CENTAC®

Utility Industry Handbook

INGERSOLL-RAND®

AIR COMPRESSORS

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Utility Industry Handbook
About The Author

W. R. "Bill" Buckley joined Ingersoll-Rand in 1974 after being employed by the Diamond Power Specialty Corporation, a subsidiary of the Babcock and Wilcox Corpration for a period of 23 1/2 years. Over that period of time, Bill gained a wealth of experience and became well acquainted with numerous engineering and production managers throughout the electric power generation industry.

Bill joined Ingersoll-Rand at a time in which the electric power generation industry was experiencing a domestic growth of approximately 6% per year in additional electric generating capacity. This added capacity resulted in substantial market potential for high pressure centrifugal compressors applied for boiler sootblowing service as well as for 100 to 125 PSIG compressors of lesser capacities applied for station and instrument air service. Within a period of approximately 2 years of Bill's marketing guidance and personal support of the Direct Sales team, the Division became established as the leading supplier of sootblowing air compressors within the power generation market, recording a dominate market share of approximately 95%.

While the annual growth of the domestic power generation market has diminished in recent years, the decade of the 70's and early 1980's saw the growth of a substantial population of Ingersoll-Rand centrifugal air compressors within this industry. The Company continues to be recognized as the leading supplier within this segment of the market.

Bill retired in August of 1990 after 16 years of service to the Company. Since his retirement he has maintained a close relationship with the Division and continues to lend support to the sales and marketing activities of the Division, within the worldwide electric power generation market.

Date of Issue: 4-Oct-1996

Utility Industry Handbook
Preface

This handbook is for use by Ingersoll-Rand representatives and others concerned with the application of air compressors within the Utility Industry.

The Utility Industry is one in which Ingersoll-Rand has enjoyed considerable success over the years. Our company is well-known within the industry, not only for the products and representation of the Air Compressor Group but also for boiler feedwater and recirculation pumps supplied by other divisions of the Corporation. During the early 1970s through the early 1980s when the construction of new facilities within the industry was at a peak, Ingersoll-Rand grew to dominance of the market, particularly with centrifugal compressors applied for air sootblowing. This position of dominance in the marketplace can be attributed both to knowledgeable representatives who have an awareness of the Utility Industry's needs and due to the company's willingness to invest in the development of quality products to meet the demands of that market. Ingersoll-Rand is well-positioned to meet the needs of this market as it experiences future growth on a worldwide basis.

There are a great many variations of steam generating boiler designs and applications, as well as many complexities that cannot be explained or understood without an in-depth knowledge of thermo-dynamics, fuel chemistries, and other technical disciplines. Taking this into account, our intention is to deal only with the specific type of boilers, boiler fuels, and applications that Ingersoll-Rand is most likely to be concerned with in our attempts to market products within the Utility Industry. Applications of compressed air to supply Station and Instrument Air, air for conveying dry ash, or, for use in connection with Flue Gas De-sulphurization Systems are recognized as important segments of this market. However, this handbook is prepared to deal primarily with the application of compressors to supply air to boiler sootblowing systems.

While it is not reasonable to expect the Ingersoll-Rand representative to demonstrate an in-depth knowledge in this area, it is reasonable to expect him to be adequately informed to be conversant and to appear credible to key buying influences within this marketplace. Experience over the years has proven that the supplier who demonstrates an understanding of the client's process and his most significant application concerns will enjoy a definite advantage over his less knowledgeable competitor in the client's evaluation of suppliers.

We trust the information offered in the following is stated in a meaningful and understandable manner that will prove to be beneficial to the representatives of the Ingersoll-Rand Company.

Date of Issue: 4-Oct-1996

Table of Contents

Date of Issue: 4-Oct-1996

Steam Power Generation

Steam generation boilers have many applications ranging from generation of steam used for heating or for supplying steam to a process of one kind or the other to that of generating steam to drive an electric power turbine generator. In an effort to avoid confusion, we will concern ourselves only with the type of boiler commonly applied to generate steam to drive a turbine to generate electricity. In this instance, the boiler is designed to evaporate or generate steam at a design pressure and temperature that is required to drive the selected electric power turbine generator. Such a boiler is therefore defined as, or "rated" as, one to generate steam at a given design pressure and temperature and at an "evaporation rate" of "X" pounds per hour.

At this point, it seems appropriate to clear up a commonly misused term to define the size or rating of a steam generating boiler. It is often said that a particular boiler is designed as a "200 or 600 MW" unit. It is inaccurate to so describe the boiler as only the electric power turbine generator is rated in terms of megawatts (MW) of power. The boiler capacity is always stated in terms of "evaporation rate" in pounds per hour of steam. Such boilers are designed over a wide range of steam pressures and temperatures, evaporation rates, fuel burning capabilities, and design configurations. For example, a lesser size electric utility boiler for a 150 MW generating unit would be required to evaporate steam at the rate of approximately 670,000 #/hr. The pressure and temperature are dependent upon the design of the turbine generator and might range from 600 PSIA (41.37 BARa) up to 4500 PSIA(310.34 BARa) with steam temperatures up to approximately 1000^{0}F (537.7^{0}C).

Boilers designed for operation within the range of pressures up to 3200 PSIA (220.64 BARa) are normally referred to as "Radiant-Drum Type." Boilers designed for higher pressures or those above the "critical pressure" of 3208 PSIA (221.24 BARa) are normally referred to as "universal pressure" or "universal pressure-once through" boilers. By way of comparison to the 150 MW unit mentioned earlier, Universal Pressure or Supercritical boilers are in operation evaporating as much as 9,775,000 #/hr. of steam at pressures up to 3845 PSIA(265.11 BARa). These are the largest boilers ever built and are designed to supply steam to a 1300 MW turbine generator.

Steam generating boilers are also designed to be fired by; or to burn a wide range of fuels that might consist of anything that has heat value expressed as "BTU's/pound of fuel." For our purpose, the most commonly referred to fuels are classified as follows:

Fossil Fuels:	Coal (normally fired as pulverized coal)
	Oil (normally a residual oil)
	Natural Gas.
Waste Heat Fuels:	These fuels can generally be defined as the by-product of any process from which heat can be recovered for the purpose of generating steam in a heat recovery boiler.

Examples of commonplace waste heat or heat recovery boilers might be an Incinerator Boiler designed to burn refuse to generate steam for any purpose or a waste heat boiler in a steel mill designed to recover heat from a blast furnace. Still another widely used waste heat boiler found in the Pulp and Paper industry is the Chemical Recovery Boiler. These boilers burn a by-product of the Kraft paper- making process known as "black liquor," and they represent a huge market for sootblowing equipment but not for compressed air for the sootblowing systems. Like most all

Date of Issue: 4-Oct-1996

other types of waste heat boilers, the fuel is considered to be cost free and; therefore, the steam generated is so low in cost that it is virtually impossible to justify the use of compressed air for sootblowing service on such boilers.

Of the fossil fuel fired boilers mentioned earlier, only the PC (pulverized coal) and oil fired boilers require sootblowing systems. Natural gas burns cleanly and produces no by-product of combustion that requires cleaning with sootblowers.

With the foregoing taken into account, we can narrow our concerns to the basics of design and application of pulverized coal (hereafter referred to as PC) fired and oil fired, electric power generation boilers designed to supply steam to drive the electric power turbine generator.

Although there are a considerable number of boiler manufacturers in the United States, there are only four who compete in the electric power generation market:

Combustion Engineering, Inc.

Babcock and Wilcox Company

Foster Wheeler Corporation

Riley Stoker Corporation.

The majority of boiler manufacturers in Europe and the Far East countries are either subsidiary companies, licensees, or joint venture companies with either Babcock and Wilcox or Combustion Engineering.

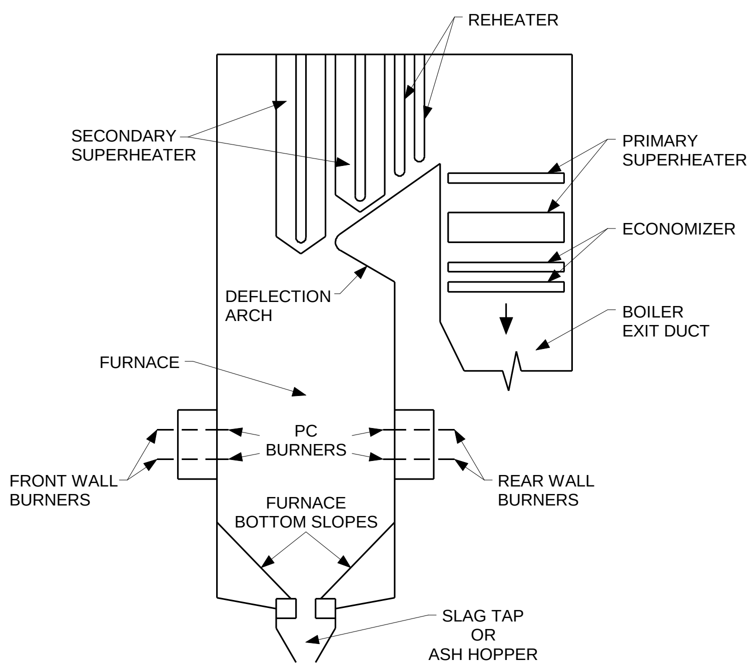


Figure 1-1
Generic boiler design

While the exact configuration and arrangement of the heating surface varies from one manufacturer to another, there is sufficient consistency to permit us to use a typical outline sketch of a generic boiler design, as illustrated in Figure 1-1, to help us understand the general configuration. The major sections of the boiler as shown in Figure 1-1 are identified as follows:

A Furnace

A-1 Furnace Bottom Slopes

A-2 Slag Tap or Ash Hopper

A-3 Deflection Arch

B PC Burners

B-1 Rear Wall Burners

B-2 Front Wall Burners

C Secondary Superheater (or Final Superheater)

D Reheater

E Primary Superheater

F Economizer

G Boiler Exit Duct

All of the boiler types with which we are concerned are known as “water tube” boilers. This simply means that the inner walls of the furnace are made up of tubes that might vary in diameter from 1 to 2.5”(25.4 to 63.5mm) through which boiler feed water is circulated. The remaining sections of the boiler are also made up of boiler tubes to complete a water/steam flow cycle, sometimes referred to as the vapor cycle of the boiler.

Pulverized coal, or other fuel, fired at the burner level of the furnace generates or releases hot gases into the furnace enclosure that makes up what is commonly referred to as the flue gas stream. The flow of these gases from the furnace are normally controlled by the application of either a “forced draft” or “induced draft” fan, and sometimes by both, depending upon the design of the boiler. As the flue gases flow over the furnace walltubes and through the balance of the boiler, heat is transferred from the “gas side” to the “water side” of the tubes and steam is generated. This is by no means as simple as it might appear. The designer’s selection of the furnace volume and total area of water tube heating surface to be designed into the furnace presents an extremely complex design consideration. The primary objective is to select the heat absorption surface in the furnace so as to control the furnace exit gas temperature, taking all of the variations of fuel chemistry into account. As you progress with your understanding of how the boiler evaporates steam, you will come to appreciate the importance of controlling the furnace exit gas temperature.

In the case of the boiler outlined in Figure 1-1, the secondary superheater and reheater sections are made up of tubes suspended vertically from headers at the top of the boiler. These vertical sections are generally referred to as “pendent” sections, which is descriptive of the manner in which they are supported. It is also commonplace to refer to these vertical tube loops as “platens” or a platen section of the boiler. The primary superheater and economizer consist of horizontal banks of boiler tubes installed in what is generally called the “down-flow section” or back end of the boiler. The term “down-flow” refers to the direction of flue gas flow in the back end of the boiler.

The pendent tube loops that make up the secondary superheater and reheat section are normally widely spaced with tubes located on something like 16” or 24”(406.4 to 609.6mm) centers. This minimizes the possibility of slag build-up bridging from one loop of tubes to another and plugging the gas flow path through the superheater. Farther back in the reheat section, the tube spacing becomes a little closer. As the gases turn into the downflow section and into the primary superheater, it is normally expected that the flue gas temperature will have dropped to something close to 1450 or 1550^{0}F (787.7 or 815.5^{0}C), and the spacing of tubes in the horizontal tube bank becomes much closer. By the time the ash particles entrained in the gas stream reach the downflow section of the boiler, it is anticipated that the ash carry over will be in a solid state, depending upon the ash content and characteristics of the fuel. It is, therefore, important that the designer arrange the spacing of tubes in such a manner that the velocity of the gas stream is controlled to lessen the possibility of causing boiler tube erosion as well as plugging of the gas flow passage between rows of tubes.

Oil fired boilers also require that careful consideration be given to the arrangement of heating surfaces and prevention of serious formations of oil ash and slag on the gas side of the boiler tubes. Sootblowing equipment is required on oil fired boilers except in the furnace section. The major consideration given to the analysis of fuel oil is that of constituents which, at elevated temperatures, will cause serious boiler tube corrosion. This will be discussed at greater length as we consider the application of sootblowing equipment.

Given an understanding of the flow of flue gases over the gas side of the boiler tubes, we must now consider the flow of boiler feedwater inside the boiler tubes, which is converted to steam as it flows in a direction counter to the gas flow. Figure 1-2 is a simplified flow diagram illustrating the flow of boiler feedwater and steam to the turbine generator. This flow pattern is commonly referred to as the “vapor cycle” of the boiler.

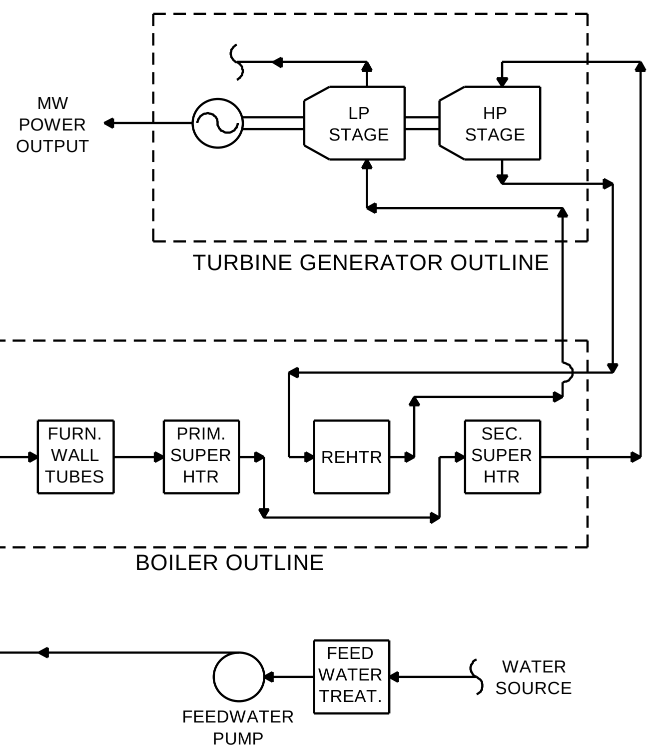


Figure 1-2
Typical vapor cycle for reheat boiler

Raw water from surface or subsurface sources invariably contains some degree of troublesome scale-forming materials, free oxygen, and sometimes acids. It is, therefore, essential that carefully designed water treatment be provided to remove these impurities. The water treatment equipment normally consists of a de-mineralizer and rather elaborate arrangements of controls to regulate the chemistry of the water. The water treatment will sometimes include heaters of some type, as the temperature of the feedwater entering the economizer should be high enough to prevent condensation and acid attack on the gas side of the tubes. The objective is to keep the metal temperature of the tubes above the dew point to prevent corrosion.

The boiler feedwater pump, required for forced circulation and super-critical boilers, is normally a large steam turbine driven pump that is frequently designed to use steam from the high pressure turbine exhaust. In Figure 1-2, we have shown the high pressure turbine exhaust steam being returned to the reheat section of the boiler. However, if this were to represent a unit with a turbine driven feedwater pump, we would show a portion of the steam returned to the reheat section being extracted and delivered to the turbine driven pump.

