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Mechanical Option

Penn State University

M Resort Spa and Casino

Henderson, Nevada

Mechanical Depth: Central Plant Optimization Study

Breadths: Electrical and Acoustical

Faculty Advisor; William P. Bahnfleth

April 7, 2009

M

resort spa casino
henderson, las vegas

project information

size	390 room tower, 35 suites, 92,000 sq. ft. of casino area, a 60,000-sq. ft. conference room, a 23,000 sq.-ft. health spa and salon, restaurants, bars, and a 100,000-sq.-ft. pool area.
floors	14
project cost	$1 billion
construction time	spring 2007 - march 2009
delivey method	design build

mechanical

The campus is served by a central utility plant(CUP). this includes 3900 tons of cooling and 46,800 mbh of heating. The low rise portion will be served by roof top air handling units and the guest suites will be served by vertical fan coil units.

project team

owner	marnell corrao associates
architecture	marnell architecture
management	marnell corrao associates
mechnical	southland industries
plumbing	jba consultanting engineers
electrical	jba consultanting engineers
lighting	cd+m

architecture

This building is comprised of 14 stories, wrapped in a non-reflective glass blanket, that rise above the las vegas strip, creating a campus including casino, spa, restaurant retail,suites, along with many other luxurious amentities. The roof is comprised of a 4" heavy weight concrete post tensioned slab with 6" of insulation.

structural

This building is classified as a type 1a structure. The first floor of the low rise is cast in place concrete. the second floor uses structural steel and spray on fire proofing. The tower is also cast in place concrete with post tentioned floor slabs. Columns are set upon large spead footings throughout the building.

electrical

12470v service is brought in to a substation where it goes through a redundant switch and is then fed to the various electrical rooms. (2) 2000kW/2500kVA electrical emergency generators , 480/277V 3Ø. Lighting fixtures are varied, but florescent is typical.

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http://www.engr.psu.edu/ae/thesis/portfolios/2009/tmc5003/

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Acknowledgements

First and foremost I would like to thank Marnell Corrao Associates for their assistance and cooperation throughout this project. Without their permission to use the M Resort for my senior thesis I really would not be where I am today. The information and guidance they provided have proved to be invaluable.

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Attaining information various equipment is not always an easy task. Many times it seems that companies try to transfer a call between departments for fun. However Tim Robinson of Carrier Corporation and Bernie Pfeifer of Solar Turbines, were both extremely helpful and very quick to answer my questions.

I cannot forget to acknowledge the help and support of my family both immediate and extended. I would like to thank my mother and father for that night in January of 1985, without it I would not be here today. They have always supported my decisions and have always been there to help me if and when I was in need. I have learned a great deal from my father that defiantly has given me an edge in this industry. My mother has been just as instrumental at instilling the qualities that shine through me today. Nothing could trump the level of common sense they built in me over the years. My brother Dan has always been there for me as well, whether it means punching me in the face or picking me up when I need a lift. He also jumped with me on my first skydive when no one else would even think about going. My sister Ann and her new husband Brady have always been very supportive. I have known Ann a bit longer but I am learning more about Brady and can’t wait to welcome their new child into this world.

The friends that I have met at college and in this world in general also deserve to be thanked especially those whom I have lived with here at college. Scott Earley, I would like to thank you for all your help, and especially for introducing me to the world of hunting and butchering. Scott Garley, I got nothing, other than my impeccable driving skills. David Miller, you have always been there, well most of the time, and have given me great advice, well most of the time. David, my grandmother is still waiting for that phone call. Dominic Manno, I don't really know where to start, most of the things you say confuse me or are just plain unrecognizable, but you have introduced me to weight lifting and have been able to put up with my sarcasm thus far, we will see how it goes. Chris Conrad, I missed living with you this year and hearing some of the most random and inappropriate statements ever made. Some of the conversations that were had with these four will, for better or worse, live in my memory forever. I would also like to thank everyone not directly mentioned in this acknowledgement for always being there, especially LC, we have had some fun times.

Finally, I would like to thank everyone at the Chambersburg Skydiving Center for helping me become licensed and progress my skydiving skills. Skydiving proved to be a great escape for those tough school weeks. The friends I have met there have turned out to be great people who always seem willing to do anything for anyone. I cannot wait to continue sky jumping this summer in the warmer weather, away from this cold. Blue Skies and clean fall out.

I am the one with the yellow and blue rig with the light blue jumpsuit.

1.0 Executive Summary

The M Resort is a multi-use casino, spa, and luxury suite space located in Henderson, Nevada just south of the Las Vegas strip. Through the analyses done in the previous technical reports, it was discovered that the building's mechanical system could be improved. The building utilizes a large amount of energy to maintain all the services offered throughout the year.

The Purpose of this mechanical redesign is to further analyze the central utility plant and the energy it consumes. The Las Vegas area attains a large amount of its power from the Hoover Dam, and with such a large demand for this power, the consumption and demand charges are some of the highest in the nation. Through an examination of all general utilities, it was discovered that the natural gas is at a marginally lower price than electricity.

