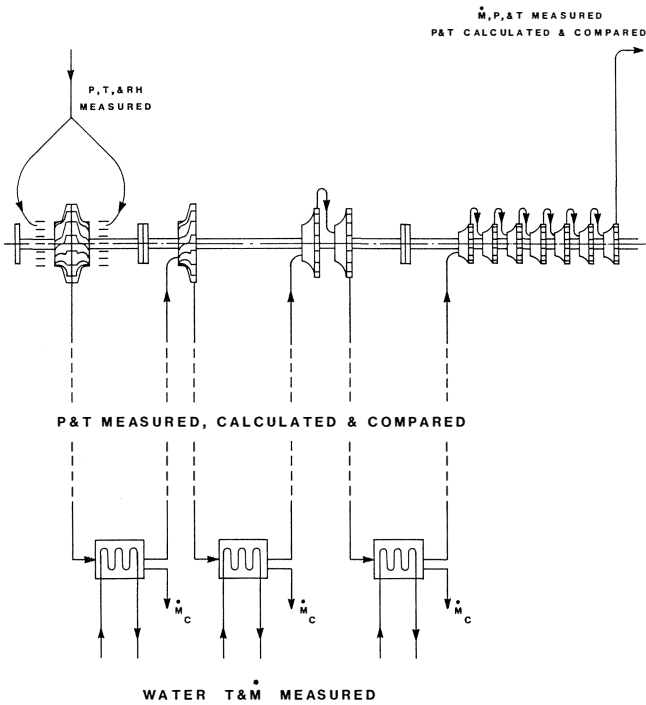
Figure 8. Model Schematic for On-Line Monitoring.

Figure 9. Effect of Speed on Operating Range.

problems, their effect can be determined in the form of efficiency and capacity decrements, allowing accurate business decisions to be made.

• It can allow an operating group to save money by minimizing costly inefficient operation.

• It can help to reduce downtime and avoid unnecessary maintenance by telling the user precisely which components are deteriorated and which are in good condition.

If only one compression section or intercooler is deteriorated, there may be no need to disassemble and inspect all of the other compression sections and intercoolers. Use of the COMPARISON program can greatly reduce lost production and maintenance costs, by eliminating all unnecessary disassembly and inspection work.

## PARAMETRIC PERFORMANCE STUDY

The development of accurate computer models of the compressor performance allowed study of the effects of all possible system parameters. The major objectives of the study were to:

• Find ways to reduce operating costs with the existing compressor system.

• Find ways to obtain maximum capacity and increased operating range with the existing compressor system.

• Find ways to reduce operating costs by modifying the existing compressor system.

• Find ways to meet new capacity and operating range requirements by modifying the existing compressor system.

*Effect of Changing Speed*

As speed increases, the envelope of possible performance greatly increases (Figure 9). On the other hand, as speed decreases, the surge line drops and the maximum open guide vane positions become part of the unstable surge range. Speeds of 8000 to 9000 rpm were evaluated because they were believed to be safe operating speeds.
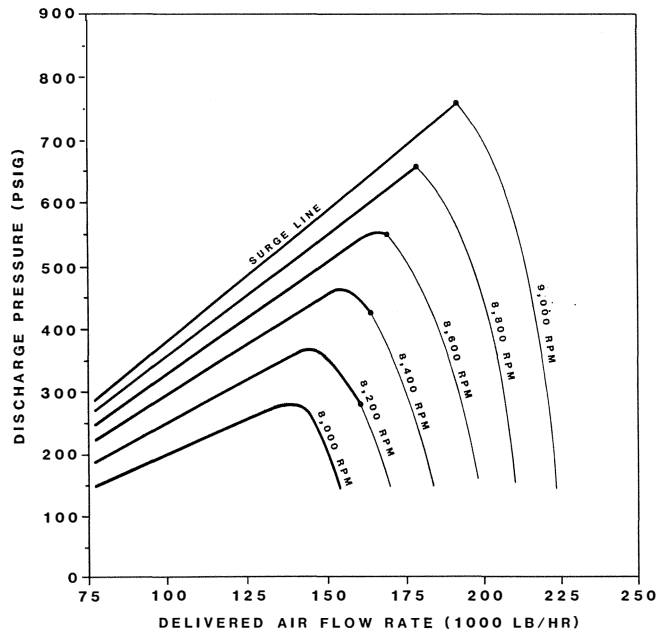
Reducing speed reduces power required and maximum flow rate of the machine. From Figure 10, it is evident that for a given operating point, reducing speed saves roughly 2.5 hp per rpm except at minimum flows where much less savings occur. As speed is reduced, the discharge temperature also is reduced so the minimum flow limit, based on temperature, is reduced. However, as speed is reduced, the surge line on the performance map drops, thereby raising the minimum flow
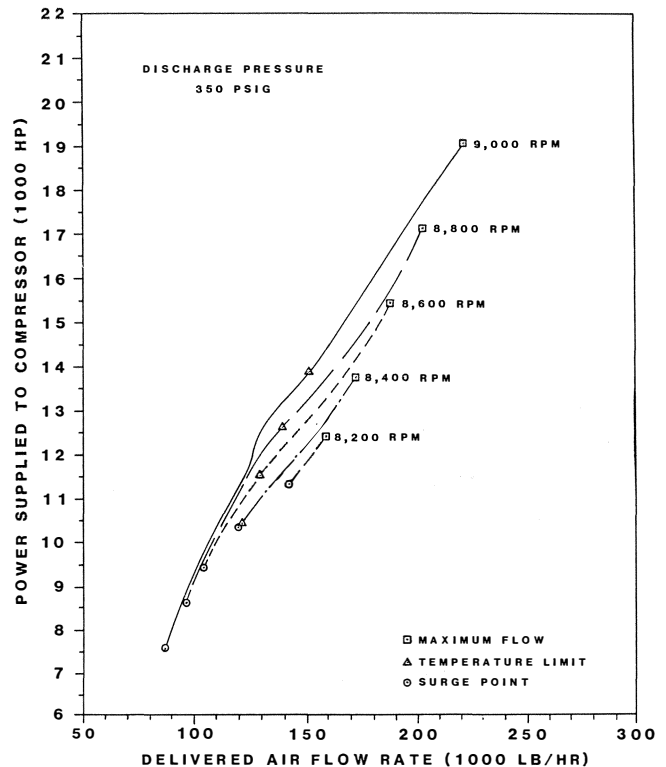
Figure 10. Effect of Speed on Power Requirements.

limit based on surge. In summary, as speed is reduced, the power requirement is reduced, but so is the maximum capacity and operating range.