At the boiler, the feedwater enters the cycle at the economizer, which serves the purpose of a heat recovery device to transfer heat from the gas stream to pre-heat the water before it enters the walltube circuit in the furnace. In this way, the economizer improves the efficiency of the furnace and lowers the boiler exit gas temperature to an acceptable level.

Within the furnace walltubes, the water is elevated in pressure and temperature to the point of saturation. This simply means that the water has been heated to reach its “boiling point” for the pressure to which it is elevated in the furnace portion of the steam cycle. While the term “boiling point” is most frequently used to identify conditions at standard atmospheric pressure, where water boils at 212^0F (100^0C.), we must bear in mind that the boiling point, or saturation temperature, is a function of pressure; as the pressure increases, the boiling point or saturation temperature does likewise.

As the saturated vapor enters the primary superheater, added heat is transferred from the gas stream, and the steam is heated beyond its saturation point, and it is said to be superheated. The extent to which the steam is superheated within the primary superheater is a function of the total area of heat transfer surface designed into the section and of the temperature of the gas stream at the location of the primary superheater tubes.

The cycle continues by leaving the primary superheater and entering the secondary superheater, sometimes referred to as the final superheater. The secondary superheater, being exposed to the highest gas temperatures leaving the furnace, adds heat to the steam to elevate it to the design pressure and temperature required at the superheater outlet.

The reheat section, as noted earlier, supports a separate steam cycle to recover heat from the gas stream to re-heat steam from the first stage or high pressure turbine exhaust and return it to a lower pressure stage of the turbine generator.

The vapor cycle of a boiler can be expected to vary to accommodate various arrangements of boiler heating surfaces and types of turbine generators. However, the illustration in Figure 1-2 and comments in the foregoing will provide a basis for understanding more complex cycles.

While the steam cycle illustrated in Figure 1-2 shows the total of the steam evaporated by the boiler being supplied to the turbine generator to generate electricity, in reality, steam is extracted from a number of intermediate points to be used for purposes other than generating power. These other uses, such as supplying steam for a boiler sootblowing system, are commonly referred to as “indirect” uses of steam.

Generally speaking, the design and application of the boiler will allow approximately 10% of the total steam evaporation rate to be applied for indirect uses. This fact should be noted as it may become important to you in future discussions regarding the use of air versus steam for a particular category of market potential.

Combustion of the pulverized coal takes place in the lower furnace. The burners (B-1 and B-2) illustrated in Figure 1-1 are typical “horizontal” burners installed at two or more elevations on both the front and rear wall of the furnace. In some cases, the horizontal burners are installed on the rear wall only; however, all boiler manufacturers except Combustion Engineering, Inc. use some arrangement of horizontal burners. Combustion Engineering and their licensees use several elevations of burners installed at each of four corners of the furnace. These are referred to as “tilting, tangential” burners. In the case of both arrangements of burners, initial ignition is in the form of oil fired igniter torches at each burner. Once initial ignition is established, the pulverized coal is atomized into the furnace where it is ignited and made to burn in suspension. Combustion of the fuel is carefully controlled by sophisticated burner controls that will vary the air to fuel ratio and other factors that contribute to maintenance of the most efficient firing rate. Even under the best of circumstances, combustion is always incomplete, and by-products of combustion become entrained in the flue gas stream that flows upward through the furnace, through all of the boiler sections and out the gas exit. From here, the gases may go directly to the stack or through a heat recovery device called an airpreheater before it is dispersed to the atmosphere from the stack. The “airpreheater,” sometimes identified as a regenerative airpreheater, or just an airheater, is a heat recovery device independent of the boiler through which air flows in a direction counter to the flow of the flue gases. This serves to preheat air that is supplied to the burners to support and control combustion. The added heat recovery device between the boiler and the stack serves to reduce the temperature of gases emitted from the stack making the effluent more environmentally acceptable.

With a mental picture of the pulverized coal being fired in the furnace and heat being transferred at the desired rate from the gas side to the water side of the boiler tubes, you should have a fair understanding of how the boiler delivers steam at the design pressure and temperature to the turbine generator.

Now, the objective is to maintain the operation of the boiler at the desired level of steam flow and pressure taking into account the reduction in heat transfer that takes place in the furnace and boiler sections as a result of the ash entrained in the gas stream being deposited on the heat transfer surfaces. With an extensive knowledge of the fuel chemistry and its historical fouling characteristics, the designer will size the furnace to contain sufficient volume and will select the arrangement and total area of heating surface required to maintain design conditions with a predicted fouling factor taken into account. The major influence upon this design consideration is the analysis of the specified fuel.

While boiler designers must concern themselves with nearly all aspects of the fuel analysis, it is really not necessary to develop an understanding of the total analysis in order to discuss intelligently the influence of ash depositions and application of boiler cleaning systems with various types of fuel.

Considering only the nature of the boiler deposits that we might expect from a particular fuel fired under certain operating conditions, we are primarily interested in the coal’s ash content, sulfur content, both expressed as a dry basis percentage, and the “fusibility” of the ash. The ash fusibility is normally stated at three levels of temperature, each of which is defined as follows:

Initial deformation temperature----- temperature at which the ash particle first begins to change form.

Softening temperature---------------- temperature at which the ash particle has fused and lost its original form. (Commonly called “ash fusion temperature”)

Fluid temperature---------------------- temperature at which the fluid mass has taken on the form of a running slag.

Date of Issue: 4-Oct-1996

With these and other contents of the fuel analysis taken into account, the boiler designer must determine just how much area of heating surface will be required in each section of the boiler to meet the design pressure and temperature at which steam is to be delivered to the turbine generator. This selection of heating surface involves a complex application of thermo-dynamics, which is further complicated by the fact that the heat absorption rate in each section of the boiler can be expected to vary depending upon the “fouling factor” of the fuel. The fouling factor is simply an index applied to a particular type of coal with which an estimate is made of the rate at which the fuel’s by-products of combustion will deposit slag or ash on the heat transfer surface to reduce the rate of heat transfer. This same analysis, or estimate of the fouling factor of the fuel, also serves to define the location as well as the operating pressure and flow of air or steam to be recommended for application of sootblowing units in various sections of the boiler.

Within the following chapter, we will discuss the various types of sootblowing units most commonly applied to maintain an acceptable rate of heat transfer within each of the various sections of the boiler.

Boiler Cleaning Systems

For as long as boilers have been fired with fuel oil of any kind to generate steam for heating or other purposes, sootblowers have been essential to the boiler’s availability and operating efficiency. In those very early years of boiler operation, the boiler cleaning equipment was aptly named “sootblower,” as the by-product of firing the fuel was indeed a mechanical carry over of soot. A very thin layer of soot deposited on the boiler tubes would result in a dramatic reduction of heat transfer. It therefore became necessary to develop a means of removing the soot from the heating surface while the boiler continued operating. As a result, the sootblower came into being.

The evolution of boiler applications in the years that followed demanded the design of larger and larger boilers that could burn lower grades and less costly forms of fuel oil as well as other fuels, primarily, coal. As these applications evolved, the sootblowers became less appropriately named in that the material deposited on the boiler tubes was no longer a soot-like deposit. The by-product of combustion was now better defined as an ash carry over, which in the higher temperature sections of the boiler turned to a molten or running slag. But, if for nothing more than the sake of tradition, the term “sootblower” prevails and is commonly used throughout the Power Generating Industry.

Early designs of coal fired boilers burned coal in the furnace section in something of a char bed spread over the furnace bottom on one of several available types of “stokers.” The most widely used were known as “spreader stokers” or “chain-grate stokers.” Even the better grades of coal burned in this manner produced a great deal of ash carry over into the boiler, which demanded more and more attention to the design and application of the boiler cleaning equipment.

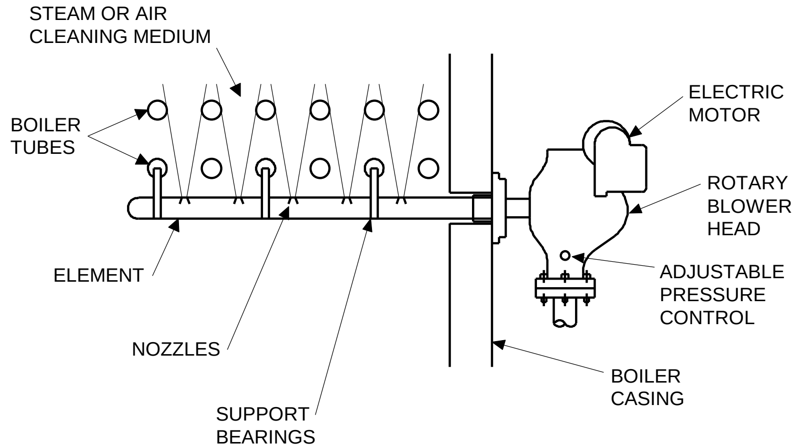


Figure 2-1
Multi-nozzle element on rotary blower head

A very early evolution of design produced a type of sootblower known as a “fixed position rotary, multi-nozzle element” sootblowing unit. While design improvements have taken place over the years, many units of this basic design continue to find application in some industrial type boilers. The portion of the unit that remains in the boiler consists of an element of an approximate 1-1/2 to 2 inch OD (38.1 TO 50 mm) that is supported by bearings welded to the boiler tubes, as illustrated in Figure 2-1.

Small diameter nozzles are spaced over the length of the element to strategically direct the steam or air cleaning medium into the adjacent bank of tubes. The element is connected to its “head,” mounted outside the boiler casing, also shown in Figure 2-1. Early designs of this type unit were manually operated by movement of a chain wheel to rotate the element. In later years, the units were made to be either air or electric motor operated, as illustrated in Figure 2-2. This unit, as manufactured by Diamond Power Specialty Company, is designated as a Model G9B fixed position rotary blower and is the type of unit applied for steam blowing.

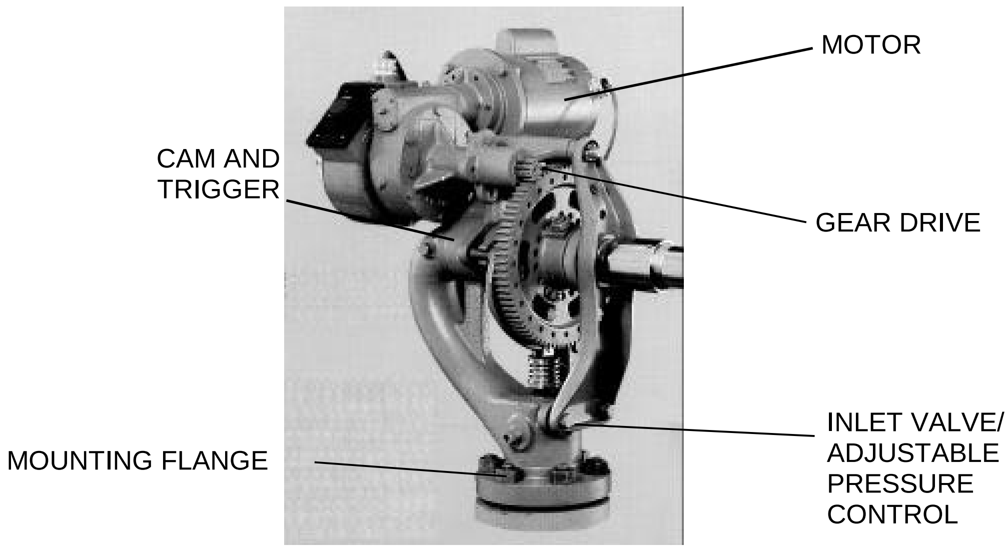


Figure 2-2

Model G9B fixed position rotary sootblower

Another version of the fixed position, multi-nozzle element unit led to the first usage of compressed air for sootblowing service. These units utilize the same multi-nozzle element as applied with the Model G9B unit but are equipped with an air actuated, piston operated air inlet valve. These units are commonly referred to as " automatic air puff" units and are designated Model A2E blowers as supplied by Diamond Power.

The automatic air puff sootblower system normally uses compressed air in the pressure range of 80 to 150 PSIG (5.52 to 10.34 BAR) supplied to the system by a small reciprocating compressor and air receiver. The combination of the compressor and receiver is sized to supply the required volume of air to sustain one "puff" of air for a duration of approximately one second or for the period of time that it takes for the receiver pressure to drop from 100% of pressure to a predetermined level. The time required to reach the preset pressure level is known as the receiver blow-down which establishes the duration of the air puff from the element. The time required to pump up, or recover, the receiver pressure determines the elapsed time between puffs. An Air Master Controller counts times and transmits the air pressure signals to the sootblower head to rotate the element through a small arc between puffs, until the unit completes one full rotation. The controller then automatically sequences the operation to the next unit. The Model A2E head, or air inlet valve, is illustrated in Figure 2-3.

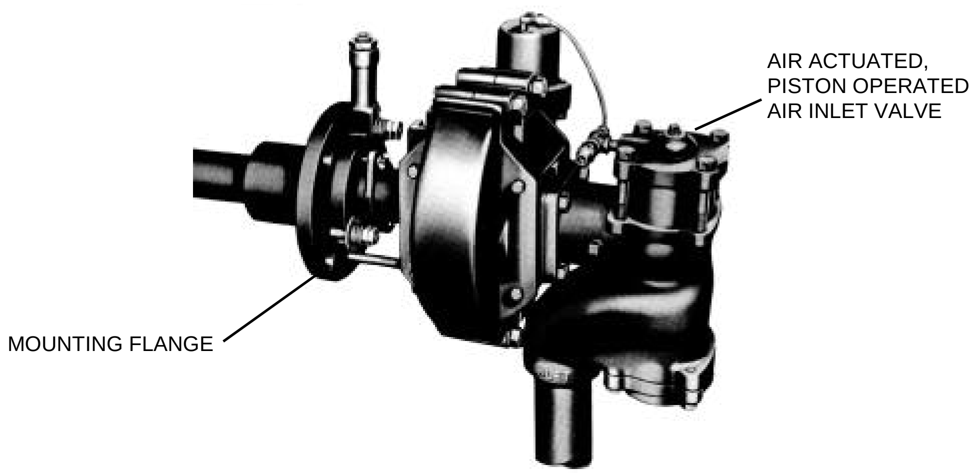


Figure 2-3
Model A2E automatic air puff blower head

Multi-nozzle rotary type sootblowers were successfully applied in numerous situations however; they were limited by both length and temperature. Elements manufactured from a high chrome alloy steel permitted application in flue gas temperatures up to approximately 1900^{0}F (1037.7^{0}C). The length of the element was limited to approximately 15 feet (4.57m) as the number of nozzles normally included in an element of this length would result in a prohibitively high pressure drop and render the unit ineffective. As boiler widths increased with the evolution of larger boilers, it became necessary to develop a unit that could be applied in higher gas temperature zones as well as in boilers of greater width than could be accommodated with the fixed position, rotary type unit. These requirements brought about the development of the "long retractable" type sootblower unit.

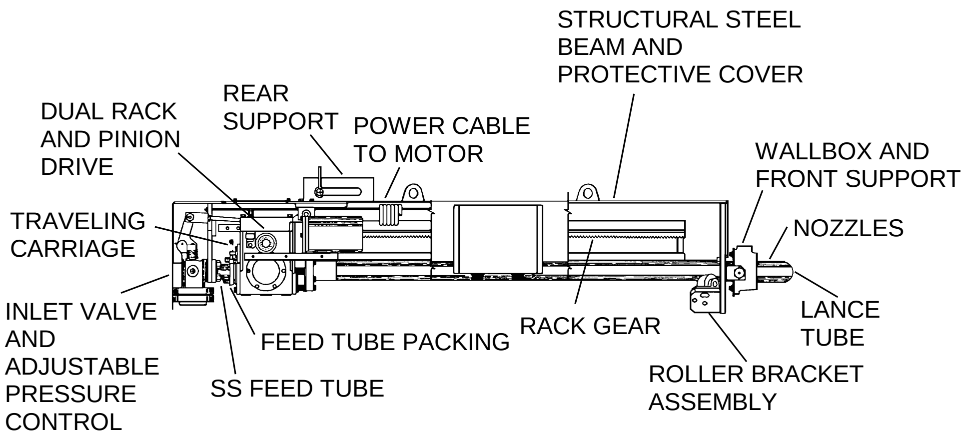


Figure 2-4
Model IK-525 long retractable sootblower
with cover cut away to show major components

Figure 2-4 illustrates a typical electric motor operated, long retractable sootblower with its protective cover removed to show the major components of the unit. The lance tube is propelled into and retracted from the inside of the boiler by a single motor driven carriage assembly. An arrangement of gears within the carriage assembly drives a pinion on a stationary rack to propel the lance tube into and out of the boiler. At the same time, appropriate gearing causes the lance tube to rotate throughout both the forward and reverse travel. The end of the lance tube is fitted with a set of nozzles through which the cleaning medium; either air or steam, is directed to the boiler tubes. The blowing pressure for each unit is regulated at the inlet valve, mounted on the unit, which is equipped with an adjustable pressure control. Similar long retractable sootblowers with variations of design are supplied by other manufacturers.