Due to the difference in natural gas and electricity prices, a system of cogeneration was selected for the redesign. This system uses natural gas to generate electricity and the waste heat from this process can then be employed for building uses. This will create a large amount of waste heat and absorption chilling would be a great alternative to use this heat and replace the centrifugal chillers.

Though this report it was discovered that generating the full electricity usage of the building generates more heat than can be efficiently used by the building. Therefore another option of generating to meet heating demands was simulated as well. These options were evaluated based on first cost and annual utility costs. The result was that generating to electrical consumption created very high gas costs, and generating to heating demand produced larger costs than the existing system. Both systems had larger fists costs as well. The emissions of the both redesigns were less than the existing system which could mean a cost effective system if carbon taxes and credits were significant in the future.

This redesign affects other systems in the central plant as well. The electrical system is most directly affected. With the differing chillers the electrical loads on the system change and thus the panel boards and wire sized needed to be augmented. Also the generating equipment creates additional acoustical loads on the central plant. Hence the walls and surrounding spaces were analyzed to verify the sounds emanating from the equipment would not affect guest areas.

SECTION 2.0 - PROJECT INFORMATION

The M Resort, opened in March 2009, is a multi use hotel, spa and casino located on the southern portion of the Las Vegas strip in Nevada. Designed and constructed by the prominent Marnell Corrao Associates, this resort spares no luxury or quality with a cost of over one billion dollars. The owners have decided to locate the complex on approximately 90 acres south of the main Las Vegas Strip on the intersection of Las Vegas Boulevard and St Rose Parkway as shown in Figure 2.1. They feel that this will be the prime location for their property based on what research has indicated for the future of Las Vegas as well as their plans to expand the resort with a large retail portion.

Figure 2.1 – Resort Location

The M Resort Spa Casino consists of two areas a low rise and a high rise. The low rise portion is approximately 500,000 square feet which includes a spa level, kitchens, ballrooms, meeting rooms, administration, and back of the house as well as a casino level that consists of kitchens, restaurants, and casino space. The high rise portion of the building includes approximately 440 guest suites and lofts along with a restaurant atop the guest tower.

Primary Project Team

Table 2.1 – Project Team

Project Team	
Role	Company
Owner	Marnell Corrao Associates
Architects	Marnell Architecture
Interior Design	Marnell Architecture
General Contractor	Marnell Corrao Associates
Construction Management	Marnell Corrao Associates
Civil Engineer	Kimley-Horn Associates
Structural Engineer	Culp and Tanner
Food service Design	Maddock Design
Landscape Architect	J. W. Zunino Associates
Mechanical and Plumbing Engineers	Southland Industries
Electrical Engineers	JBA Consulting Engineers
Lighting Design	CD+M
Life Safety/Fire Protection	JBA Consulting Engineers
Property Management	Marnell Corrao Associates

Architectural Overview

The M Resort creates an awe inspiring campus that provides breath taking views of the valley as well as the Las Vegas Strip. This building utilizes non-reflective floor to ceiling glass to take advantage of this view. An open air environment is coupled with a beautiful canyon water fall feature to set apart the M Resort from other Las Vegas resorts. The residential tower is wrapped in the above mentioned glass, being topped off with loft suites on the top floor, creating rooms with 270 degree views of the Las Vegas landscape.

Mechanical Overview

The low rise commercial section of the building is served by air handling units and make up air units. Some of the spaces such as the casino floors are served with 100 percent outdoor air while other areas have some recirculation. The high rise guest tower utilizes vertical stack fan coil units that are connected to integrated wall mullions which supply outdoor air. Each room has its own unit which uses constant volume exhaust in the bathrooms to draw in the outdoor air. The building's utilities are housed in a central utility plant (CUP). This allows services to be combined and thus become more efficient for the overall building. The CUP includes 3900 tons of cooling and 46,800 MBH of heating with large expansion spaces available. Heat exchangers are used to heat the heating hot water, kitchen hot water, and pool heaters.

Electrical Overview

Electrical service enters the building at 12470 volts and it goes into a substation where it passes through a redundant switch. It is then fed to the various electrical rooms where it is broken down into 277v, 120v, and other voltages based on equipment and power needs in the subsequent spaces. The lighting system uses a variety of fixtures based on the space usage, however fluorescent fixtures are typical. The system also includes two 2000kW/2500kVA emergency generators for life safety backup.

Structural Overview

The first floor of the low rise portion of the building uses cast in place concrete. However the second floor of the same part of the building incorporates structural steel with spray on fire proofing. The tower is cast in place concrete columns along with cast in place post tensioned concrete floor slabs measuring four inches thick. The columns are set upon large spread footings throughout the entire building.

Transportation Overview

The guest tower is served by nine elevators, three of which are service elevators, along with stairwells. The low rise section uses multiple elevators throughout each of the different space types as required.