### Effect of Changing the Number of Impellers

The effect of removing one, two, or three impellers from the seven stage fourth compression or booster section was evaluated for speeds of 8594 (Figures 11 and 12) and 8850 rpm. Removing impellers changes the performance map differently from reducing speed. In the map of discharge pressure versus flow rate, reducing speed moves the map down and to the left; but removing impellers only moves the map down. Removing impellers
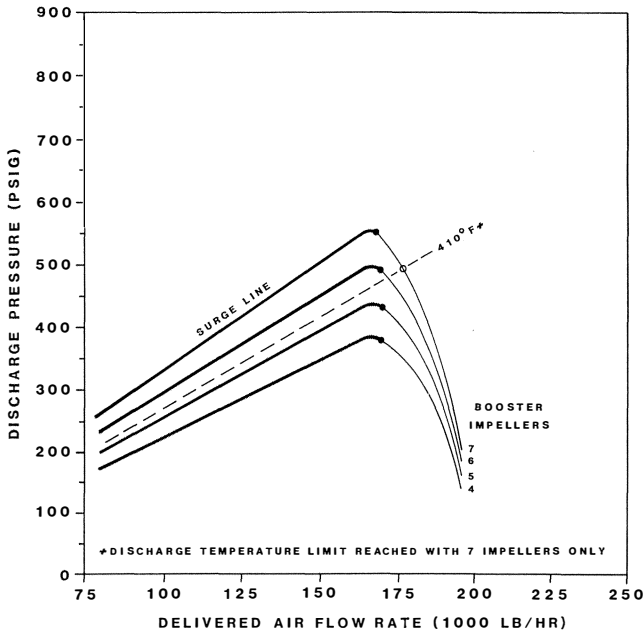


Figure 11. Effect of Number of Fourth Section Impellers on Operating Range.

reduces required power for a given operating point by roughly 290 hp per impeller. The maximum capacity is reduced very little but the surge flow increases significantly, reducing the allowable operating range.

Discharge temperature is significantly reduced possibly causing process problems if the third intercooler's water flow is not carefully controlled at a low flow rate.

### Effect of Changing Discharge Pressure

Lowering the compressor discharge pressure reduces the required power and increases the range of allowable flow rates (Figure 13). Plots of power versus flow rate for different discharge pressures indicate that power requirements can be reduced by about nine hp for each psi reduction in discharge pressure. Slight increases in maximum capacity are also gained with decreases in discharge pressure. However, as discharge pressure is decreased, the minimum allowable flow is greatly reduced, regardless of whether the maximum temperature or surge limit is considered. At minimum production rates, large power reductions are available by reducing the mininum allowable air flow in addition to discharge pressure.

### Effect of Using Inlet Refrigeration

Refrigerating the compressor inlet air increases the maximum capacity without changing the minimum flow limitations (Figures 14 and 15). Most of the increased capacity is due to the
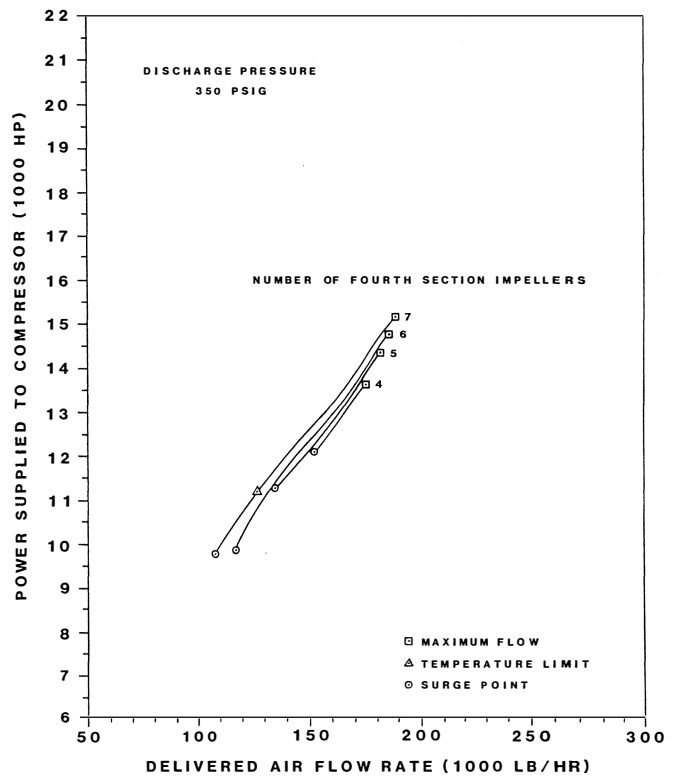