Opening of the inlet valve admits the blowing medium to be supplied to the nozzles at the end of the lance tube through a stationary “feed tube.” The blowing medium is prevented from leaking to atmosphere from between the lance and feed tube by a packing gland located at the outboard end of the lance tube. The accessibility of the feed tube packing gland is of particular importance when steam is applied as the cleaning medium. Exposure to high temperature steam in addition to mechanical wear results in deterioration of the packing. This makes frequent replacement of the packing necessary. The use of compressed air as a cleaning medium virtually eliminates the need to replace the feed tube packing. Normally, these units are mounted in what is called a “double installation” in which the lance tube travels a distance equal to one half of the boiler width from units mounted on each side of the boiler. Later in this text, we will consider the variables that influence the selection of the number and location of units on the boiler.

A variation of design and application of the long retractable unit is that of the semi-retractable sootblower, sometimes referred to as an extended lance unit or as a half -track. For the purpose of this discussion, we will refer to the unit as a semi-retractable blower.

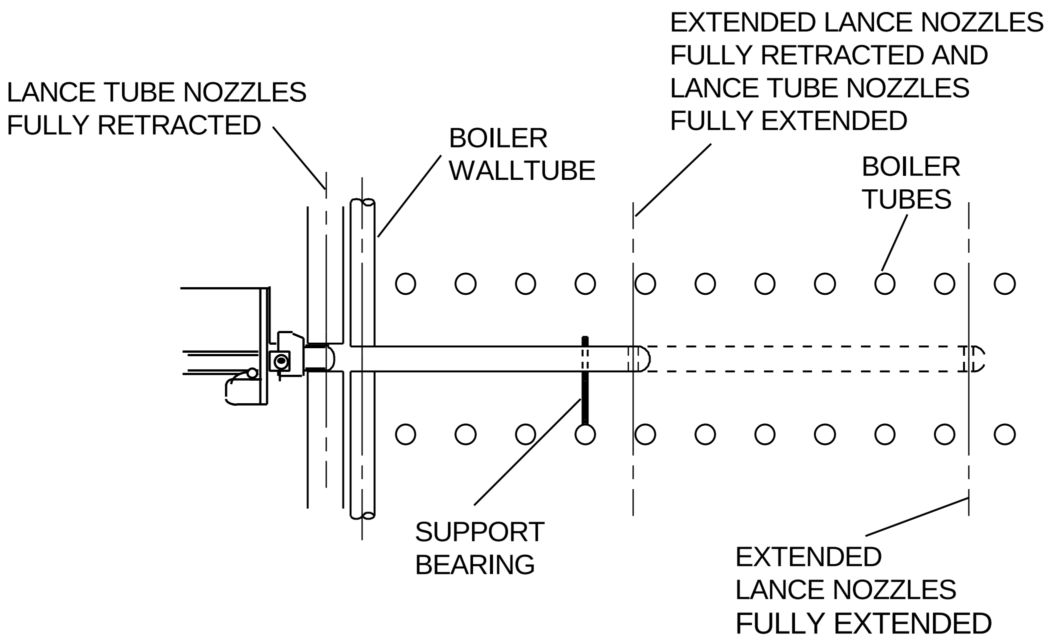


Figure 2-5
Semi- retractable or extended
lance tube sootblower

A semi-retractable unit is one to which an extension has been affixed to the lance tube, normally equal in length to the lance tube as well as to one half of the required blower coverage. At full retraction, the lance tube extension remains inside the boiler setting, supported by a bearing welded to a boiler tube located in the horizontal tube bank directly beneath the location of the sootblower. This arrangement of the semi-retractable unit is illustrated in Figure 2-5. As the lance tube extension remains in the boiler at all times, the locations at which the unit may be applied must be at a flue gas temperature in which the extended lance tube can survive without cooling. Normally, such a unit is not applied in gas temperature zones above 1600 ^{0}F. (871^{0}C). This generally makes the semi-retractable unit acceptable for installation to clean the economizer section of the boiler where gas temperatures entering the economizer would not be expected to exceed 1000 ^{o}F. (538^{0} C). The choice of applying a semi-retractable, rather than that of a full travel unit is typically made as a cost consideration. However, there are instances in which there is no reasonable alternative to that of applying a semi-retractable unit. Since the travel, and consequently the length of the blower is only one half that of a full retractable unit, the clear space or clear platform area required outside the boiler for its installation and access for maintenance is also one half that of a full retractable unit. There are times at which the installation of the sootblower is impeded by the lack of clear space or platform area outside the boiler.

The remaining blower type with which we are concerned is commonly referred to as a "furnace wallblower," as illustrated in Figure 2-6. This unit, shown here with the protective cover removed to expose the working mechanism, is designated the Model IR-3 Furnace Wallblower as supplied by Diamond Power.

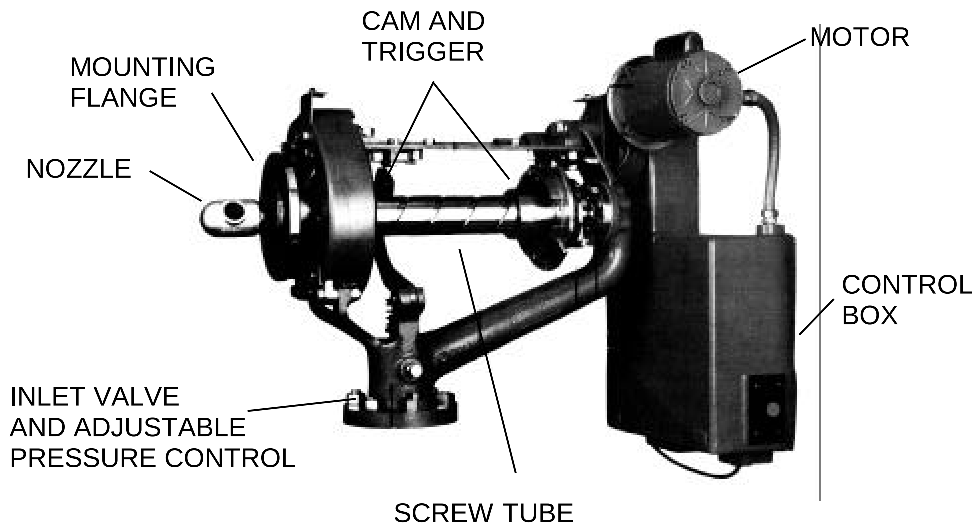


Figure 2-6
Model IR-3 furnace wallblower

This blower, sometimes referred to as a “short retractable” blower, also retracts its nozzle from inside the boiler while the unit is at rest. The stationary portion of the unit, mounted outside the boiler setting, includes an inlet valve at which the pressure of the blowing medium is regulated by means of an adjustable pressure control. The single ported nozzle, positioned inside the wallbox between blowing cycles, is propelled into the boiler and rotated by a single motor driving through a screw tube. Depending upon the severity of the application, the nozzle can be set for either one, two, or three revolutions.

Furnace wallblowers are applied only to pulverized coal fired boilers. Oil fired boilers do not develop the kind of furnace wall deposition that requires cleaning with sootblowers.

Date of Issue: 4-Oct-1996

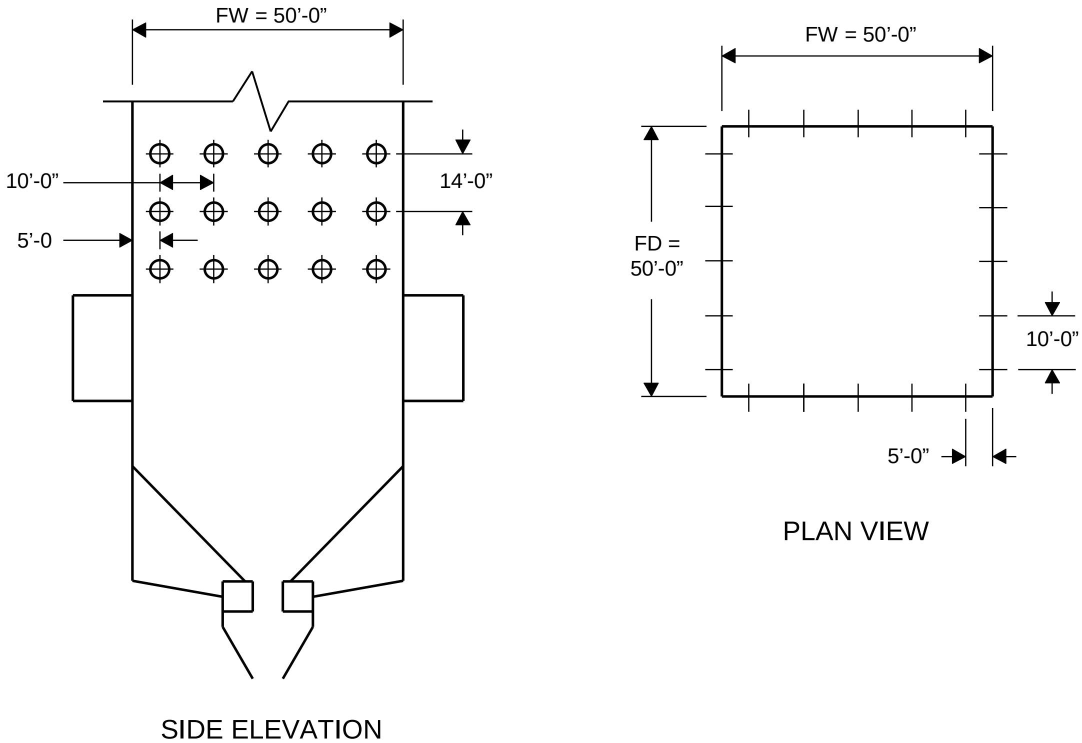


Figure 2-7
Typical furnace wallblower arrangement

Figure 2-7 illustrates both a side elevation and plan view of a typical boiler-furnace of a PC fired boiler with a "typical" arrangement of furnace wallblowers. For the sake of discussion, we will make the following assumptions:

Turbine Generator Design Rating	600 MW
Boiler Evaporation Rate	4,000,000#/hr.
Fuel:	PC
Furnace Width	50'-0" (15.24m)
Furnace Depth	50'-0" (15.24m).

Unless the boiler manufacturer or designer is dealing with a very poor grade of fuel with which he might expect very severe slagging in the furnace, he would normally assume that each furnace wallblower would be effective over an elliptical pattern with a radius of about 7 feet (2.13m) vertically and 5 feet (1.52m) horizontally. Therefore, it would be logical to place the wallblowers approximately 5 feet (1.52m) from the corner tubes of the furnace and on about 10 foot (3.0m) horizontal centers across the width and depth of the furnace. A furnace of this magnitude would probably be of sufficient height to require three elevations of wallblowers on 14 foot (4.27m) vertical centers. In this case, there would be a total of 60 wallblowers on the furnace walls.

When fully retracted and at rest between blowing cycles, the wallblower nozzle is positioned in the wallbox as illustrated in Figure 2-8.

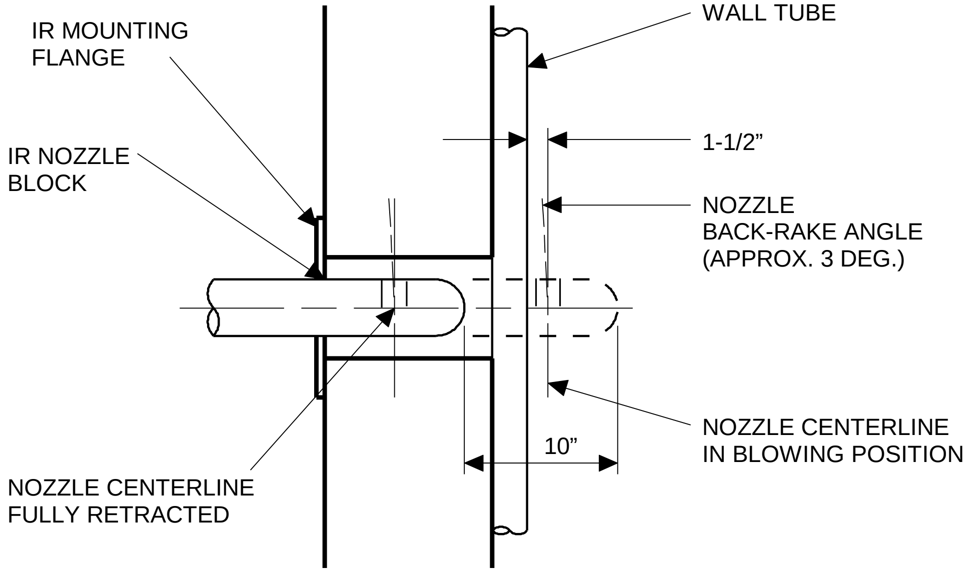


Figure 2-8
Wallblower nozzle
At rest and in blowing position

When called upon to operate, the motor is energized and the nozzle block begins to rotate. As it rotates, it is advanced into the furnace by a screw type drive that completes one revolution in approximately 20 seconds. After two revolutions the nozzle is advanced to the blowing position, and the cam operated inlet valve is opened to admit the flow of cleaning medium to the nozzle. The nozzle normally completes two blowing revolutions and is then reversed to close the inlet valve and to retract the nozzle to its at- rest position in the wallbox. Two revolutions are required to retract the unit and two additional revolutions to reset the unit’s controls. In total, each furnace wallblower will complete eight (8) revolutions in a period of 160 seconds, or 2.67 minutes. The actual blowing time, or period of time that the unit is consuming air or steam is only about 40 seconds, or .67 minutes. These time increments are based upon 60 Hz. operation and will vary slightly for 50 Hz. Operation.

The function of furnace wallblowers is to control the transfer of heat from the furnace gases to the furnace water wall tubes that line the furnace. There are a number of indicators that tell the boiler operators whether or not the furnace wallblowers need to be operated. One such indicator is visual observation. Some plants will employ roving operators to view the inside of the boiler through observation doors and to report upon their visual observation of slag depositions in any part of the boiler or furnace. Another indicator is the monitoring of the furnace exit gas temperature. A higher than normal or design furnace exit gas temperature suggests that the furnace walls are not absorbing enough heat and should have additional cleaning with sootblowers. Under absorption of heat in the furnace can result in the formation of severe slagging in the superheater or other sections of the boiler. The combination of inadequate heat transfer in the furnace and severe slagging in the pendent sections will result in severe upsets or excursions of final steam temperature at the superheater outlet. Conversely, if the monitored furnace exit gas temperature were to be lower than normal, the indication would be that the furnace walls are absorbing too much heat. In this case, the operation of sootblowers would be curtailed in an effort to reduce the rate of heat transfer to the furnace walls. Generally speaking, the furnace wallblowers on a PC fired boiler firing coal with a medium fouling index would be operated a minimum of once each shift, or three times a day. In many cases, the wallblowers are programmed to operate simultaneously with long retractable blowers in the boiler on a continuous basis.