Fire and Life Safety Overview

The M Resort uses a sprinkler system throughout the majority of each section of the building. Each exit stairwell is pressurized by a dedicated supply fan connected to emergency power. The low rise spa and casino levels have smoke control systems that use either the supply fan or exhaust fan of the air handling unit serving the various areas with smoke control based on pressurization and exhaust requirements. The guest tower corridors have supply and exhaust air grilles at each floor with fire smoke dampers to control airflow.

SECTION 3.0 - BUILDING MECHANICAL SUMMARY

Guest Tower Summary

The guest tower includes the guest suites and lofts on floors 2 through 14. These spaces will primarily be occupied during the evening and overnight hours and will require individual temperature control. Each of the guest rooms will be served by a vertical fan coil unit located along the exterior wall of the suite. Outside air will be brought in through integrated wall mullions by means of a negatively pressurized space via the bathroom exhaust system. Common non-ducted gypsum board exhaust shafts will be used for adjacent suites with an exhaust fan mounted on the roof. Chilled water will be supplied from the central plant and heating will be achieved through an electric resistance coil in the fan coil unit.

These systems must not only deliver a comfortable climate for the guests, but also they must remain reasonably quiet as to not disturb the guests. The units are required to meet a minimum of an NC 30 rating. They have also been designed and installed to facilitate maintenance and repair although it is important that the systems are reliable enough that they do not require constant attention.

The guest corridors are conditioned by two constant volume makeup air units mounted on the roof of the tower. The maid and service spaces will be served by fan coil units similar to those used in the guest rooms.

Low Rise Summary

The low rise portion of the M Resort contains a variety of spaces which include restaurants, spa areas, food preparation, administration, ballrooms, casino space, lobbies and other pertinent areas that have variable occupancies and requirements. The offices and meeting rooms could be completely filled or empty on a given business day while the ballrooms and restaurants would tend to be occupied mainly during the evening hours. The widely variable occupancies of each of the spaces thus lent itself to a variable air volume (VAV) system fed by air handling units on the roof. Smoke control through the spaces is another key priority. Smoking is permitted in the casino spaces and thus the air system must not re-circulate the particulate in the air. One hundred percent outdoor air is a requirement for the casino and gambling spaces to purge the contaminated out of the air. Smoke control has also been integrated into the control of each air handler in the event of a fire.

Service Space Summary

The central chilled water plant is designed to serve the guest suite fan coil units, restaurants and low rise spa and casino level air handling units as well as support space fan coil units. It is sized to handle an initial 3900 tons of cooling with a future capacity of 7500 tons.

The central hot water plant is designed to serve the guest tower restaurant, low rise spa and casino level air handling units and support service space fan coil units. In addition it will serve the domestic hot water heat exchangers, kitchen hot water heat exchangers and the pool heat exchanger. This plant is capable of expanding from 46,800 MBH to a future capacity of 70,200 MBH.

The central steam plant will serve the main kitchen steam equipment on the spa level. The boilers will have a combined capacity of 6,400 lb/hr of high pressure steam. Each of the boilers is sized for sixty percent of the total load.

SECTION 4.0 - EXISTING MECHANICAL CONDITIONS

The heart of the mechanical system for the M Resort is found in the central utility plant (CUP) located at the northeast corner of the building site, which includes the chillers, boilers, heat exchangers, all necessary pumps and support equipment, as well as the cooling towers on the roof of the CUP. The air handling units and make up air units are located on either the low rise roof or the roof of the tower. Each guest room has a modular high rise fan coil unit with the support spaces also using fan coil units.

The chiller plant consists of three water cooled centrifugal chillers that distribute water to the air handling units and fan coil units. For clarity the condensing and evaporative water loops have been separated.

Condenser Water Loop

Water flows through the condenser where heat is rejected into the water; the water enters at 85 F and leaves at 97 F, shown in Figure 4.1. The water then flows to the cooling towers located on the roof where an induced draft cross flow of air, evaporatively cools the water. The water collects in the basin, which is kept level through an equalizer, and is then pumped back through the condensers. This basin also is the point in which make up water can be added. A heat exchanger has been provided to allow free cooling of the chilled water directly from the condenser water piping.