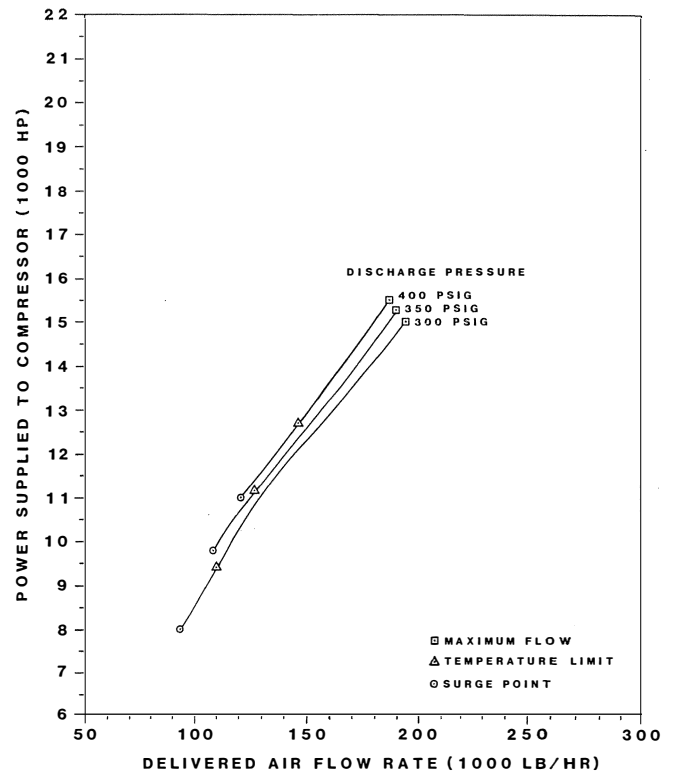Figure 12. Effect of Number of Fourth Section Impellers on Power Requirements.



Figure 13. Effect of Discharge Pressure on Power Requirements.

increased inlet air density, but it is partially caused by reduced water content of the inlet air. Power reductions are roughly 10
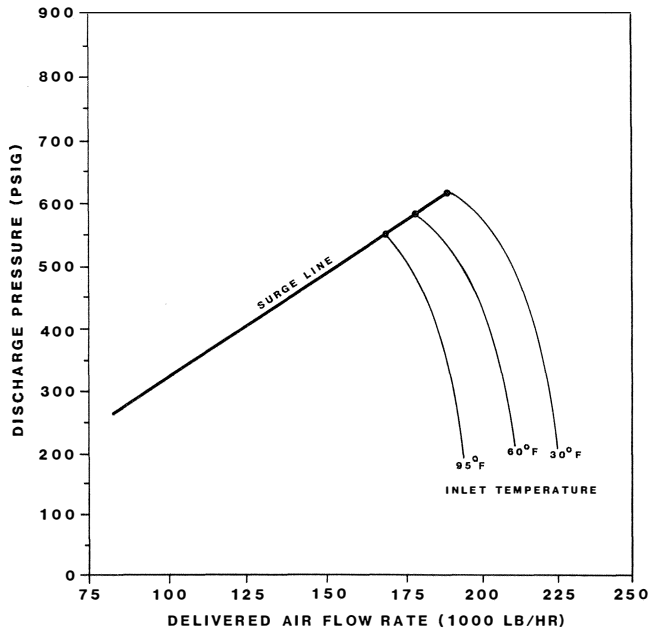
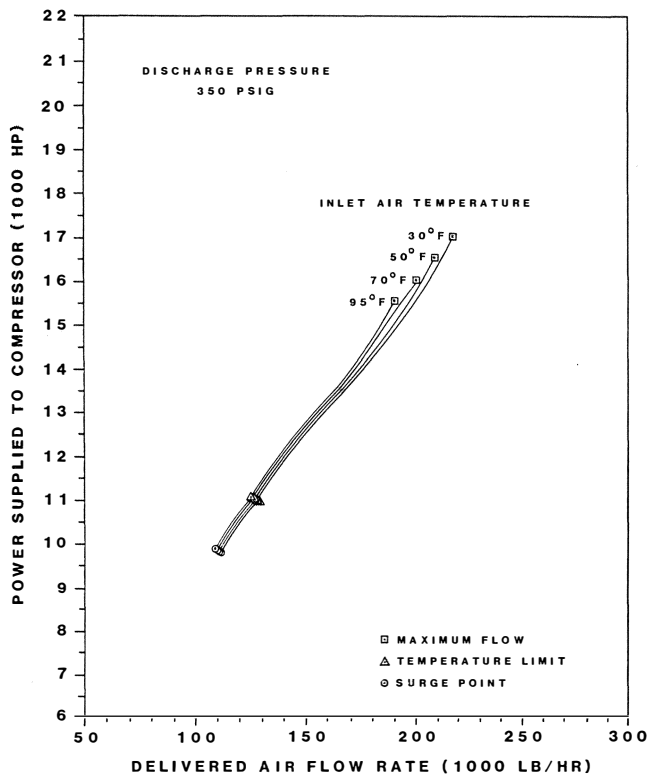Figure 14. Effect of Inlet Refrigeration on Hot Day Operating Range.



Figure 15. Effect of Inlet Refrigeration on Hot Day Power Requirements.

hp per °F of reduced inlet temperature at high air flow rates. Power reductions at lower air flow rates are less.

Since power is required to refrigerate the inlet air, the greatest benefit of inlet air cooling is increased maximum capacity with no change in low flow restrictions.