While the air or steam flow rate required of a furnace wallblower will vary depending upon the cleaning demand, it is normally estimated that the air or steam required per unit is approximately as follows:

Steam Flow Rate:	160#/min. (9600#/hr.)
Air Flow Rate:	2300 SCFM (1.25 m^3/sec)
No. and Size Nozzle	One (1) 1 inch diameter
Blowing Time (2 Revolutions)	40 Seconds (.67 min.)*
Steam Required Per Blow:	107#
Air Required Per Blow:	1541 cu.ft. (43.64 cu. meters)

*For 50 HZ power, time is 25 sec./revolution or 50 seconds per blowing cycle.

The location of long retractable units in the boiler must take a number of considerations into account. The most significant of these considerations are the following:

Fuel chemistry and expected fouling tendencies,

Expected flue gas temperature at the blower location,

Expected effective cleaning radius,

Expected lance tube deflection at full extension,

Boiler expansion from the cold to the hot condition,

Clear space available outside the boiler.

Figure 2-9 outlines the balance of the hypothetical boiler with the location of long retractable units shown.

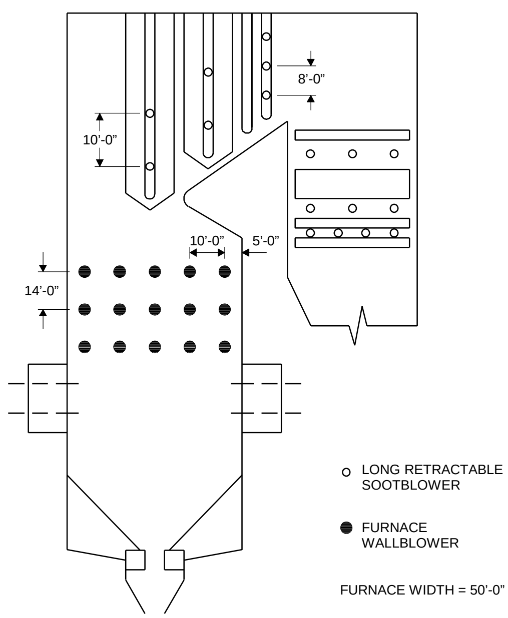


Figure 2-9

Under reasonably normal circumstances, a long retractable unit can be expected to have an effective cleaning radius of 5 to 7 feet (1.5 to 2.13m). It is, therefore, common practice to locate these units on 10 to 12 foot (3.05 to 3.66m) vertical centers. The horizontal or side spacing depends entirely upon the arrangement of the heating surface inside the boiler. Designers of blowers installed in vertical cavities between loops of pendent tubes as shown in the superheater and reheat sections of the boiler must consider the lateral deflection of the lance tube at full extension. This is to preclude the possibility of positioning the nozzle too close to the tubes at full extension. Allowing the nozzle to be positioned too close to the boiler tubes creates the potential of causing serious tube erosion. In the case of the blowers in the downflow section of the boiler, installed in horizontal cavities, the consideration of lance tube deflection may be even more important. Apart from the consideration of tube erosion, the designer must consider that the height of the cavities required for sootblower installation are a major influence upon the cost of the boiler.

The sootblower lance tubes are carefully designed and manufactured to minimize deflection as much as possible. Lance tubes are normally 3 1/2 or 4 inch OD (88.9 or 101.6mm) tubes of materials suitably selected for the gas temperature in which they are to be installed. Lance tubes exceeding 20 or 25 feet (6.1 or 7.6m) in length are normally fabricated of two or more sections, the rear section having the greater wall thickness and the outboard end having a lesser wall thickness to minimize the deflection at full extension.

The material of which the lance tube is fabricated ranges from the standard of a low chrome alloy (1-1/4 chrome) to the very highest yield strength of stainless steel. The selection depends upon the gas temperature in which the unit is installed and upon the cooling medium (air or steam) available to cool the lance tube when exposed to elevated gas temperatures. In some unit locations, the air or steam flow recommended to serve as the cleaning flow rate will be sufficient to maintain a safe metal temperature of the lance tube. At higher gas temperature locations, the flow rate required to maintain a safe metal temperature may exceed the flow rate required for cleaning. At these locations, the sootblower is said to require a lance tube cooling flow rate. Low temperature compressed air is generally regarded as a more effective cooling medium than higher temperature steam. This cooling flow rate is always established by the sootblowing equipment manufacturer.

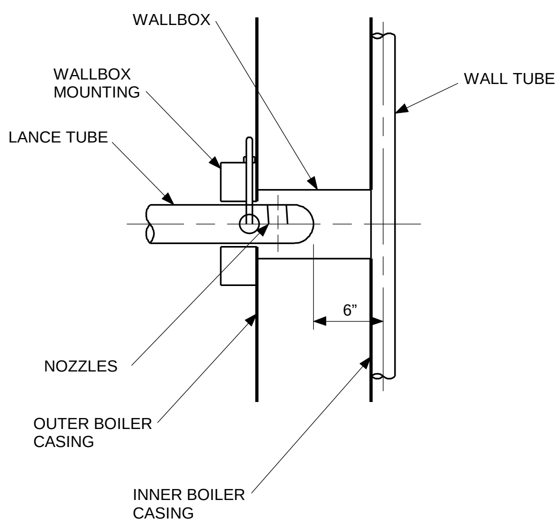


Figure 2-10
Lance tube in fully retracted position

When fully retracted and at rest between blowing cycles, the nozzle block on the end of the lance tube is positioned in the “wallbox” as illustrated in Figure 2-10. At initial operation of the unit, the drive motor is energized and the lance tube begins its forward travel. The unit will advance from its at-rest position to a point approximately six (6) inches from the centerline of the water wall tubes.

At this point, a stationary cam on the unit engages a moving trip pin that opens the blower’s inlet valve to admit the flow of cleaning medium, either air or steam. The lance tube continues to travel forward and to rotate until it reaches the centerline of the boiler. At this point, a limit switch reverses the travel of the lance tube, and the cleaning medium continues to flow from the nozzles during the reverse travel of the blower. At a point approximately six (6) inches (152.4mm) from the centerline of the wall tubes, the trip pin engages the cam to close the blowing medium inlet valve. The lance tube is then fully retracted into the wallbox, and the operation is automatically transferred to the next sootblower in the operating sequence.

The travel speed and, consequently, the total operating time and the actual blowing time varies from unit to unit and is generally selected on the basis of the flue gas temperature zone in which the blower is to operate. The most commonly applied travel speeds are 70, 100, 140 inches/minute The total operating time is a function of the total travel distance, both forward and reverse, multiplied by the travel speed. The actual blowing time is the distance traveled after the inlet valve is opened during the forward travel until it is closed near the end of the reverse travel.

There are other types of sootblowers that may become a topic of discussion, but only on such infrequent occasions that it does not seem appropriate to burden the reader with detailed descriptions.

Note: The values noted in the foregoing represent travel speeds, operating times, blowing times, air and steam flow rates are typical of those applied to equipment supplied by Diamond Power Specialty Company

The sootblowing control system is a vital part of the boiler cleaning system and is available to the user in varying degrees of sophistication. The primary purpose of the control system is to provide a means of selecting the sequence in which the sootblowers are to operate and to provide the necessary protective functions and fault alarms needed to safeguard the system. However, control systems are available to provide complete computerized control to operate sootblowers on demand, in response to various strategically placed detection devices.

Compressed Air Versus Steam Usage

For as many years as a choice has been available, the question of which cleaning medium is more effective—compressed air, saturated steam, or superheated steam – has been asked by the majority of concerned persons. Within the industry there are advocates of each who believe they have good reason for their preference. The consulting engineer called upon to recommend a choice of blowing media based upon an evaluation would do well to determine his client's personal preference and to evaluate in favor of that preference. Given the incentive to do so, the engineer should not find this task especially difficult to accomplish.

If one is to influence a decision to use compressed air rather than steam for sootblowing service, he must do so at a very early point in time with the key buying influences representing the end user. This generally involves contact with the ultimate customer's senior executives responsible for engineering and for production or operations. It is also necessary to make early contact with the senior engineers employed by the architectural engineer or other consulting engineering firm. It is unlikely that the architect or consulting engineer will make the decision to use either compressed air or steam but; it is important that he be made aware of the benefits that are being discussed with the end user. In the absence of a preference stated by the end user, the architect will normally defer to the boiler manufacturer who will generally elect to use steam. Such a choice made by the boiler manufacturer, may not always be in the best interest of the ultimate customer.

With due respect given to those who state a strong preference for one blowing medium over the other, the most logical and best supported statement regarding the comparative effectiveness of cleaning media is simply this: "Properly applied, either compressed air, saturated steam, or superheated steam will clean the fireside of boiler tubes with equal effectiveness." Aside from the observations made in numerous cases over the years by sootblowing equipment suppliers as well as by the end users, this position is supported by the following logic and tests.

The ability to remove ash deposits from the fireside of boiler tubes is a function of the kinetic energy of the jet stream at the point of impact and, hence, is a function of weight flow and fluid velocity. This conclusion is supported by extensive experimental and theoretical work involving the testing of numerous nozzle designs aimed at determining the magnitude of the energy of a freely expanding jet stream at varying distances from the nozzle exit. The earlier reference to the function of weight flow and fluid velocity as related to the function of kinetic energy at the point of impact suggests a concern for the value of fluid horsepower developed at the nozzle exit. This value can be calculated for any given fluid exiting a nozzle and is expressed as follows:

$$\text{Fluid Horsepower} = \frac{W\,V\,(P \times 144)}{33{,}000}$$

Where: W = Flow (in lb./min.)

V = Specific Volume (in cubic ft./lb.)

P = PSIG (at the nozzle)

This simply means that if one wants to develop a level of fluid horsepower, or work energy, at the exit of a nozzle with compressed air to be equal to a known value of energy at the exit of a nozzle using steam he may do so by varying the nozzle diameter (flow area) and nozzle pressure until the desired level of fluid horsepower is reached. The real test of this theory is that of comparative measurement of impact pressure at various distances from the nozzle exit. This

Date of Issue: 4-Oct-1996

measured value is commonly referred to as PIP or Peak Impact Pressure because it represents the maximum of impact pressure detected by multiple samplings of impact pressure across the effective cone of energy expelled from the nozzle. In order to equate the calculated value of fluid horsepower to kinetic energy at the point of impact upon the ash deposition and, consequently, upon cleanability, testing of Peak Impact Pressure measurements have been made with a test apparatus as illustrated in Figure 3-1.

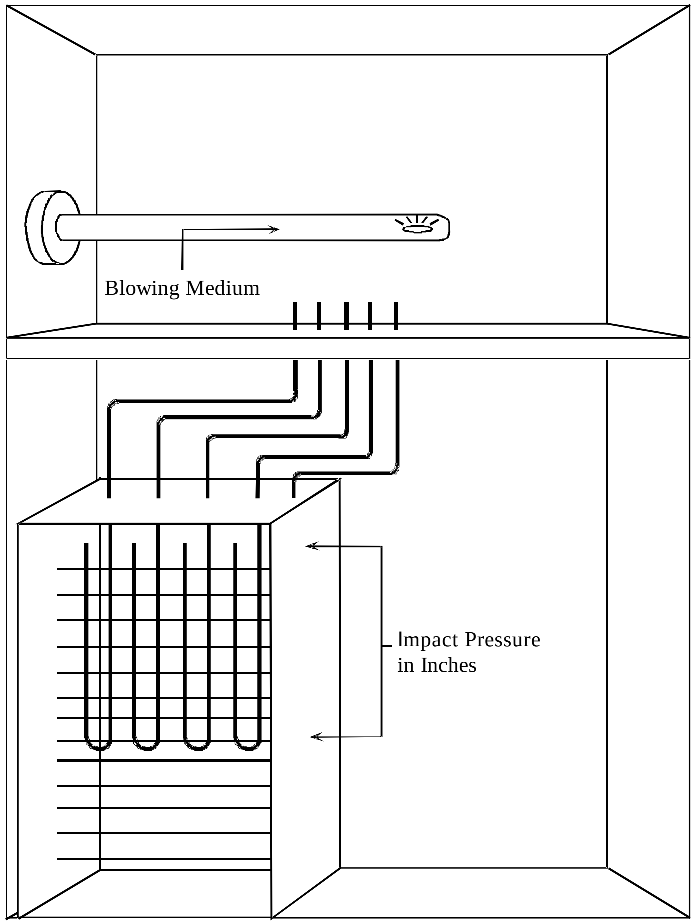


Figure 3-1
Peak impact pressure test apparatus

This test involves the measurement of stagnation pressure by impingement of the jet on the bank of impact tubes located transversely in the jet stream at any given distance from the nozzle. This resultant value of PIP has been correlated to a factor of "cleanability" as a result of extensive field testing with various types of fuel ash depositions. What such testing really tells us is that the removal of the ash deposition on boiler tubes is accomplished with kinetic energy at the point of application and that it really makes no difference whether the work energy, or kinetic energy, is developed by saturated steam, superheated steam, compressed air, or by some other means. There will always be those who will challenge this logic, which lends support to the suggestion that the selection of either compressed air or steam for sootblowing service is largely a matter of personal preference based upon some years of past experience.

The question of a comparative economic evaluation of the cost of compressed air versus that of steam often arises. With the exception of a few specific cases, such an evaluation can be made to favor either air or steam, depending upon the client's known preference. A more likely situation, though, would be one in which the economic evaluation results in a "toss-up" between the cost of air versus the cost of steam, in which case the client can justify his personal preference. However, there are specific instances in which an economic evaluation will realistically favor the use of compressed air.

One such instance would be in the case of a new plant construction located in an arid region such as the Southwestern part of the United States or comparable arid regions abroad. The primary concern in this instance would be the availability and the cost of make-up boiler feedwater and the cost of demineralization and other treatment of the feedwater to make it suitable for use. This cost is still a greater concern when a low rank, or poor grade of fuel is expected to be used. The low rank fuel would require more extensive use of sootblowing equipment, therefore, adding to the cost of feedwater make-up and feedwater treatment. This cost can be quite substantial and is one reason that the majority of central station utility boilers located in the Southwestern region of the United States use compressed air for sootblowing service. This concern would also apply to an arid region anywhere else in the world.

Another such situation is that of the existing boiler being converted to burn a low rank coal out of consideration for the control of sulfur dioxide, or other objectionable pollutants in the effluents discharged into the atmosphere from the stack. In this case, one can be sure that the change of fuel will make it necessary to place additional sootblowing equipment on the boiler and will, no doubt, require more frequent operation at higher pressures and resultant flow rates than was previously needed. Assuming that the existing boiler used steam for sootblowing this would require additional make-up water, demineralizing capacity, and other associated costs of increased usage of boiler feedwater. These costs can sometimes show justification, even for the very costly conversion of an existing steam blowing installation to one utilizing compressed air.

Still another consideration for the owner or operator of a boiler converted to burn a low rank coal and to continue the use of steam for sootblowing is that of the increase in the instantaneous demand for steam from some point in the vapor cycle brought about by the addition of certain high flow long retractable sootblowers. Depending upon where in the superheater the steam is taken from, the instantaneous withdrawal of steam at the rate of 15,000 to 20,000 #/hr. from the superheater circuit to supply the sootblower, could very possibly create an upset condition in the boiler commonly known as "starving" the superheater.

This is detrimental to the operation of the boiler and is a situation that the operating people must be careful to avoid. This, like so many other influences, is one of those factors that favor the use of air sootblowing but is not quantifiable to the extent that it can be considered in an economic evaluation.

A major "downside" effect of converting an existing boiler to burn a low rank coal is a result of the fact that the existing boiler, particularly the furnace volume of the boiler, was not designed to burn a low rank, low heat value coal. This being the case, and again depending upon the circumstances, it may become necessary for the owner to "de-rate" the boiler capacity with a corresponding reduction to the turbine generator's rated output in terms of electrical power. If this should happen to be the case, the owner should consider every potential for limiting the extent to which it is necessary to de-rate the capacity of the boiler. This would mean the limiting or elimination of as many "indirect" usages of steam as possible; and, as mentioned earlier, sootblowing demands are generally the largest indirect user of steam. If the predicted volume of steam required per hour to sustain the sootblowing system is sufficiently high to make a meaningful difference in the quantity of steam that could be used to drive the turbine, there will quite possibly be justification for conversion of the sootblowing system to use compressed air.