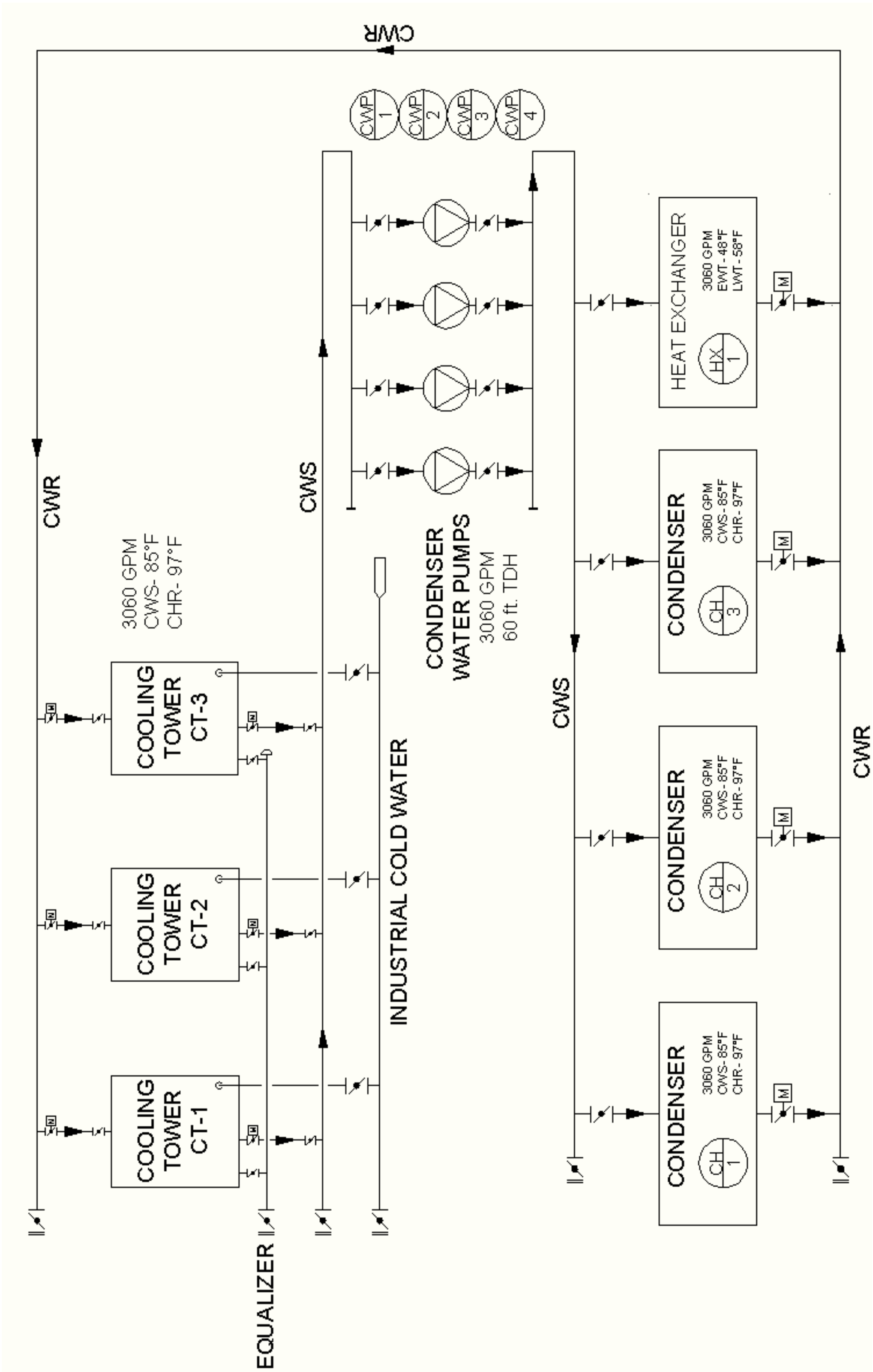


Figure 4.1 – Condenser Water Loop

Evaporator Loop

Refrigerant 123 transfers the heat-load from the evaporator and rejects it to the condensing water loop, shown in Figure 4.2. Water enters the evaporator at 58 F and leaves at 42 F. From the evaporator the water is pumped to the various loads throughout the building. Upon returning from the building loads, the water enters an air separator where makeup water can be added and is then returned to the evaporator. A heat exchanger is shown in this schematic as is shown in the condenser water loop. These heat exchangers are the same piece of equipment and allow for the free cooling of the chilled water from the condensing water piping.