*Effect of Using an Inlet Booster Blower*

Using an inlet booster blower has a similar effect to using inlet refrigeration. The maximum air flow rate is increased with no change in low flow limitations (Figures 16 and 17). The increased capacity is obtained by increased inlet air density caused by the increased pressure from the inlet booster blower. Some reduction in power required is seen, but this would be offset by the power required to operate the inlet booster blower.
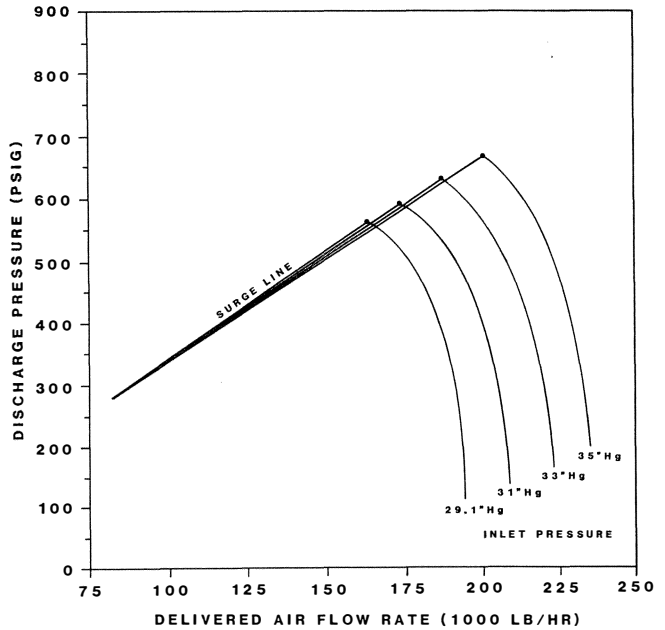


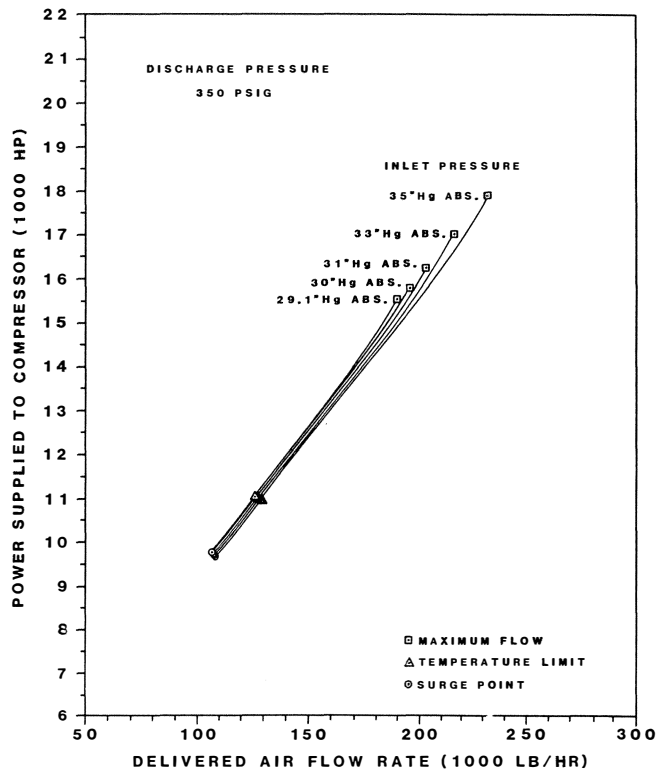Figure 16. Effect of Inlet Booster Blower on Operating Range.



Figure 17. Effect of Inlet Booster Blower Power Requirements.

One serious consideration is that the pressure at the discharge of the third compression section might slightly exceed the case design pressure under certain conditions. This requires the addition of relief valves in this section and automatic control provisions to limit inlet guide vane positions based on pressure measurements. Typical conditions leading to excessivley high pressure include an extremely cold day with extremely cold cooling water.

### Effect of Reducing Cooling Water Temperature

The effect of entire system cooling water temperature changes was determined (Figures 18 and 19) along with the effect of individual intercooler water temperature and special combinations of intercooler water temperatures. Reducing in-
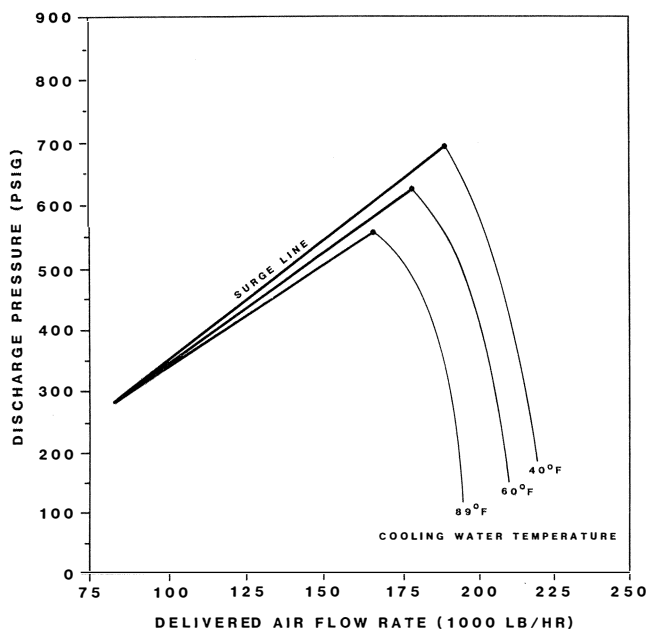


*Figure 18. Effect of Cooling Water Temperature on Operating Range.*

tercooler water temperature could be accomplished by refrigeration or by some special cooling tower arrangement for compressor cooling water. From a performance standpoint, the effect of reducing cooling water temperature is similar to that of reducing the inlet air temperature or using an inlet booster blower. It raises the maximum capacity of the compressor without greatly changing the minimum flow limitation. Actually, the minimum flow limitation will be reduced with decreased water temperatures, because discharge temperatures will also be decreased, and the surge flow will be slightly reduced. The power reduction is roughly six hp per °F of water temperature reduction. This might not be worthwhile if a refrigeration system were used, but may justify the operation of additional cooling tower fans that are already installed.