While it appears unlikely that we will see major growth of new boiler construction in the United States, such is not the case on the international scene. It is within this area that we have need to discuss confidently those advantages of compressed air usage over that of steam, which we have referred to as being “qualitative” but not easily “quantified.”

It can be stated with confidence that the major contributor to high maintenance of a steam blowing sootblower piping system is the inadequate drainage of condensate that accumulates in the sootblowing unit and in the piping system between sootblower operations. Steam piping for sootblowing systems must be carefully designed and installed with an adequate slope away from each unit to a low point in the piping where an arrangement of automatic drain traps and strainers are installed. Additionally, the piping must allow for both vertical and lateral thermal expansion of the boiler with the installation of sizable horizontal and vertical expansion loops. During even short intervals of “no flow” in the system, the piping tends to cool, forming condensate in the system. Ideally, all of this condensate would drain to the low point in the piping and would not present a problem; however, this is nearly always something less than an ideal situation, and the system is rarely ever completely drained of condensate. Contributing to the accumulation of condensate is the well-known fact that most automatic drain traps are not as effective as one would like them to be. It is also well- known that the drain traps and strainers rarely ever get the kind of maintenance attention that they should have to make them reasonably reliable. Whatever the reason might be, the fact is that condensate does accumulate in the piping and within the internals of long retractable sootblower units. As the system is again pressurized and flow is established, the initial flow of steam through the long retractable blower nozzle tends to purge the condensate from the unit and on to the boiler tubes. These initial slugs of condensate ejected from the nozzle are commonly known as “water hammers,” and are generally a factor over the first three or four feet of the retractable blower’s travel into the boiler. These droplets of condensate, entrained in the steam being blown from the nozzle at a very high velocity, can cause very harmful erosion of the boiler tubes. This erosion is often visible in the form of a “polished” appearance of the tube surfaces over the first few feet of the blower’s travel. Failure to detect, and to correct, such tube erosion could result in a rupture of a boiler tube, which could have catastrophic consequences.

Another potential problem area attributable to the inadequate drainage of condensate from the system is that of internal corrosion within the sootblower itself. It takes only very small quantities of condensate trapped inside the sootblower lance tube while it is at rest and retracted from the boiler to combine with small quantities of boiler gases to form agents that will corrode the inside of the lance tube.

Internal lance tube corrosion generally occurs at the point at which sections of the lance tube are welded together, as the internal weld is likely to form a small ridge to act as a dam at which the condesate can accumulate. Unfortunately, there is no way to detect the formation of internal corrosion of the lance tube, which can, and often does, cause a lance tube to bend to the extent that it cannot be retracted or to break off completely and fall into the boiler. This occurrence has also been known to rupture boiler tubes and to cause a shutdown of the boiler.

Figure 3-2 is typical of the manner in which lance tubes of lengths greater that twenty or twenty five feet are likely to be fabricated.

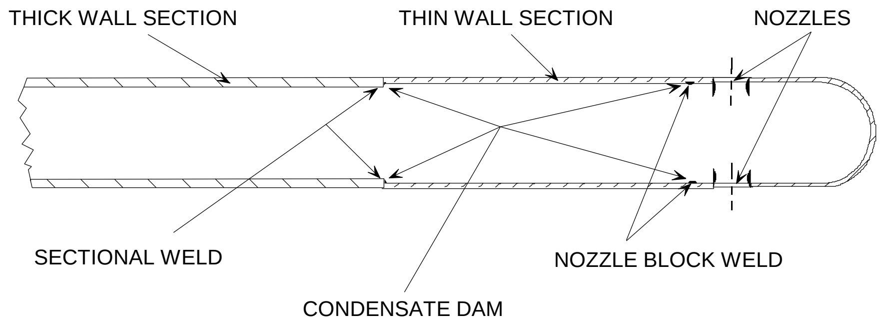


Figure 3-2
Two piece lance tube

As mentioned earlier, the lance tube is normally made up of two or more sections with a constant outside diameter and a varied inside diameter to reduce the deflection of the lance tube at its full extension. While a smooth finish is maintained on the outside diameter, it is virtually impossible to completely clean and smooth the welds on the inside of the lance. These are the points at which some quantity of condensate is likely to accumulate between operations of the sootblower. The potential of causing serious boiler tube erosion as a result of this accumulation of condensate has been sufficiently common to cause sootblowing equipment manufacturers, as well as some users, to provide a means of “pre-heating” the outboard end of the retractable blower prior to its operation to evaporate the accumulated condensate. This would seem to attest to the fact that the potential of this problem is very real. Once again, the value of preventing the possible occurrence of such damage by utilizing air, rather than steam, is difficult to quantify. However, if the elimination of such an occurrence was to prevent one forced outage of the boiler for the repair or replacement of one or more boiler tubes, a reasonable premium investment to cover the cost of changing to a compressed air system might be justified.

Another concern related to the cleanability of the boiler, which is also linked to the potential presence of condensate or moisture entrained in the steam blown from the sootblower, is that of possible “plugging” of the flue gas passages between boiler tubes. This is most likely to occcur in the downflow section of the boiler where the tubes are installed with comparatively close side spacing. It is fairly typical for low rank fuels with a high ash content to produce a very fine, almost powder- like consistency of ash in the lower gas temperature sections of the boiler, such as in the economizer. Mixing moisture, or condensate droplets, from the steam blowing sootblower with this powder- like dry ash tends to form a hardened material that will bridge over from one row of tubes to the next until the gas passage between tubes is totally blocked. Such plugging in the back end of the boiler has also been known to cause unscheduled outages of boilers.

Still another such concern relates to the cleanability and availability of a regenerative airpreheater that is commonly installed between the boiler gas exit and the entrance to the stack. The most commonly applied regenerative airpreheater is supplied by The Air Preheater Corporation and consists of several fine mesh- like revolving “baskets” through which gases flow from the boiler exit and ambient air is made to flow in a counter direction. The objective of the process is to recover the heat from the flue gases to preheat the air that is conveyed to the windbox of the furnace burners to support and improve the efficiency of fuel combustion. The airpreheater is normally equipped with a cleaning device located at the “cold end,” or air entrance end, and sometimes another cleaning device at the “hot,” or gas entrance end. Once again, the carry-over of this very fine, powder-like ash, mixed with moisture blown with the steam from the cleaning device, will sometimes result in plugging of the surface in the airpreheater. In this instance, an even more serious problem results from the plugging, or resistance to flow, through the gas side of the airpreheater.

It is necessary to maintain sufficient gas flow to keep the temperature of the “cold end” of the airpreheater at a level above the dew point. Otherwise, the material of which the baskets are made is subject to severe cold end corrosion.

This occurrence was so commonplace a considerable number of years ago that The Air Preheater Corporation issued a notice to all users stating that if steam is applied as a cleaning medium it should be of a quality equal to 250 or 300 degrees of superheat. In a great many cases, steam of this quality is not available, and in those cases in which it is available, the cost of applying it to clean the airpreheater is all but prohibitive. Steam of this quality would be of such a high temperature that all of the piping, valves, and fittings from the steam source to the point of application would have to be made of a high grade stainless steel, which is very costly. The logical alternative is the use of compressed air.

Boiler sootblowing equipment is also adversely affected by exposure to high temperature steam. This is particularly true in the case of maintaining the feedtube packing that prevents leakage of steam from the outboard end of the feedtube and lance tube while the unit is in operation. Severe wear that is the result of the lance tube’s motion, combined with exposure to high temperature steam, results in deterioration of the feed tube packing. Replacement of feedtube packing is not an expensive maintenance item. However, with thirty to forty long retractable units installed on a boiler, it becomes a difficult task, and is one that is often not attended to. While one or more sootblowing units are awaiting maintenance attention, they remain out of service to the detriment of the boiler cleaning effectiveness. The potential of excessive feedtube packing failure due to severe wear is virtually eliminated with air blowing systems by the use of a type of packing made of a Teflon or similar material. The same material can not be applied with steam blowing, as it has a temperature limitation of approximately 650 degrees F.

An economic evaluation of air versus steam usage should, at a minimum, consider the items listed in the following tables in which comparative investment costs and annual operating and maintenance costs of air versus steam sootblowing systems are tabulated. The comparative costs presented in the following are hypothetical and are based upon the following assumptions:

- **Boiler Type And Size:** Pulverized coal fired with a steam evaporation rate of 3,200,000 #/ hr. to drive a 400 MW turbine generator.
- **Fuel Type:** Sub-bituminous coal; with severe slagging and fouling characteristics.
- **Sootblowing Steam Source:** From an intermediate superheater header at 2600 PSIG and 880^{0}F., reduced to 600 PSIG at 729^{0}F.
- **Highest Steam Flow Rate:** 21,940 #/Hr.
- **Sootblowing Air Source:** Two (2) multi-stage centrifugal compressors, each rated at 6000 ICFM capacity at 325 PSIG discharge pressure. Compressor motor nameplate rating, 2000 HP. The multiple compressor selection supplies 100% of the sootblowing demand with 100% standby air capacity.
- **Maximum Instantaneous Air Flow Rate:** 5322 SCFM (Compressors rated in ICFM)

Based upon the assumptions noted in the foregoing, the table presented in Figure 3-3 is a representation of the comparative investment cost of compressed air versus steam sootblowing.

INVESTMENT COST COMPARISON
AIR VERSUS STEAM SOOTBLOWING

Equipment And Installation Cost:		Steam:	Compressed Air:
1)	Sootblowers & Accessories	Equal Cost	Equal Cost
2)	Piping, Valves and Fittings	$175,000	$110,000
3)	Piping Installation	$235,000	$170,000
4)	Piping Insulation (Installed)	$126,000	None Required
5)	Added Make-up and cooling water capacity.*	$50,000	None Required
6)	Increased Boiler Cost (at $25.00/lb./hr of steam)	$658,200	None Required
7)	Air Compressors And Receiver	None Required	$800,000
8)	Added Transformer Capacity, Switch Gear & Cables	None Required	$120,000
9)	Installation Of Compressors & Accessories	None Required	$200,000
	Total Investment Cost:	$1,244,200	$1,400,000
	Differential Investment Cost	(Base)	(Plus) $155,800

* Make-up water requires demineralization

Figure 3-3

Figure 3-4 tabulates the major items of cost to be considered in the evaluation of the comparative annual operating and maintenance costs of air versus steam sootblowing:

COMPARATIVE ANNUAL OPERATING
AND MAINTENANCE COSTS
AIR VERSUS STEAM SOOTBLOWING

Annual Costs:		Steam:	Air:
1)	Fuel cost at $8.00 per million BTU to supply steam for sootblowing.	$404,506	Not Required
2)	Fuel cost at same rate as above to supply power to air compressors.	Not Required	$356,033
3)	Cost of make-up water at $3.00/1000 gallons	$27,000	Not Required
4)	Estimated maintenance cost per year. Including boiler sootblowers and compressors.	$150,000	$75,000
	Total Annual Operating Costs:	$581,506	$431,033
	Differential Annual Operating Costs:	(Plus) $150,473	(BASE)

Figure 3-4

In the example shown here, the small difference in the initial investment cost favoring the use of steam is practically off set by the differential in the annual cost of operation and maintenance favoring the use of compressed air.

Date of Issue: 4-Oct-1996

Beyond those costs tabulated in Figures 3-3 and 3-4, such an evaluation should consider the higher rate of boiler availability and maintenance of the boilers' design thermal efficiency resulting from the use of the compressed air sootblowing system. The thermal efficiency of a steam generating boiler is very dependent upon the availability of the sootblowing system to maintain a norm of heat transfer throughout the boiler. This factor of thermal efficiency is quantifiable to some extent.. For example; the cost of evaporating steam as tabulated in Figure 3-3 assumes a boiler efficiency of 89% expressed as an efficiency factor of .89. Based upon this design efficiency, the cost of fuel required to generate 3,200.000 # / hr. of steam, stated as the capacity of the boiler, for a period of 24 hours is $5,874. If the boiler efficiency were to drop only four (4) points, to .85, the calculated added cost of fuel per day required to maintain the design evaporation rate would be $6,151. Or an increase of $ 277. A day. The value of efficiency loss that is not easily quantified is that of the period of time over which the thermal efficiency is effected by unavailability of the boiler cleaning system. This may be only for a period of a few days in which case it becomes a minimal concern or, the lack of cleaning system availability may be extended over a period of time great enough to permit the formation of slag deposits is some sections of the boiler that may require a boiler outage for removal of the slag deposits. This is by no means an abnormal or exaggerated situation and is one that results in a daily cost based upon the loss of revenue resulting from the loss of generating capacity over the period of time that the boiler remains inoperative. This is not to suggest that the thermal efficiency of the boiler will never be adversely effected by something less than 100% availability of the boiler cleaning system using compressed air rather that steam. It does suggest that the potential for such a loss of thermal efficiency or the need to take the boiler out of service is less likely with compressed air usage than with steam usage.

The cost of lost boiler efficiency or possibly the loss of generating capacity resulting from the loss of boiler cleaning system availability is rarely ever considered in the make up of an economic evaluation, largely because these costs are subject to a great many variables and are therefore difficult to quantify. If this factor is not considered as a quantified value, it should certainly be considered as a qualatative influence to favor compressed air usage.

To summarize, the meaningful points to be made when discussing the use of compressed air rather than steam for sootblowing service are as follows:

- Compressed air, properly applied, will clean fireside deposits from the boiler with effectiveness equal to either saturated or superheated steam.
- Properly evaluated, the lesser cost of annual operation and maintenance will off set the small premium paid initially for an air system versus that of a steam system.
- When using compressed air, frequent replacement of feedtube packing is not required. This results in lower maintenance costs and higher availability of sootblowing units.
- Compressed air sootblower piping does not require insulation. This reduces the initial cost of materials and installation. Low skin temperature of air piping eliminates hazard to personnel that is a concern with steam piping.
- With compressed air, drainage of condensate from the piping and sootblower internals is not a concern as it is with steam piping. This simplifies design and installation of the piping and minimizes potential of boiler tube erosion resulting from excessive moisture being entrained in the blowing medium. This also eliminates the potential of internal lance tube corrosion, which can result in costly replacement of the lance tube and possible damage to boiler tubes.

- Low temperature compressed air applied to those long retractable sootblowers installed in high gas temperature zones of the boiler, is generally regarded to be a more effective lance tube cooling medium than high temperature steam.

Compressor Selection And Application For Sootblowing Service

There have been cases in which the customer, committed to the use of compressed air, has specified the number of compressors to be applied, as well as the pressure and capacity at which the compressors are to be designed, without the benefit of discussion with the air compressor manufacturer. In such cases it becomes very difficult to influence a change to the specified requirements, based upon the compressor suppliers' knowledge of the application. It is for this reason that those persons representing the compressor manufacturer must be totally aware of those factors that influence the selection of air compressors for this particular application. It is essential that the compressor suppliers' representative establish early contact with the customer's key personnel to provide appropriate guidance to those responsible for preparation of the specification for the air sootblowing compressors. It is for this reason that we present the following overview of those considerations that are to be taken into account To assure the customer of a proper application of air compressors to his boiler cleaning needs.

This selection process begins by obtaining a Side Elevation General Arrangement drawing, a Sootblowing System Air Data Sheet, and a Sootblowing System Sequence Bar Chart, all of which should be in the client's possession. These charts and drawings are normally originated by the sootblowing equipment supplier and submitted to the customer either by the boiler manufacturer or by the sootblowing equipment supplier. It is unlikely that the compressor supplier's representative will ever be called upon to originate documents of this kind; however, it is important that we understand both the origin and the meaning of the information that appears on these data sheets so that we might apply the data to the selection of our best offering of compressors and auxiliary equipment to serve the requirements of the sootblowing system.

Figure 4-1 is a sketch illustrating the outline of the general arrangement of a boiler of typical configuration, which shows the location of sootblower units as they might be selected by the boiler manufacturer.