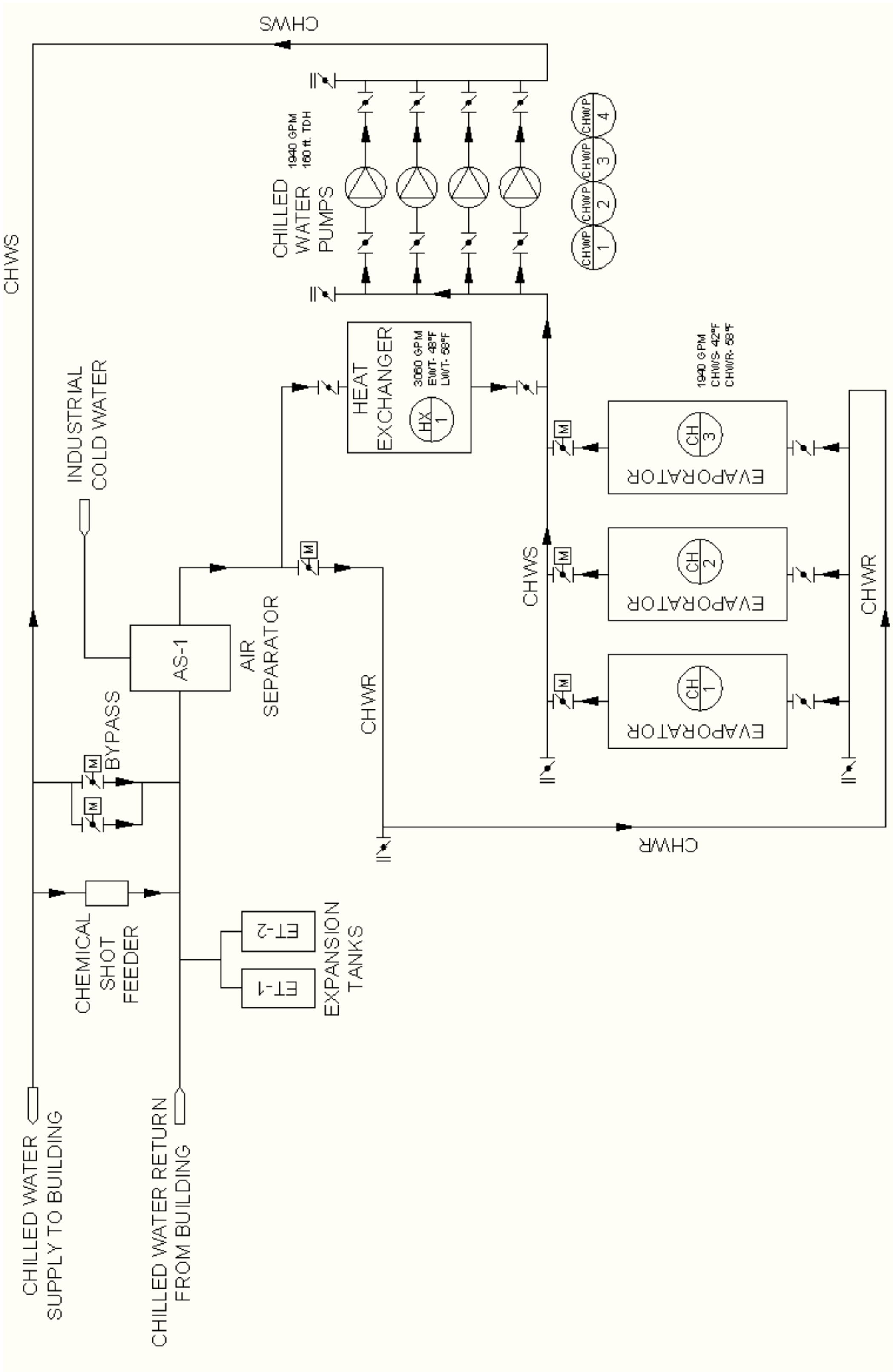


Figure 4.2 – Evaporator Loop

Heating Hot Water Loop

Heating hot water returns to the central plant and is pumped through the hot water boilers by the primary hot water pumps, shown in Figure 4.3. The hot water then travels through an air separator and then moves to the secondary hot water pumps where it is pumped to the system for hot water heating. These pumps also direct the hot water to three heat exchangers for domestic hot water, kitchen hot water, and the pool heater. Also shown are the expansion tanks and chemical shot feeder to treat the water and reduce scale and biological growth.