Evaluation of the effect of changing water temperature to individual coolers showed that reducing water temperature to the first intercooler increased maximum capacity most, with no effect on low flow limitations. Cooling the second intercooler water had only slight effects at high and low flows. Reducing the third intercooler water temperature allowed significant reduction of the low flow limit, by reducing the discharge temperature and the surge point, but made only small increases to the maximum capacity. All water temperature reductions decreased power requirements slightly.
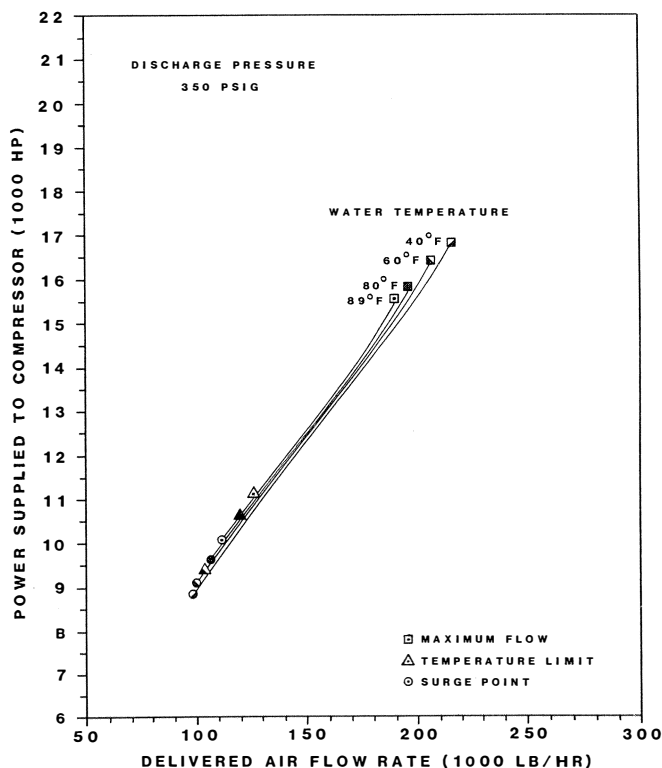


*Figure 19. Effect of Cooling Water Temperature on Power Requirements.*

The effect of reducing water temperature to the first and second intercooler and the second and third intercooler was examined. Reducing the first and second intercooler water temperature increases maximum capacity without changing minimum flow restrictions. Also the power requirement is reduced by about four hp per °F of decrease in water temperature. Reducing water temperature to the second and third intercoolers increases the maximum capacity slightly, and reduces the minimum flow restriction significantly, both based on surge and maximum discharge temperature. Power requirement is reduced by about two hp per °F of decrease in water temperature.

### Effect of Intercooler Fouling and Cooling Water Flow Rates

The effect of intercooler fouling and water flow rates on overall performance and power requirements is presented in Figures 20 through 22. The effect of changing cooling water flow rates was found to be small between normal flow and one half of normal flow (Figure 20). However, the effect of fouling was found to be extremely significant. Although fouling has very little effect on power requirements (Figure 22), maximum capacity is significantly reduced with fouling; and minimum flow restrictions due to increased discharge temperature and increased surge point flow are significantly increased. With as much as 1/16 in thick water side fouling in all intercoolers, operation at discharge pressures above 305 psig would be impossible because of excessive discharge temperatures.

The effect of individual intercooler fouling is presented in Figure 21. The fouling of each intercooler reduces the capacity of the compressor. Fouling of the second or third intercooler moves the surge line toward the operating range, increasing the minimum allowable flow rate significantly. Fouling of the third intercooler also increases the compressor discharge temperature and greatly increases the minimum allowable air flow rate.
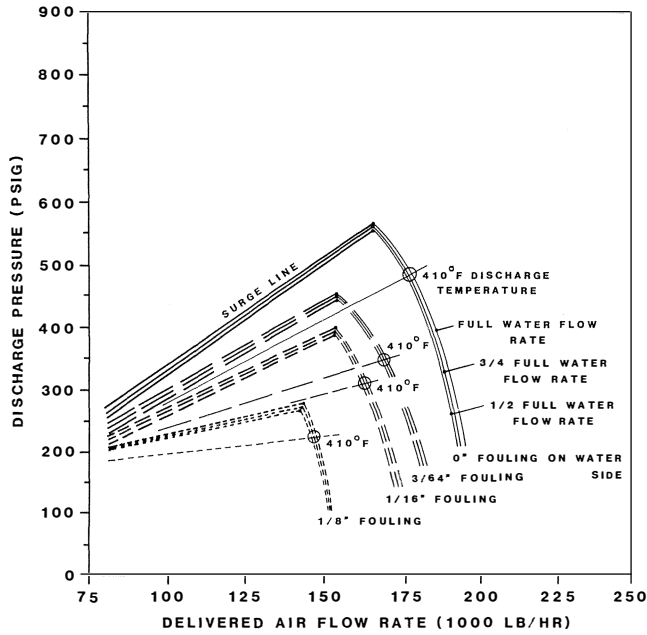
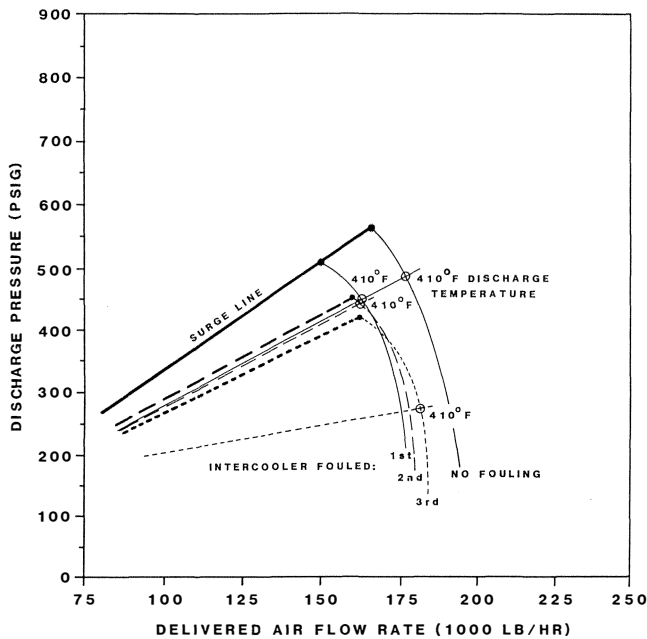Figure 20. Effect of Combined Intercooler Fouling on Operating Range.