Date of Issue: 4-Oct-1996

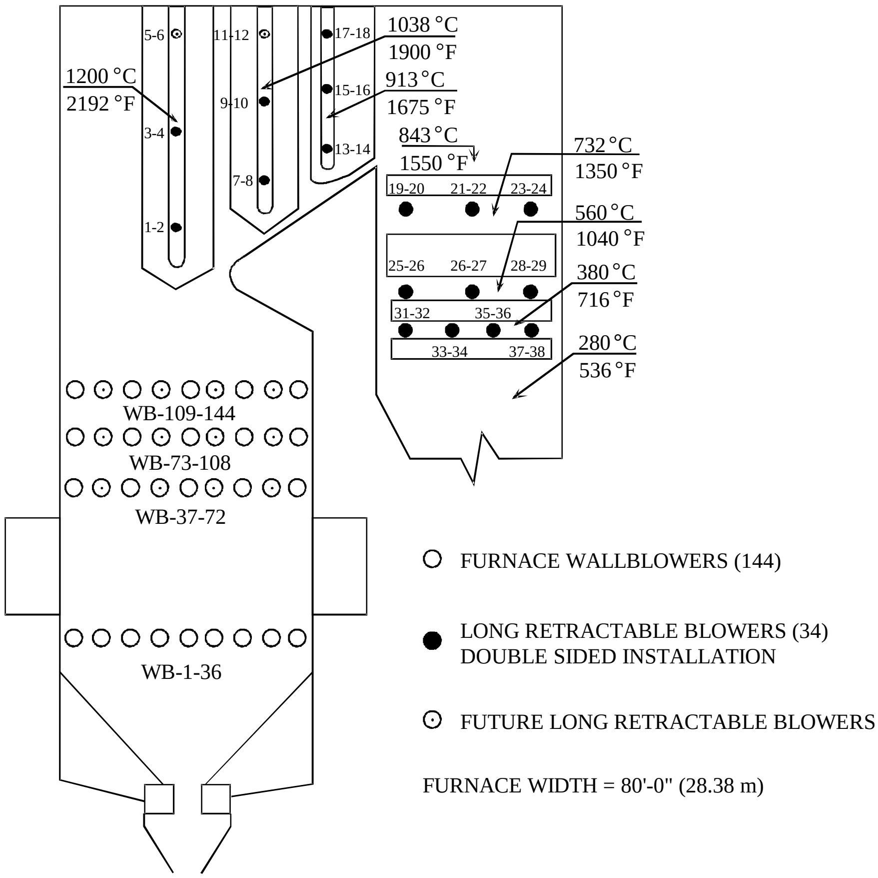


Figure 4-1

To avoid making reference to a particular supplier, we have used the designation **WB** for furnace wallblowers and **LR** for long retractable sootblower units. The illustration shows the furnace wallblowers as located on one side wall of the furnace and is representative of the wallblower locations on each of the remaining three furnace walls. Therefore, there are four elevations of wallblowers with nine units at each elevation or 36 units on each of four furnace walls. This makes up a total of 144 furnace wallblowers.

The indicated locations of long retractable units represent units installed on each of two side walls of the boiler in what we have previously defined as a “double installation.”

As these units travel into and are retracted out of the boiler, they have a coverage of one half of the furnace width, which is noted here as being 80 feet (29.38 m). The temperatures noted on the sketch at each long retractable unit location, in both SI and metric units, are the estimated temperature of the flue gases at that location. We have shown a total of seventeen (17) initial unit locations or 34 long retractable blowers and provisions made for two (2) future locations or four (4) future units.

The sootblowing equipment manufacturer, in conjunction with the boiler manufacturer, must determine how much air these sootblowers will require and over what period of time. This information is interpreted from the Sootblowing System Air Data Sheet, also originated by the sootblowing equipment supplier. A representation of this data is illustrated in Figure 4-2.

Sootblowing System Compressed Air Requirements

MODEL:	WB-1	LR-2	LR-2	LR-2	LR-2	LR-2	LR-2
NUMBERS:	1-144	1-4	7-10	13-18	19-24	25-30	31-38
Nozzles							
Number	1	4	2	2	2	2	2
Size	1"	¾"	¾"	5/8"	5/8"	5/8"	5/8"
Head Pressure (psig)	200	250	215	190	175	150	125
Travel Speed (in./min)		140	100	100	70	70	70
Total per Operation	8 Rev.	81'-0"	81'-0"	81'-0"	81'-0"	81'-0"	81'-0"
Travel per Blow	2 Rev.	79'-0"	79'-0"	79'-0"	79'-0"	79'-0"	79'-0"
Time in Minutes							
Unit Operation	2.67	6.94	9.72	9.72	13.88	13.88	13.88
Unit Blow	0.67	6.77	9.48	9.48	13.54	13.54	13.54
Cycle Operation	164.33	27.60	38.88	58.32	81.24	81.24	108.32
Cycle Blow	163.20	27.08	37.92	56.88	81.24	81.24	108.32
Total Air Requirements							
SCFM/Unit	2,300	6,700	5,000	3,200	2,250	1,500	1,250
SCF/Unit	1,541	45,359	47,400	30,336	30,465	20,310	16,925
SCF/Cycle	221,904	181,436	189,600	182,016	182,790	121,860	135,400

Note:
1. Wallblowers operate on timed start with one (1) unit started each 68 seconds: one unit blowing and three units operating at once.
2. WB = Furnace Wallblower.
   LR = Long Retractable Sootblower.
3. Furnace width = 80'-0". Long retractable blowers are double installation.

Customer Name: Hometown Electric Power Generation Company
Location: Hometown, Texas
Prepared by: WRB
Date: November 15, 1995
Note: AIR FLOW RATES ARE STATED ON THE BASIS OF SCFM DELIVERED AT STANDARD CONDITIONS: 14.7 PSIA, 60 DEG. F, DRY.

Figure 4-2

This document defines the minimum acceptable air flow rate for those long retractable units located in the higher gas temperature zones for lance tube cooling requirements. The remaining flow rates and pressures are those flow rates recommended for effective cleaning of the type of deposition anticipated at the various temperature locations. Once placed in service, these pressures and flow rates will be adjusted upward or downward, depending upon the observed results.

Interpretation of the information on the Air Data Sheet is reasonably self evident with the exception of the basis on which the operating times and blowing times are stated. It is appropriate that we take some time to develop an understanding of the origin of these figures. The first line of the data sheet defines the Model of the wallblower as WB-1, of which there are 144 units, each fitted with one (1), one-inch diameter nozzle. The recommended head pressure, or blowing pressure, is 200 PSIG (13.79 Bar). The travel speed column is left blank because the travel speed of the wallblower is defined in terms of time per revolution rather than in inches per minute. The total travel is stated as eight (8) revolutions, which as stated in Chapter 2 and illustrated by Figure 2-9 is 10 inches (.254 m) to move the nozzle from the at-rest position in the wallbox to its blowing position in the boiler. For 60 Hz. operation, the speed per revolution is 20 seconds and for 50 Hz. is 25 seconds. We then show the time required to complete the operation of one wallblower to be 2.67 minutes, which is eight (8) revolutions at 20 seconds per revolution, or 160 seconds divided by 60 equals 2.67 minutes. The unit blowing time shown as .67 minutes represents two (2) revolutions in the blowing position, each taking 20 seconds for a total of 40 seconds divided by 60 equals .67 minutes. The total cycle time for 144 wallblowers is stated as 164.33 minutes, or two (2) hours and forty-four (44) minutes, rather than as the time required per unit of 2.67 minutes times 144 units, which is 384.48 minutes, or six (6) hours and 24 minutes. This period of time would be totally unacceptable for the completion of one cycle of furnace wallblowers. The solution lies in operating the wallblowers on a "timed start" basis, sometimes called "wallblower speed-up" in which one wallblower is started every 68 seconds. This is the normal increment of time delay for 60 Hz operation. For 50 Hz operation, the timed start increment is normally 86 seconds. In this instance, the time required to complete one cycle of operation is 68 seconds times 144 units, plus 68 seconds for the last unit in the cycle to complete its operation, for a total of 9860 seconds, or 164.33 minutes. The cycle blowing time is the product of two (2), twenty (20) second revolutions plus an elapsed time of twenty-eight (28) seconds between the end of blowing one unit to the start of blowing the next unit. This is a total of sixty-eight (68) seconds times 144 units, or 9792 seconds divided by 60 or 163.2 minutes.

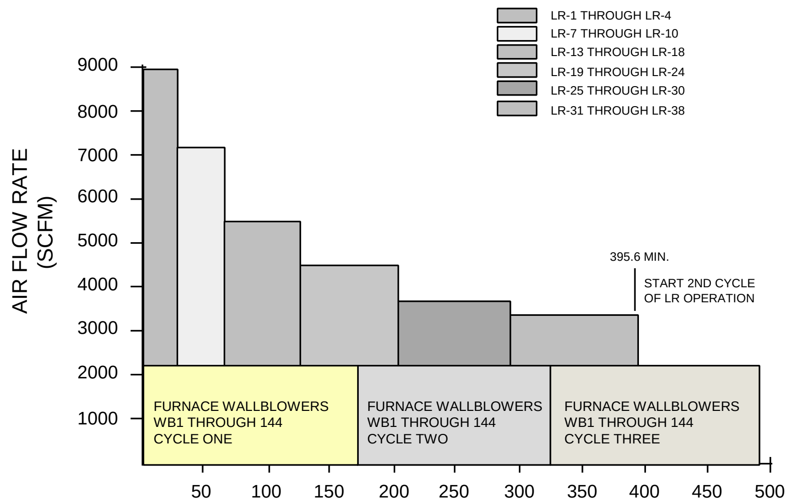
The unit flow rate is 2300 SCFM (1.25 cubic meters), which when multiplied by the unit blowing time of .67 minutes results in 1541 SCF (50.25 cubic meters) required per unit. The total air required per wallblower cycle is 144 times the air required per unit or 221,904 Standard Cubic Feet (7236 cubic meters).

The next line of the Air Data Sheet identifies the long retractable units as being Model LR-2 units with four (4) units in the first group. The long retractable units are arranged in groups on the data sheet depending upon the gas temperature zone in which they are installed. The data sheet tells us that this group of blowers consists of four (4) units equipped with four (4), ¾--inch nozzles to be operated at a blowing pressure of 250 PSIG (17.25 Bar). The specified travel speed is 140 inches per minute, and the total operating travel is 81 feet (24.68 m).

The data in Figure 4-1 tells us that the furnace width is 80 feet (29.38 m) and Figure 2-11 in Chapter 2 shows the nozzle of the long retractable unit retracted approximately 6 inches (.152 m) into the wallbox when fully retracted. To complete one operation, the blower must travel this six inches (.152 m) to the center line of the wall tubes then to the centerline of the boiler, which totals 40 feet, 6 inches (12.344 m) of forward travel plus 40 feet, 6 inches (12.344 m) of reverse travel to be fully retracted from the boiler on completion of its operation. Since the unit does not start blowing until the nozzle is extended 6 inches (.I52 m) into the boiler and stops blowing when the nozzle is retracted to within 6 inches (.152 m) of the centerline of the wall tubes, the blowing traverse is 79 feet (24.079 m). The time required per unit operation of 6.94 minutes is the total distance traveled, or 81 feet (25.063 m) times 12 inches (305 m) divided by the travel speed of 140 inches per minute (3.556 m/min or .059 m/sec). Likewise, the unit blowing time is 79 feet, 0 inches (24.079 m) multiplied by 12 and divided by the travel speed to result in 6.77 minutes per unit. The unit air flow rate of 6700 SCFM (3.65 m/sec.) multiplied by the unit blowing time of 6.77 minutes results in 45,359 SCF (1243.43 cubic meters) of air required per blower. Therefore, the four retractable units in the group will use 181,436 SCF (5137 cubic meters) of air.

The remaining document to consider is that of the Sequence Bar Chart, which is a plot of air flow rate versus time in minutes.

SINGLE BLOWER INSTALLATION
AIR SOOTBLOWING SYSTEM SEQUENCE BAR CHART



Customer Name:	Hometown Electric Power Generation Company
Location:	Hometown, Texas
Prepared by:	WRB
Date:	November 15, 1995

Note: Three (3) cycle of wallblower operation completed in 495 minutes provide for just slightly less than nine (9) cycles per day.

Long retractable units operated simultaneously with wallblowers complete one cycle in 396 minutes, providing for 3.6 cycles per day.

Figure 4-3

The purpose of the Sequence Bar Chart is to make provisions for the most logical sequence of sootblower operation to most effectively clean the heating surfaces of the boiler and to make the best possible use of the air compressors supplying air to the system. Having said that, it is most important that we emphasize that the top priority to keep in mind when attempting to predict a sequence of sootblower operation is that the cleaning requirements of the boiler must come first. There have been a number of instances in the past in which the sequence of sootblower operation has been arranged to best suit the performance characteristics of the compressors. This is unfortunate, and will generally result in an unsatisfactory installation. While the fouling tendencies or cleaning requirements of the boiler are practically unpredictable, there are several somewhat standard considerations given to the initial sequence of sootblower operation. This common logic suggests that it makes sense to sequence the sootblowers in such a way that the particulate blown loose from the heating surface will be picked up in the flue gas stream and carried through the boiler. In keeping with this logic, most sequence bar charts will deal with the furnace wallblowers first.

Referring to the air requirements stated for the WB furnace wallblowers in Figure 4-2, we see that we could operate one complete cycle of wallblowers at the beginning of the sequence in just a little over 164 minutes, or about 2 hours and 44 minutes. However, if we were to do that it would take approximately 6 hours and 33 minutes to operate the remaining long retractable blowers, one at a time. While we may not know exactly how often it might be necessary to clean the furnace walls, it would seem logical to provide for repeat operation of the wallblowers in something less than a six-hour period. Rather than place the operator in a position of having to interrupt the operation of long retractable blowers to repeat the operation of furnace wall blowers, we have, as have the majority of clients, made provisions for continuous operation with the long retractable blowers operated simultaneously with the wallblowers. As noted on our Sequence Bar Chart, this allows one cycle of wallblowers to be completed in 164.3 minutes. This would allow nearly three cycles to be completed in an eight (8) hour operating shift, or nearly nine (9) cycles per day.

In the example identified as Figure 4-3, we have chosen to operate all of the long retractable blowers, one at a time, and in a sequence that starts with the retractable blowers in the secondary superheater, followed by groups of blowers moving toward lower gas temperature zones, or toward the boiler gas exit. This would allow the operator to be selective and to operate any retractable blower at any point in the sequence that he might choose. In this case, the maximum air demand would be the first group of four retractable blowers identified as LR- 1 through 4 at 9000 SCFM (4.89 m^3/sec.). Reference to the Air Data Sheet, Figure 4-2, will show that this maximum air demand is the sum of 2300 SCFM (1.25m^3/sec.) required of the wallblowers and 6700 SCFM (3.64m^3/sec) required of retractable blowers LR-1 through LR-4.

At this point, it is appropriate to point out that the Sequence Bar Chart is something of a misrepresentation of the air demand. This is understandable if we remember that the chart is a plot of the required air flow rate over a time period required for operation of the sootblowers and not for the blowing time. In reality, the actual air usage is intermittent over the period of time plotted for the unit operating time. This is best understood by re-examining the time increment required for completion of one cycle of wallblower operation. In this case we have stated the cycle blowing time to be 163.2 minutes; however, the wallblowers do not require a constant air flow of 2300 SCFM (1.25 m3) over that period of 163.2 minutes. We mentioned that there was a 28 second elapsed time period from the completion of blowing of one unit until the start of blowing of the next unit in the sequence. If we were to represent this graphically on the sequence bar chart, we would show 144 bars, each of 40 seconds' duration with an elapsed time of 28 seconds between each of the 144 bars. This increment of elapsed time is added to avoid having more than one unit blowing at a time. This recognizes the fact that there is some lag time, both electrical and mechanical, in the transfer time from one unit to the other.