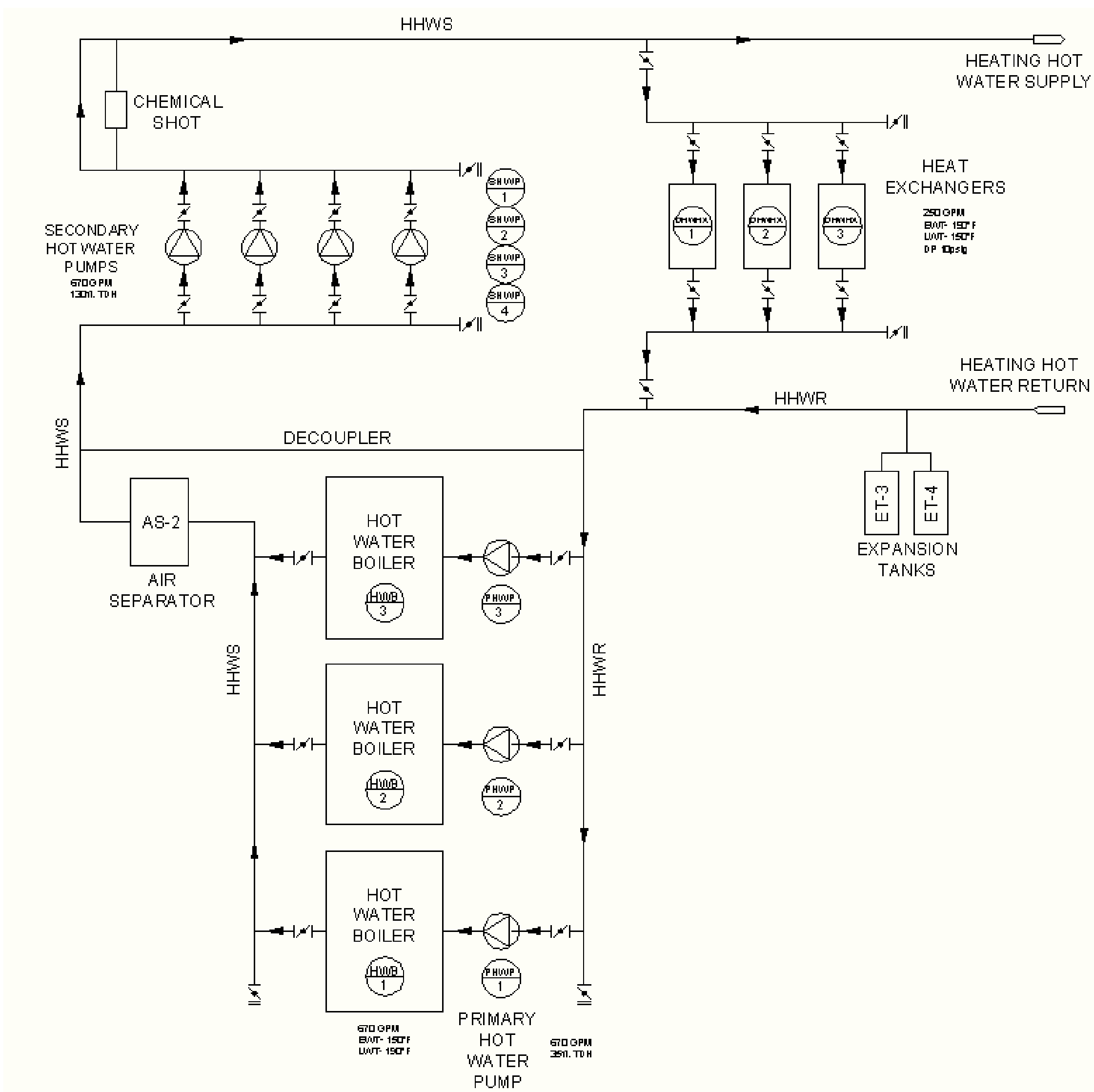


Figure 4.3 – Heating Hot water Loop

Steam System Loop

Steam at 80 psig exits the steam boilers where it goes to the required equipment, shown in Figure 4.4. It also braches off to the feed water tank for pre heat. Condensate is pumped from the kitchen equipment while gravity condensate returns from the buffet. The return enters the boiler feed water tank where it is mixed with make up water and preheated. Boiler feed water pumps then directs the water to the steam boilers where it is heated up and leaves the boilers again. Also shown is a blow down separator to remove any particulate and scale that may accumulate in the boilers.