Figure 21. Effect of Individual Intercooler Fouling on Operating Range.

Analysis shows that intercooler fouling has very little effect on power requirements, but has an extreme effect on maximum capacity, surge point, and discharge temperature. Intercooler fouling is the most important factor influencing, the range of operation.

*Effect of Inlet Filter Plugging*

Inlet filter losses up to 10 in $H_2O$ were evaluated (Figure 23). The resulting effect on maximum capacity and minimum allowable flow is small. However, at high flow rates the power requirement can be reduced by roughly 25 hp for each one inch $H_2O$ reduction in inlet filter loss.
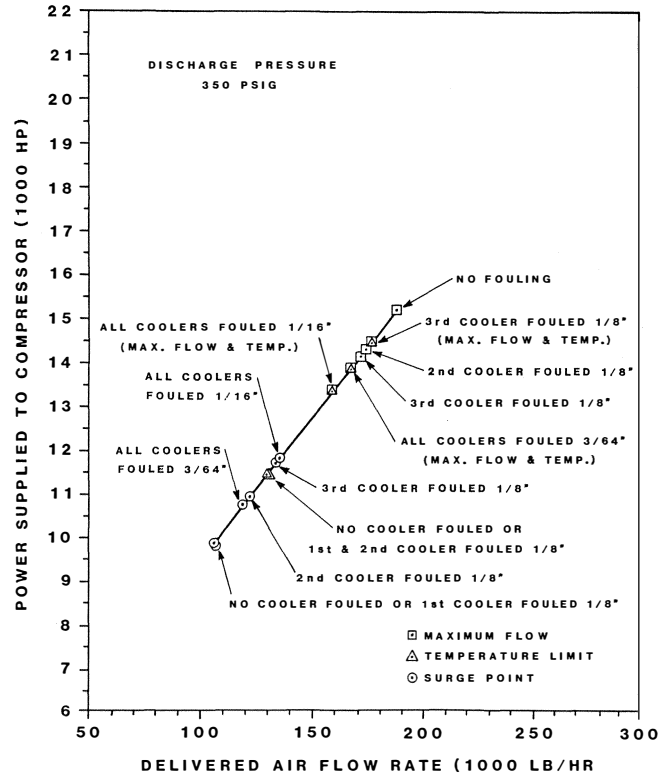


Figure 22. Effect of Intercooler Fouling on Power Requirements.
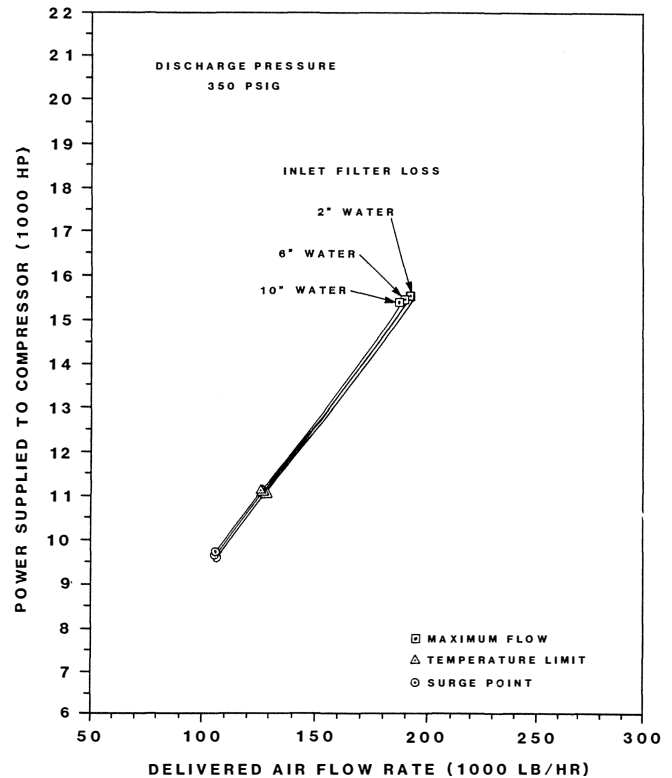


Figure 23. Effect of Inlet Filter Loss on Power Requirements.

*Effect of Relative Humidity*

The effect of relative humidity for hot and cold day operation is shown in Figure 24. As can be expected for a cold day
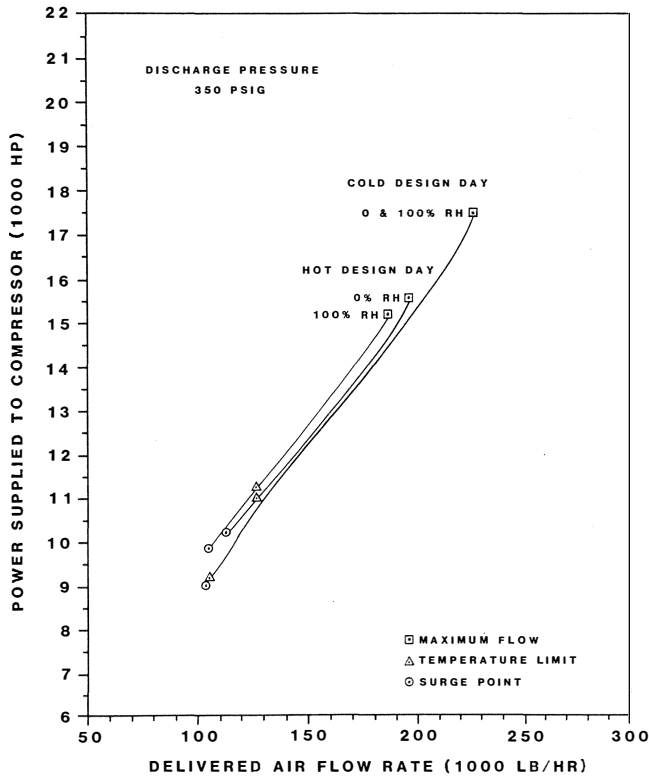
*Figure 24. Effect of Relative Humidity on Power Requirements.*