The long retractable units also have an elapsed time between the end of blowing one unit to the start of blowing the next unit. This is equal to the time required to retract the nozzle from the point at which it stops blowing to its at-rest position in the wallbox, plus the time required to extend the nozzle of the subsequent unit approximately 6 inches (.152 m) into the boiler, where the inlet valve on the blower is opened. This is generally equal to 24 inches (.069 m) of travel divided by the travel speed, which in this case of a 100 inches / min.(152.4 m/s) would be .24 minutes, or about 14.4 seconds. This period of elapsed time will also vary depending upon both the electrical and mechanical lag time that exists during the transfer of operation from one blower to the next.

Referring to Figure 4-3, which we have identified as a single boiler installation, provisions are made for three (3) cycles of long retractable blowers per day and either continuous operation or selective operation of furnace wallblowers up to as many as nine (9) times a day. We must now select the most logical size of compressors to meet this requirement. In the majority of cases, the user would simply assume that he should install one compressor of 100% capacity to meet the maximum flow rate, with a second compressor of equal capacity to serve as a stand-by, or back-up compressor. In this case, two (2) units, each with a capacity of approximately 9000 SCFM (4.89 m /sec.), would be required.

Date of Issue: 4-Oct-1996

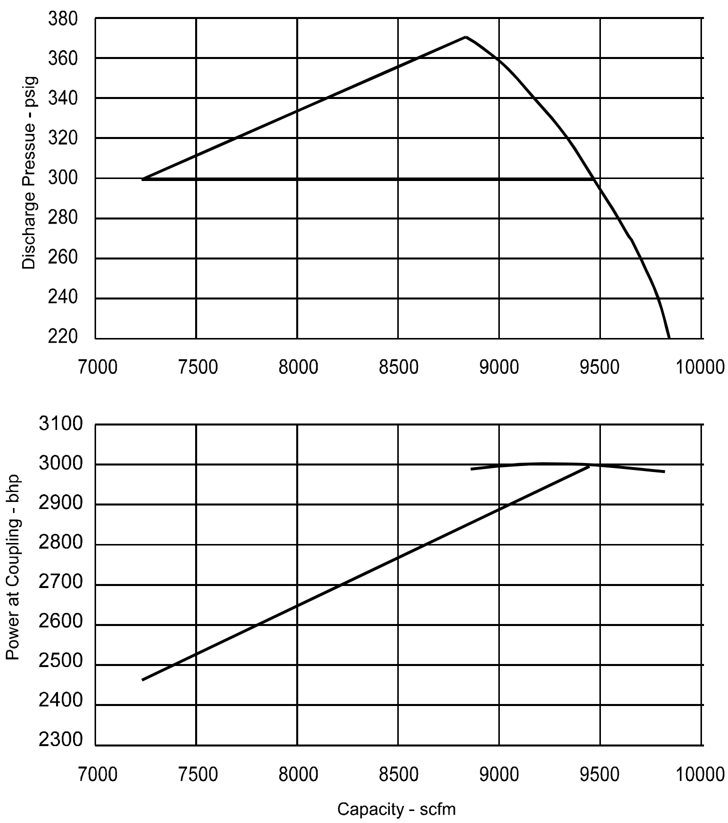


Figure 4-4

To give proper evaluation to this selection of compressors, we refer to Figure 4-4's plot of the performance of a 4C110M5 unit at standard ambient conditions, which is to deliver approximately (9474) SCFM (5.04 m^3/s) at 300 PSIG (20.7 Bar) with a design power requirement of 3006 BHP. The plotted curve shows a stable pressure rise to surge of 22.46% , resulting in a turndown at the design pressure of 300 PSIG (20.7 Bar) of 23.6%. This is a favorable performance characteristic for this application in that we want the greatest capability possible of modulating the flow of the compressor. In this case, we could throttle the flow of the compressor down to 7236 SCFM (3.85 m^3/s) before having to bypass air or to unload the compressor. However, reference to our sequence bar chart, Figure 4-3, shows that the air demand is only above 7236 SCFM (3.85 m^3/s) while the long retractable Units 1 through 4 are being operated simultaneously with the furnace wallblowers. This indicates that the 100% capacity compressor would either be blowing off air to the atmosphere and wasting considerable electrical power or would be running unloaded for the better part of the time. This observation suggests that it might be appropriate to look at three lesser capacity machines that could be operated more efficiently over a wider range of the estimated air demand, thereby conserving considerable electrical power.

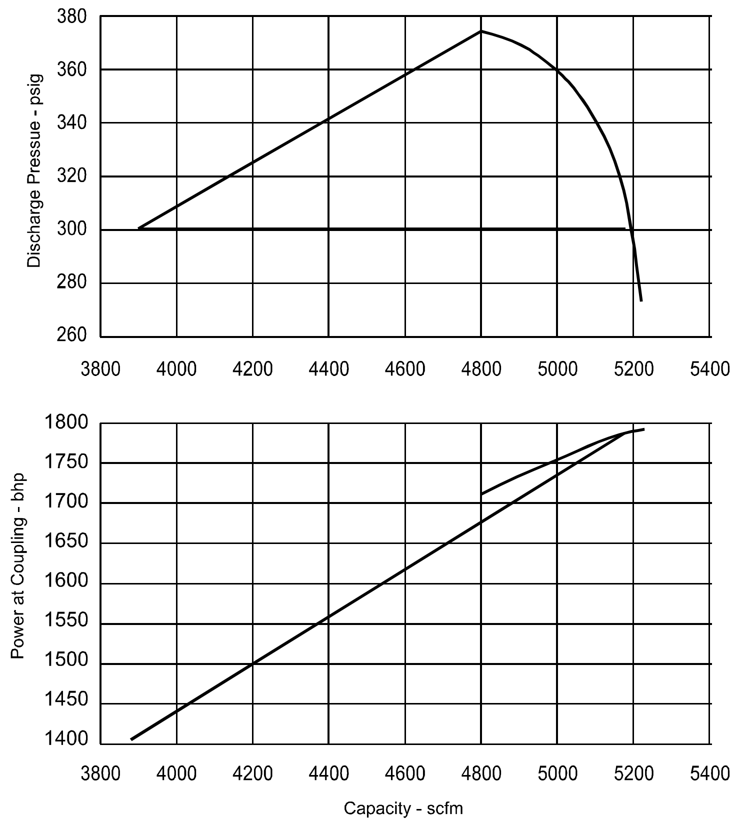


Figure 4-5

Figure 4-5 plots performance of a 3C60M5 unit that at standard conditions is to deliver 5189 SCFM (2.82 m^3/sec.) at 300 PSIG (20.70 Bar), which is about 57 % or 58% of the maximum air demand shown on the Sequence Bar Chart, Figure 4-3. This compressor selection would require Units A and B to operate together, each within its throttle range, to satisfy the maximum air demand while Unit C was either unloaded or idle in a standby operating mode. In this case, Unit A would be designated the “Lead” compressor, Unit B the “Lag” compressor, and Unit C the “Standby” unit. Figure 4-5 shows that this selection also has a high stable pressure rise to surge resulting in a good throttle range that allows the machine to be throttled back to 3884 SCFM (2.08 cubic meters per sec.) before unloading or blowing off air to the atmosphere. It is obvious that Lead and Lag compressors could be operated together to satisfy the indicated air demand somewhat more efficiently than the 100% capacity compressors. The operation of multiple compressors to satisfy such an air demand is made considerably more efficient by the application of suitable compressor controls. In this case, it would be most desirable to apply the Centac Energy Master (CEM) to control the three compressors. Of the many control features made available by the CEM, which we will address later in this text, the feature best suited to this multiple installation of compressors is the load sharing capability. The load sharing of this predicted air demand would be between the Lead compressor, Unit A, and the Lag compressor, Unit B. Load sharing is accomplished as the CEM senses the position of the inlet valve of the Lead compressor and sends a signal that duplicates that inlet valve position to the Lag compressor, resulting in equal sharing of the load.

As long as the total air demand is within the throttle range of both the Lead and Lag compressors, the CEM will send a new valve position signal every 10 seconds, assuming that the air demand has changed to require either more or less flow. After 20 seconds of not receiving a new valve position signal, the Lag compressor will resort to its own control. In this case, the Lag compressor would unload because the Lead compressor has sufficient capacity to satisfy the air demand. When the Lag unit unloads, the CEM stops sending valve position signals for a period of two minutes to allow the system to stabilize. The CEM then resumes sending signals to the Lag compressor so that it will re-load on an increase in air demand to again share the load with the Lead compressor. In the event that the air demand should not increase beyond the capacity of the Lead compressor for a pre-determined period of time, the CEM will shut down the Lag compressor. This prevents the Lag compressor from continuing to run unloaded and wasting power, while the air demand remains within the capacity of the Lead compressor. The control system automatically monitors the number of compressor stops and starts to avoid exceeding the number of cold and hot starts allowed daily by the compressor motor manufacturer. After a preset period of time, the CEM will rotate the assignment of the Lead, Lag, and Standby units in an effort to keep the running times of the compressors approximately the same.

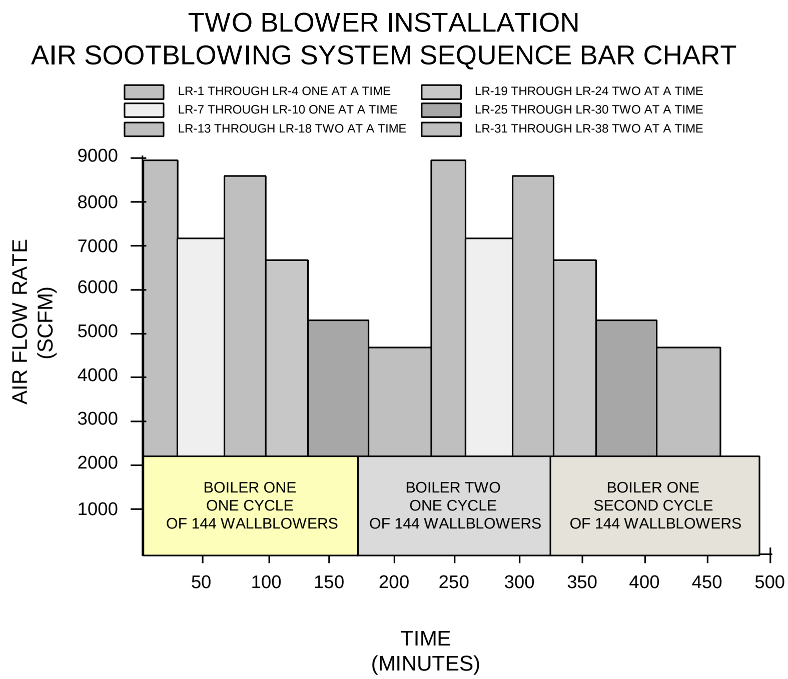
In the case of the lesser capacity compressors described by Figure 4-5, we could expect the CEM to share an air demand of over 5189 SCFM (2.82 m^3/sec) between the Lead and the Lag compressors and allow the Lag unit to unload as the Lead compressor would supply the demands from 3884 SCFM (2.11 cubic meters per sec.), the throttled capacity, and 5189 SCFM (2.82 cubic meters per sec.), the design capacity of the Lead unit.

While the initial cost of three lesser capacity units is going to be somewhat greater than the initial cost of two 100% capacity compressors, the reduced cost of operation and maintenance over the life of the equipment will normally justify the selection of the three lesser capacity compressors for an installation of this kind.

Multiple Boiler Installations:

Another situation to consider is the selection of compressors that might be applied to a multiple boiler installation rather than a single boiler installation. Using the same air requirements as shown on the air data sheet, Figure 4-2, we might look upon these as being the per boiler requirements for the installation of two duplicate boilers. In this case, we could probably expect the boiler supplier and the sootblowing equipment manufacturer to recommend a somewhat different Sequence Bar Chart than that shown for a single boiler installation in Figure 4-3.

The objective would be that of compressing the cycle time by operating some long retractable units two at a time in order that one cycle of operation of all blowers might be completed on both boilers within a close approximation of eight (8) hours. This would most likely be accomplished as illustrated in Figure 4-6 and as described in the following:



Customer Name:	Hometown Electric Power Generation Company
Location:	Hometown, Texas
Prepared by:	WRB
Date:	November 15, 1995

Figure 4-6

Operate furnace wallblowers simultaneously with long retractable blowers to provide three cycles per shift of operation. This will provide time for approximately eight (8) cycles per day or approximately four (4) cycles per day for each boiler. Operate long retractables as follows:

UNITS	**OPERATE**	**TIME IN MIN (SEC)**	**TIME IN MIN (SEC)**	**AT SCFM (M^3/S)**
LR 1 - 4	1 at a time	From 0 (0)	To 27.6 (1656)	9000 (4.89)
LR 7 - 10	1 at a time	From 27.6 (1656)	To 66.4 (3984)	7300 (6.95)
LR 13 - 18	2 at a time	From 66.4 (3984)	To 95.5 (5734)	8700 (4.73)
LR 19 - 24	2 at a time	From 95.5 (5734)	To 137.14 (8228)	6800 (4.67)
LR 25 - 30	2 at a time	From 137.14 (8228)	To 178.78 (10,727)	5300 (2.88)
LR 31 - 38	2 at a time	From 178.78 (10,727)	To 234.3(14,058)	4800 (2.61)

The same units on number two (2) boiler could be operated .for one complete cycle from 234.3 minutes to 486.6 minutes. This would extend the blowing cycle for both boilers one and two just 6.6 minutes over the 8 hour operating shift.

It is obvious that the compressor selections previously made for the single boiler installation will also meet the requirements of the two (2) boiler installation. However, with two long retractable units operating simultaneously with the wallblowers, the required flow rates are high enough to keep one 100% capacity compressor loaded for approximately 3 hours and 11 minutes of the operating shift, after which the total air demand of long retractable units 19 through 38, on both boilers, would fall below the throttled capacity of the selected 100% capacity compressor. Under the circumstances, the selection of the three (3) lesser capacity compressors can be justified provided the purchaser evaluates the operating and maintenance costs of the machinery over the life of the equipment. Such considerations as these are not likely to be taken into account by the average equipment purchaser unless they are brought to his attention by the I-R representative.

Air Receiver Sizing:

The remaining concern is that of determining a suitable requirement of receiver volume for the sootblowing air system. In the installation of centrifugal compressors, the air receiver serves both to stabilize the header pressure while the compressor, or compressors, are loading and unloading, and also to isolate the compressor from pulsations of pressure resulting from rapid changes in flow demands. In the case of an air blowing sootblowing system, these pulsations of pressure are the result of the rapid opening and closing of inlet valves at each sootblower unit at time intervals corresponding to the start and finish of blowing of each unit. As pointed out earlier, the elapsed time interval between the end of blowing one unit to the start of blowing the next unit is only a matter of seconds, but it is sufficient to create pulsations in the header that must be dampened by adding receiver volume to prevent the unstable header pressure from interfering with the control of the compressor. It is for this reason that the pressure-sensing devices from which the compressors are controlled are always installed between the compressor discharge and the air receiver inlet.

The air receiver capacity required to stabilize the header pressure is a function of the time that is required for the compressor to load or unload.

This time increment is largely dependent upon the response time of the inlet and blow-off valves to move from full open to closed or from closed to full open. For the air-actuated butterfly type valves commonly applied with the CENTAC, this valve response time varies from 4 to 8 seconds. Taking this into account and allowing a little added time to stabilize the pressure, we recommend that the receiver be sized to supply the maximum air demand for a period of 12 seconds or one-fifth of one minute. In the case of the system that we have used as an example with a 9000 SCFM (4.89 m^3/s) maximum demand, we would ask for a minimum of 1800 cubic feet of receiver volume. It is well to repeat that this is a minimum requirement. Added air receiver volume is in no way detrimental to the operation or control of the air system.