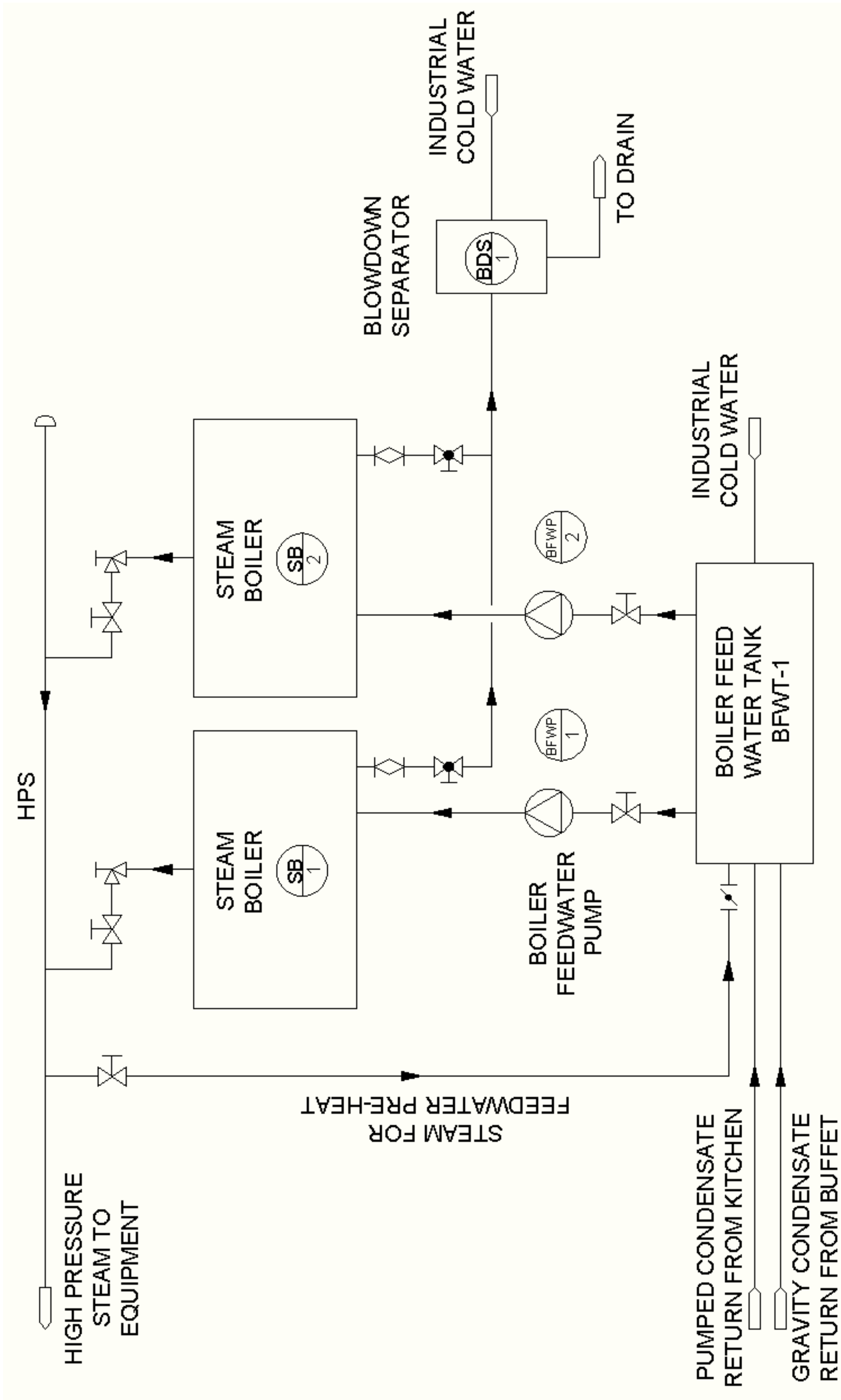


Figure 4.4 – Steam Water Loop

Air Handling Summary

The low rise and parts of the tower are conditioned from air handling units located on the low rise roof as well as the tower roof. Table 4.1 describes the types of spaces being served as well as the scheduled CFM and scheduled outdoor air volumes.

Table 4.1 – AHU Scheduled Data

AHU Scheduled Data			
Equipment	Area Served	Scheduled Supply CFM	Scheduled Min OA CFM
AHU B-1	Wine Bar/ Kitchen	11,000	11,000
AHU 1-1	Chiller Room	23,000	4,300
AHU 1-2	Spa/Fitness Center	21,000	7,050
AHU 1-3	Promenade	38,000	18,525
AHU 1-4	Meeting Rooms	29,000	19,735
AHU 1-5	Back of House Offices	15,000	4,510
AHU 1-6	Main Kitchen Bakery EDR Kitchen	37,000	22,300
AHU 1-7	Promenade	19,000	3,560
AHU 1-8	Ballroom	63,000	31,850
AHU 1-9	Warehouse Facilities Offices	11,000	725
AHU 1-10	Back of House Offices	24,000	8,730
AHU 1-11	Promenade	15,000	3,225
AHU 2-1	Steak/Seafood Restaurant and Kitchen	25,500	11,540
AHU 2-3	Italian Restaurant and Kitchen	15,500	8,170
AHU 2-4	Entertainment and High Limit Salon	27,000	17,660
AHU 2-5	Casino Floor SW	50,000	50,000
AHU 2-6	Casino Floor SE	55,000	55,000
AHU 2-7	Casino Floor NW	62,000	62,000
AHU 2-8	Casino Floor N	62,000	62,000
AHU 2-9	Café and Grille	13,000	7,445
AHU 2-10	Casino Floor NE	62,000	62,000
AHU 2-11	Buffet	38,000	16,690
AHU 2-12	Food Court	16,000	6,370
AHU 2-13	Sports Book and Poker Room	29,500	29,500
AHU 3-1	Executive and Corporate Offices Registration and Lobby	42,000	14,710
AHU 3-2	Public Circulation, Lobby and Patisserie	36,000	13,700
AHU 3-3	Offices, Money Cage, and Room Service	24,000	7,300
AHU T1	Tower Restaurant/Kitchen	16,000	8,600

SECTION 5.0 - MECHANICAL DEPTH

Purpose

The purpose of the Mechanical Depth is to perform a central utility plant optimization study. As described in Section 4.0, the current plant utilizes electric chillers and gas fired boilers. Thus the central plant uses a great deal of electricity throughout the year. Nevada's electricity rates are some of the highest in the nation for consumption and the demand charges are also high, especially in the peak summer months.

Methods

The energy analysis conducted in Technical Report Two shows that the amount of energy required to run both the Casino/Spa area and the Guest suite is significant especially in the summer months when the demand charge is at its highest. All of this power is taken from the grid which depending on the sources of the electricity can have very large emissions. In the building industry today there is a large push towards energy efficiency and environmental design, this redesign is intended to examine both of these principles.

A combined heating, cooling, and power system will be the focus of this redesign. The gas turbine engines being proposed, work based on a combustion process in which the exhaust gas flow drives a turbine in turn generating electricity. The waste heat can then be used for varying building processes.

The most obvious use for some of the waste heat would be for domestic water heating, kitchen hot water heating, process hot water heating and steam heating. This however will not require use of the full exhaust gas flow. With the large cooling loads, absorption cooling would be ideal for using this exhaust heat. In absorption cooling two different fluids, a refrigerant and an absorbent are used in a thermo chemical process that transfers heat from one place to another resulting in the cooling of some fluids.