operation, the actual moisture content difference between zero percent and 100 percent relative humidity is small, so the effect on compressor performance is negligible. However, for a hot day, the moisture content is significant at 100 percent relative humidity; and compressor range and power requirements are affected. Due to the increased water content at 100 percent relative humidity, the maximum capacity is reduced by about 11,000 lb/hr, and the power requirement is increased by roughly 250 hp. Inlet air refrigeration can be used to eliminate the power increase and capacity limitations caused by compressing and removing the water vapor, and to increase compressor capacity by increasing inlet air density.

## COST AND CAPACITY CONCLUSIONS

For the existing compressor system, conclusions on ways to minimize operating costs and maintaining maximum capacity and operating range are the following:

*Ways to Minimize Operating Costs*

• Operate at the lowest practical discharge pressure. Over normal operating ranges, about nine hp is saved for each psi reduction in discharge pressure.

• Minimize the air flow rate used for antisurge controls or due to seal leakage. Roughly 75 hp can be saved for each decrease of 1000 lb./hr. in the low flow rate range.

• Minimize the cooling water temperature. Power savings are roughly six hp for each °F drop in cooling water temperature.

• Keep inlet pressure losses to a minimum by keeping clean filters. Inlet losses cost about 25 hp per in-$H_2O$.

*Ways to Maintain Maximum Capacity and Full Range of Operation*

• Avoid operation with fouled intercoolers. Intercooler fouling can greatly reduce maximum capacity and increase mini-

mum allowable flow rates. Fouling moves the surge point and the maximum temperature point to higher than normal flow rates. Even small amounts of fouling have a significant effect on capacity. The effect of fouling is limited to capacity and range with little effect on power requirements. Power is only wasted due to fouling when minimum flow conditions are desired, but cannot be reached because of excess discharge temperatures or surging.

• Keep cooling water temperatures at their lowest practical level. Decreases in water temperature decrease discharge temperatures and increase maximum capacity.

• Operate at the lowest practical discharge pressure. This greatly reduces the minimum flow restriction and slightly increases the maximum capacity.

If large changes in performance are required so that compressor system modifications are practical, the following methods should be considered to reduce operating costs, change capacity, or both.

*Ways to Reduce Operating Costs*

• Reduce compressor speed to reduce power requirements by roughly 250 hp per 100 rpm. Reducing speed also reduces the maximum capacity and increases the minimum flow restriction, based on the surge point. Discharge temperature is decreased by reducing speed, so the minimum flow restriction, based on the discharge temperature limit, is decreased. However, high flow discharge temperatures may be decreased enough to cause process problems, unless third intercooler water flow rates are appropriately reduced.

• Remove booster impellers to reduce power requirements by roughly 290 hp per impeller removed in the present case. When impellers are removed, the maximum capacity is only slightly reduced; but, the minimum operating flow based on surge is increased. The minimum operating flow based on discharge temperature is decreased, and discharge temperatures at high rates may be lowered enough to require reduced water flow to the third intercooler. A combination of speed change and impeller removal may offer advantages in meeting some requirements.

*Ways to Increase Maximum Capacity and Range of Operation*

• Increase speed to increase maximum capacity and decrease low flow limit based on surging. Increasing speed increases the low flow limit based on discharge temperature, and raises power requirements. This would generally be used to obtain a higher maximum capacity.

• Refrigerate the inlet air to increase the maximum capacity without changing low flow restrictions. Capacity is increased by increasing inlet density and removing water vapor from the inlet air stream. Compressor power requirements are decreased, but the savings are countered by the cost of installing and operating the refrigeration system.

• Install an inlet booster blower to increase maximum capacity without changing minimum flow restrictions. In this case, the compressor power requirements are reduced, but savings are countered by installation and operation costs of the inlet booster blower.

• Reduce cooling water temperature through a higher cooling tower capacity or an individual refrigeration system to increase maximum capacity and reduce minimum flow restrictions based on both surge and temperature. Cooling water temperature reduction slightly decreases power requirements, but savings are offset by installation and operation costs of new water cooling equipment.

## PERFORMANCE MONITORING CONCLUSIONS

The COMPARISON program is useful for both on and offline monitoring of compressor component performance, to provide early warnings of deteriorating intercooler or compressor stage performance. The most common problems that may be detected are decreased intercooler performance due to buildup formation in and plugging of tubes, or fouling of shell side fins. Other problems that may be detected are lost compression stage performance or efficiency due to damaged labyrinth seals, excessive impeller clearances, or excessive buildup on impellers.

When intercooler problems are detected early, it may be possible to change cooling water treatment to avoid the expense of an early shutdown for tube cleaning. When compression problems are detected, it is advantageous to know that the problem has occurred and exactly where it exists. Knowing about the problem early could possibly help avoid a very serious mechanical failure. Knowing precisely where the compression problem exists will minimize production losses from downtime and minimize maintenance costs by avoiding disassembly of more compression sections than are actually required. Knowledge of the lost performance and efficiency will provide data for calculation of excess power usage and loss of compressor capacity, if an early shutdown for corrective action is undesirable.