Added Compressor Control System Features:

In the case of the multi-unit installation of compressors for boiler sootblowing systems, as with any other applications of multiple compressors, the careful selection of the compressor control system is of paramount importance. In addition to having provided the necessary monitoring, permissive and fault alarms to protect the compressors, the control system should be carefully selected to respond to the changing air demands of the system in such a manner that will utilize the installed compressors in the most efficient way possible. Each CENTAC is normally equipped with a local compressor- mounted control panel. This panel’s functions include all of the commonly applied permissive start functions and fault alarms required to protect the compressor and to annunciate vital system operating conditions. In the case of single unit installation, this local control panel is normally quite adequate for most applications. Multiple unit installations of compressors are greatly enhanced by the addition of a control capability that will allow all of the compressors in the system to respond to changing conditions and varying air

Date of Issue: 4-Oct-1996

demands in such a way that all compressors will be utilized most efficiently. This is accomplished by applying the Centac Energy Master (CEM), which is a state-of-the-art compressor management system designed for the control and protection of multiple air compressors. The CEM system utilizes the compressor-mounted control panel at each unit connected to the CEM host computer. The CEM software provides an advanced compressor protection package as well as the control capabilities required to facilitate stable and efficient compressor control, even under constantly changing operating conditions.

Earlier in this text we described both the function of load sharing between multiple compressors to optimize the system efficiency and also the capability of rotating units to alternate the assignment of Lead and Lag compressors in an effort to equalize operating time and thereby minimize maintenance and operating costs. Load sharing is but one of many features available to the user with the CEM software. It is most important that the IR representative be totally aware of these control features to be presented for the consideration of the purchaser of multiple units of centrifugal air compressors.

Other Compressor Applications Within The Power Generation Market

The Power Generation Industry worldwide represents a major market for compressed air for applications other than sootblowing service. Ingersoll-Rand has enjoyed considerable success in this marketplace over the years and continues its emphasis upon its leadership role in responding to the compressed air needs of this industry on a worldwide basis.

The following is intended to give brief explanation of the various uses of compressed air within the industry for which Ingersoll-Rand has products to offer within appropriate ranges of air flow and pressure.

Station Air Compressors:

Every electric power generating facility requires station air compressors, sometimes referred to as plant air compressors. These compressors are normally applied at pressures ranging from 100 PSIG (6.89 BAR) to 125 PSIG (8.62 BAR) and at delivered air flows that might range from as little as 500 SCFM (16.32 cubic meters per min.) to as much as 10,000 SCFM (326.43 cubic meters per min.),depending largely upon the size of the facility and, to some extent, upon the design of the generating units.

Station air compressors are installed primarily to supply general purpose air for a multitude of uses such as for general housekeeping and sometimes for the operation of air-operated tools. As such, the applications for station air normally will not require oil free air. Consequently, the smaller electric generating facilities may find it economically advantageous to apply small rotary or reciprocating rather than centrifugal compressors. On the other hand, larger multi-unit generating facilities may have a need for station air in the range of 4000 SCFM to 10,000 SCFM (130.57 cubic meters per min. to 326.43 cubic meters per min.) or even more, in which case the installation of centrifugal compressors are generally preferred. Additionally, in the case of larger, multi-unit facilities, it is common practice to use a portion of the station air to serve as either a primary source or a back-up source of air for lesser flow requirements of oil free air.

The sizing of station air compressors for a new generating facility is generally defined by either the end user or the responsible engineering firm and is normally based upon the user’s or the designer’s past experience.

Control Air Compressors:

Control air compressors, sometimes referred to as instrument air compressors, are normally applied at pressures ranging from 100 PSIG (6.89 BAR) to 125 PSIG (8.62 BAR) at delivered flow rates ranging from as little as 500 SCFM (163.21 cubic meters per min.) to as much as 3000 or 4000 SCFM (97.93 or 130.57 cubic meters per min.), depending upon the size and design of the facility. In years past, it was quite common to design large central station generating facilities using pneumatically operated and controlled instrumentation for control and monitoring of both the steam generating boiler and the electric power turbine generator.

In more recent years the trend has been toward the use of solid state or microprocessor controls for this purpose, however, there are still certain circumstances under which the end user or the design engineer will maintain a preference for the use of either a complete or a partial pneumatic control system.

When required the control air compressors must supply clean, dry, oil free air. This air may be supplied by dedicated control air compressors, or the control air may be extracted from the station air compressors and given the necessary treatment at the point of extraction to meet the control air quality requirements. In this instance, the station air compressors discussed in the foregoing, are likely to be specified as oil free compressors.

Pneumatic Conveying Of Fly Ash:

The current worldwide emphasis upon environmental control imposes stringent clean air requirements upon both existing and newly designed electric power generating facilities. The most common modification made to both new and existing facilities involves the addition of various devices designed to remove fly ash particles from the boiler's flue gasses. This separation of fly ash particles from the gas stream takes place before the gasses are discharged to the atmosphere as a part of the effluents discharged from the stack of the steam generating unit. The volume of ash particulate entrained in the boiler flue gasses is related proportionally to the ash content of the coal that fuels the boiler. In order to comply with the stringent clean air requirements imposing limitations upon the quantity of sulfur dioxide (SO_2) contained in the emission from the stack, many operating companies have chosen to modify their steam generating units to accommodate the burning of low rank coal with an acceptably low sulfur content. While this change of fuel will normally meet the required limitation of SO_2 discharged into the atmosphere, it does, as discussed earlier in this text, add to the boiler cleaning requirements because nearly all low sulfur coals have a high ash content. The high ash content also adds to the quantity of ash particulate per unit volume of flue gasses that are carried through the boiler and to the unit's discharge stack.

While several means of separating the fly ash particulate from the boiler gases are available, the most common practice is to install an electrostatic precipitator between the final heat exchanger and the inlet to the smokestack to clean the gases before they are exhausted to the atmosphere. The flyash is collected in hoppers beneath the precipitators then conveyed to large silos, where it is loaded into trucks and hauled to landfills or transported to other facilities where it might be used as an aggregate in highway resurfacing material or for other purposes.

Largely due to the limitation of space beneath the boiler, the silos in which the fly ash is stored are generally located a considerable distance from the hoppers in which it is collected from the electrostatic precipitator. This requires some means of conveyance between the collection hoppers and the silos. An open conveyance system consisting of conveyor belts or other means of mechanical movement is not acceptable for this application, as such a system would also allow the very fine fly ash particles to spill over into the surrounding atmosphere. The most suitable solution is a closed conveyance system that uses single stage centrifugal compressors to provide the motive air for the pneumatic conveyance of the fly ash particles.

The number of compressors required per boiler will vary depending upon the size and physical configuration of the precipitator. A single section precipitator with as many as twenty (20) collection hoppers feeding into a common conveying line may be sufficiently large to handle the fly ash from one boiler.

In such a case; the conveying air system would normally consist of two (2) compressors, one sized for 100% of the required capacity and a second unit of the same size to serve as a stand-by unit. A larger boiler might require a precipitator with two (2) individual sides, each with as many as twenty (20) hoppers feeding into two (2) conveying lines. In this case, the conveying air system would consist of three (3) compressors, two (2) units sized for 100% of the required capacity with a third unit of the same size to serve as a common spare for both conveying compressors. In the case of a multiple boiler power generating station, each boiler would require its own precipitator and pneumatic conveying system.

The sizing of the compressor is dependent upon the rated size of the conveying system in terms of tons per hour of particulate to be conveyed, the diameter of the conveyance line, and the distance over which the material is to be transported. Ingersoll-Rand's Single Stage CENTAC or X-Flo products can be selected to meet these requirements with discharge pressures ranging from 12 to 35 PSIG (.829 t o 2.41 BAR) and delivered flows ranging from 300 to 75000 ICFM (8.5 to 2123.7 cubic meters per min.).

Flue Gas Desulpherization:

Flue gas desulphurization is a process in which the flue gases produced by the combustion of fuel in a coal fired boiler are treated to remove harmful pollutants, primarily sulfur dioxide (SO_2), before discharging the effluents from the smokestack to the atmosphere. Wet scrubbing is the process by which the flue gases are treated with a limestone slurry to form a calcium sulfite. The calcium sulfite is then oxidized by adding air from an oxidizing air compressor. The oxidation serves to convert the sulfites to sulfates to form calcium sulfate, known commercially as gypsum. While the primary purpose of the wet scrubber is to remove sulfur dioxide (SO_2) from the stack effluents, there is a marketable by-product in the form of gypsum that serves to defray a portion of the cost of the wet scrubber.

A typical process schematic of a Wet Scrubber Flue Gas Desulphurization System is illustrated in Figure 5-1.

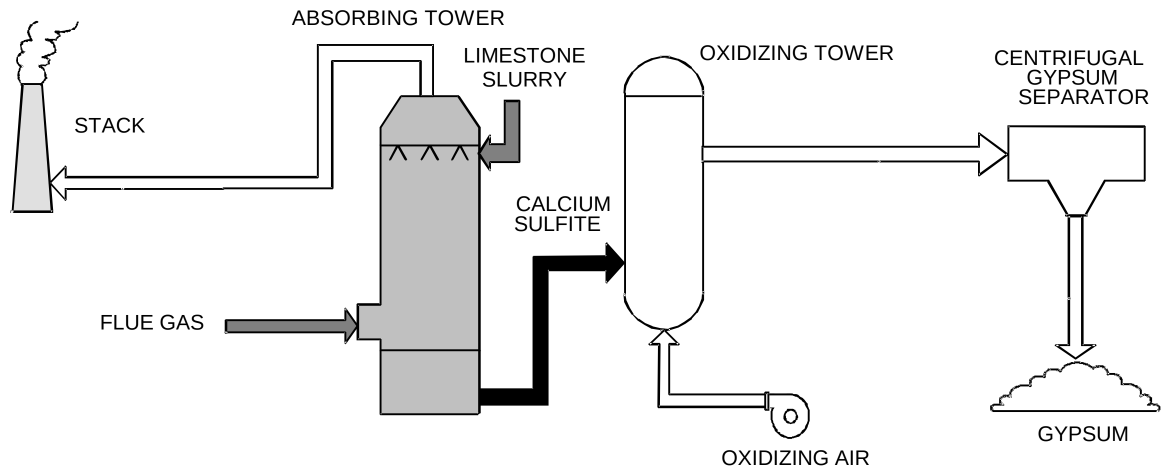


Figure 5-1
Typical flue gas desulphurization process

As previously stated, the majority of utility operators have chosen the alternative of modifying their boilers to accommodate the burning of low sulfur, high ash content fuels to meet the clean air standards regulating the emission of sulfur dioxide (SO_2). This is a popular choice because the conversion to a low rank fuel is a presently available and proven technology by which the user can expect to reduce the emission of sulfur dioxide (SO_2) by 50 to 80%, compared to that of a high sulfur coal. The application of wet scrubbers for flue gas desulphurization is proven to be somewhat more efficient in terms of removal of sulfur dioxide (SO_2) from the stack effluents; however, the capital cost as well as the space required for the installation of such a system is often found to be prohibitive. None the less, the use of wet scrubbers to remove sulfur dioxide (SO_2) from the flue gasses is a viable alternative and is a commercially available, proven technology.

The air compressor pressure and capacity required of the wet flue gas desulphurization system varies with the size and design of the scrubber but, for the most part, will be within the following ranges:

Capacity:	From 2000 to 20,000 ICFM (57 to 570 m^3/min.)
Pressure:	6 to 18 PSIG (0.4 to 1.24 BAR)

The compressed air requirements of these systems are best suited to the application of Ingersoll-Rand's X-Flo design which can be expected to outperform competitive offerings in the hostile environments in which such compressors must operate.

There are other types or designs of flue gas desulphurization systems available; most of them are presently being evaluated.

Mechanical Vapor Recompression (Mvr):

While we are generally conditioned to think in terms of supplying equipment to compress air, we must also recognize the capability of applying single stage centrifugal compressors for the compression of steam as required by the Mechanical Vapor Recompression (MVR) process. There are several applications within the fossil fueled and the nuclear fueled Power Generation Industry utilizing MVR evaporation systems that represent a market potential for Ingersoll-

Rand's single stage X-Flo product line. A brief definition of these applications is offered in the following:

1. Cooling Tower Blowdown:

In both nuclear and fossil fueled power generation plants, MVR evaporation systems are being used in the cooling tower blowdown stream. This system provides additional water for re-use as feed water make-up for the steam generating unit and also serves to reduce the volume of blowdown water, which contains small quantities of damaging salts, normally disposed of in large waste ponds. The application of an MVR system can greatly reduce these disposal costs as well as the steam generator's feed water make-up requirements with a resultant reduction in the cost of operating the steam generating unit.

2. Concentration Of Radiation Waste:

In nuclear fueled steam generating plants, MVR evaporation technology may be used to concentrate low-level radioactive waste. The concentration process separates this waste into a thick liquor that is highly radioactive and is nearly pure condensate, which can usually be vented to the atmosphere with no implications. The disposal volume and, consequently, the disposal cost is greatly reduced by the concentration of the radioactive waste.

3. Processing Nuclear Fuel:

An MVR evaporation system has been installed at the Kerr-McGee Nuclear Corporation in Gore, Oklahoma. This system, utilizing a single stage centrifugal compressor, is used to concentrate uranyl nitrate solution. The uranyl nitrate solution is processed further to produce uranium hexafluoride (UF_6), which is used in the manufacture of nuclear fuel. The steam capacity range of the compressors applied for this purpose typically varies from 5000 to 70,000 CFM with inlet pressures near atmospheric conditions. To understand the principles of operation of the MVR evaporation system, it is helpful to have an understanding of the thermodynamics involved in the process. A suitable explanation of these principles is contained in the publication, "Industrial Steam Compression," authored by George J. Vince and is available from Ingersoll-Rand.

Glossary Of Terms And Phrases

Evaporation Rate -------- Term used to define the capacity of a steam power generation boiler, expressed in #/hr. of steam.

Critical Pressure --------- 3208.2 PSIA and 705.5^{0}F. (231.2 BAR and 374.17^{0}C).

Steam Quality ------------ The weight fraction or percentage of steam in a water-steam mixture.

Saturated Steam --------- Steam at saturation temperature for a given steam pressure.

Superheated Steam ------ When heat is added to saturated steam, out of contact with liquid, it is said to be superheated.

Turbine Generator Capacity ------------------- Expressed in MW (megawatts) of electric power.

Dry Bottom Furnace ---- Phrase used to describe the furnace design, particularly applicable to boilers burning coals with high ash fusion temperatures, provided with a hopper bottom and with sufficient cooling that ash on the furnace wall or hopper bottom is solid and can be removed as dry ash particles.

Wet Bottom or Slag Tap Furnace -------------- Phrase used to describe the furnace of a boiler designed to burn a low ash fusion temperature coal in which the slag in the furnace bottom tends to be molten or sticky. From the lower furnace, slag drops in liquid form to a floor below where a pool of liquid slag is maintained and tapped into a slag tank containing water where it is solidified for removal and disposition.

Boiler Tube Bank ------- A section of boiler tubes arranged in multiple horizontal rows of tubes. The depth of the tube bank depends upon the center to center vertical spacing and the number of rows of tubes in the tube bank.

Boiler Cavity ------------- The height of the spacing between horizontal tube banks, normally provided for the installation of long retractable sootblowers.

Economizer --------------- An arrangement of boiler tubes, just preceding the boiler exit where heat from the boiler flue gasses leaving the boiler is recovered to pre-heat the boiler feedwater before it enters the furnace wall portion of the water/steam vapor cycle.

Reheater ------------------- A section of the boiler, normally located between the primary and secondary superheaters, in which steam exhausted from the high pressure stage of the turbine is reheated and returned to a low pressure stage of the turbine generator.

Superheater --------------- A section of the boiler located nearest the furnace exit, designed to transfer heat from the high temperature flue gasses to elevate the temperature of the steam to the level of superheat required by the turbine generator.

Airheater ------------------ A heat recovery device installed between the boiler exit and the stack to which air is introduced in a direction counter to the flow of flue gas. The heat transferred from the flue gas stream preheats air that is supplied to the burner windboxes to improve the efficiency of fuel combustion.

Furnace Width ----------- The dimension from the center line of the water wall tubes on one side of the boiler to the centerline of the walltubes on the opposite side of the boiler.

Single Installation
Of Sootblowers ---------- Long retractable sootblowers installed on only one side of the boiler with lance tube travel equal to the entire width of the boiler.

Double Installation
Of Sootblowers ---------- Long retractable sootblowers installed on both sides of the boiler with lance tube travel equal to one half of the furnace width.