## BOOSTER BLOWER CONCEPT

One of the most interesting possibilities coming from the parametric study is the use of an inlet booster blower (Figure 25). The booster blower could be turned on when additional capacity is needed, and turned off at all other times. An inlet booster blower and aftercooler combination capable of producing 120 in of water discharge pressure was shown to increase maximum capacity on a 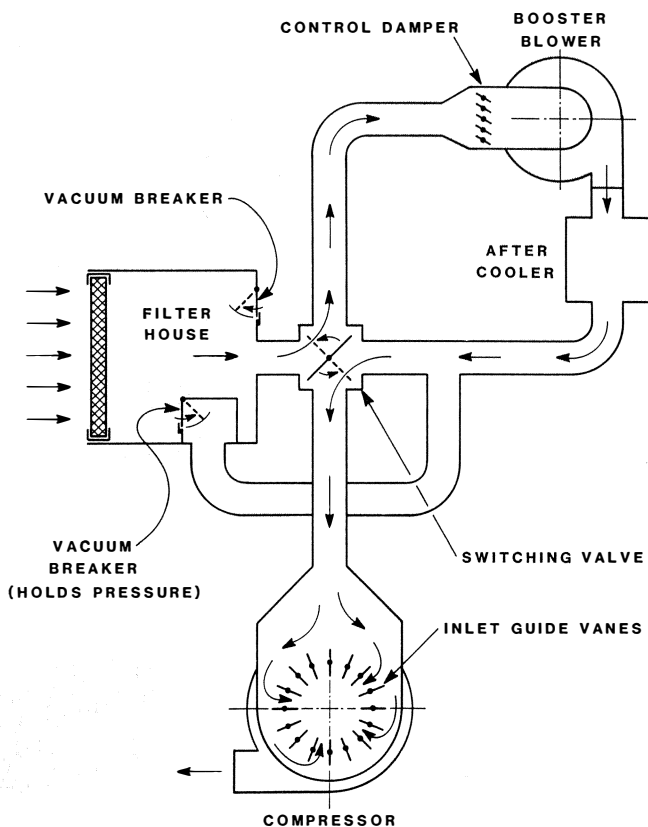hot day by up to 33 percent, and on a cold day by up to 22 percent. This compares favorably to a speed increase up to the maximum speed of 8850 rpm which produces a capacity increase of about ten percent for all ambient conditions. Problems with the use of an inlet booster blower are possible excessive pressures at the discharge of the third main compression section and "stonewall" conditions in the fourth main compression section.

The possible excess pressure problem in the discharge of the third main compression section could be solved by installing relief valves for safety, and an inlet booster blower spin damper override control system to maintain the pressure safely below the relief valve set point.

The possible "stonewall" effects in the fourth main compression section could be minimized by raising discharge pressure. This reduces the Mach number by decreasing the velocities and increasing the temperatures. Dangers of "stonewall" operation are minimized by extensive data and experience operating in the vicinity of "stonewall" on cold days with reduced back pressure.

## NOMENCLATURE

| | |
|---|---|
| $A_{eff}$ | Effective surface area of intercooler, $ft^2$ |
| $C_{ht}$ | Imperical conductance constant for intercooler, Btu min/hr °F$lb_m$ |
| $C_{pa}$ | Specific heat of air at constant pressure, Btu/$lb_m$°F |
| $C_{pw}$ | Specific heat of water at constant pressure, Btu/$lb_m$°F |
| E | Potential to motor, Volts |
| GHP | Gas power, hp |
| g | Gravitational constant, $ft/sec^2$ |
| H | Actual head, ft |
| $H_a$ | Adiabatic head, ft |
| $H_p$ | Polytropic head, ft |
| $h_{fg}$ | Latent heat of vaporization of water, Btu/$lb_m$ |
| I | Current, Amps |
| K | Pressure loss factor for intercooler, $lb_f$ min/$lb_m$ $ft^3 in^2$ |
| k | Ratio of specific heats for air |
| $\dot{m}$ | Air flow rate, $lb_m$/min |
| $\dot{m}_{COND}$ | Condensate flow rate, $lb_m$/min |
| $\dot{m}_{H2O}$ | Water vapor flow rate, $lb_m$/min |
| $\dot{m}_w$ | Cooling water flow rate, $lb_m$/min |
| N | Compressor running speed, rpm |
| n | Polytropic coefficient |
| P | Pressure, $lb_f/in^2$ absolute |
| $P_{H2O}$ | Partial pressure of water vapor, $lb_f /in^2$ absolute |
| PF | Power factor |
| $P_f$ | Power loss to friction, hp |
| $P_{tghp}$ | Total gas power, hp |
| $P_{thp}$ | Total power, hp |
| R | Thermal resistance, hr $ft^2$°F/Btu |
| $R_{air}$ | Gas constant for air, ft $lb_f$/$lb_m$°F |
| $R_f$ | Fouling resistance for intercooler, hr$ft^2$°F/Btu |
| $R_{H2O}$ | Gas constant for water vapor, ft $lb_f$/$lb_m$°F |
| T | Temperature, °F |
| $T_w$ | Water temperature, °F |
| U | Unit thermal conductance, Btu/hr$ft^2$°F |
| $\overset{\circ}{V}$ | Volumetric flow rate, $ft^3$/min |
| v | Impeller tip speed, ft/sec |
| X | Intercooler variable for temperature calculations |
| Z | Compressibility factor |
| $\eta_a$ | Adiabatic efficiency |



*Figure 25. Inlet Booster Blower Concept.*

$\eta_P$      Polytropic efficiency

$\mu_P$      Polytropic head coefficient

$\rho$      Density, $lb_m/ft^3$

*General Subscripts*

i      Compression calculation inlet location designation

i+1      Compression calculation outlet location designation

j      Intercooler calculation inlet location designation

j+1      Intercooler calculation outlet location designation