Absorption chillers are of two main types, ammonia water and lithium bromide. Due to the availability of manufacturers and the local knowledge base a lithium bromide based absorption chilling system will be examined.

Justification of Mechanical Redesign

This new system will replace the existing chillers, and some other equipment located in the northwest corner of the low rise building. There are large areas of expansion in the CUP and the proximity to the current equipment makes it a prime choice for the new systems.

This redesign will focus on reducing the impact on the environment though emissions as well as reducing the life cycle and operational costs. Although it is uncertain whether these objectives will be met, the educational value of this redesign makes it a worthwhile endeavor, especially considering the direction of the building industry.

Mechanical System Redesign

This section will focus on the process and theories used to redesign the central plant to utilize a combined heat, cooling, and power system. Las Vegas is a community driven on money and luxury; therefore it is necessary to verify the financial benefits of any redesign made. The entire system will be examined in terms of a life cycle cost analysis comparing the current and redesign options.

The turbine engines being studied in the redesign run on natural gas; thus it is important that the fuel cost be justified in the initial phases. To do this the spark spread or spark gap will be used. This is difference in cost to generate heat and power using natural gas versus utilizing commercial electricity. Table 5.1 shows the various utility costs that were used in all calculations.

Table 5.1 – Utility Rates

Electric Utility Rates (Nevada Power Company) Rate Structure LGS-3				
Period	Time	Service Charge Per month	Consumption Charge Per kWh	Demand Charge Per kW
Summer On-Peak	1PM-7PM	$167.70 + $0.00627/kWh	$0.10034	$8.47
Summer Mid-Peak	10AM-1PM, 7PM-10PM		$0.08649	$0.63
Summer Off Peak	10PM-10AM		$0.06281	$0.50
All Other Periods	Winter (October-May)		$0.06281	$0.50
Natural Gas Utility Rates (Southwest Gas Corporation) Rate Structure SG-5L				
Period	Time	Service Charge Per month	Consumption Charge Per therm	Demand Charge Per therm
All Periods	All Times	$150.00	$1.1310	$0.00
Water Utility Rate (Las Vegas Valley Utility)				
Period	Time	Service Charge Per month	Cost per 1000 gallons	
All Periods	All Times	151.27	3.09	

As can be seen in the electric utility section of Table 5.1, Nevada Power has very high demand charges for peak usage. For spark gap calculations the consumption charge and service charge per kWh will be used for electricity rates which will result in a conservative answer, favoring electricity.

Table 5.2 – Spark Gap

Spark Gap						
Electricity			Natural Gas			Spark Gap
Usage	($/kWh)	$/10^6 btu	Usage	$/therm	$/10^6 btu	$
Peak	0.10661	31.25	Peak	1.131	11.31	19.94

Table 5.2 shows the cost per units of each fuel and then it calculated the cost for a comparable amount of each. The spark gap is then the difference in cost between the electricity and natural gas per 10^6 btu is $19.94. Generally a spark gap greater than 10 to 12 supports the use of a CHP system as is the case in this instance.

The gas turbines will subsequently produce a large amount of waste heat. In order to the system to remain theoretically cost effective the waste heat must be used in some manner. As stated in Section 4.0, the Resort will not utilize the full exhaust flow for domestic water heating; therefore absorption chillers will use the waste heat.

Absorption Chiller

To understand how the combined system will work it is necessary to determine how an absorption chiller functions. This type of chiller replaces the mechanical compression of a refrigerant as found in most electrical chillers. It accomplishes this through a thermo-chemical compression process. Basically, two different fluids are used, a refrigerant, water, and an absorber, lithium bromide salt.

Absorption chillers are made of four main components a generator, condenser, evaporator and an absorber. The single effect process initiates in the generator where a heat source is introduced, in this case the steam from the gas turbine engine. The steam from the engine will flow into a heat exchanger where the lithium bromide and water solution will begin to boil and separate. The water vapor then will be carried to the condenser where it is drawn out of the air by a heat exchanger. This removes the latent heat from the water and rejects it to a cooling tower. The liquid water is then transported to the evaporator where it will flow over an evaporator coil. This will remove heat from the from the chilled water loop. The chilled water will then flow to the air handlers and other processes and loads. Concurrent to this process, the lithium bromide salt collected in the generator is pre-cooled through another heat exchanger before making its way to the absorber. In the absorber the lithium bromide and water vapor combine over the absorber coil. The heat of dilution and condensation is again rejected to the cooling tower. The resulting solution is pumped back to the generator and the process begins again.

In a double effect chiller a second generator is added, one is a high temperature generator and another low temperature. The high temperature generator is driven off of the heat source while the low temperature generator attains its heat from the heat rejected from the process of absorbing refrigerant vapor into the liquid stream. This extra heat exchange is the driving force to generate more refrigerant vapor and subsequently allow for a more efficient chiller. Figure 5.1 illustrates a double effect absorption refrigeration cycle.

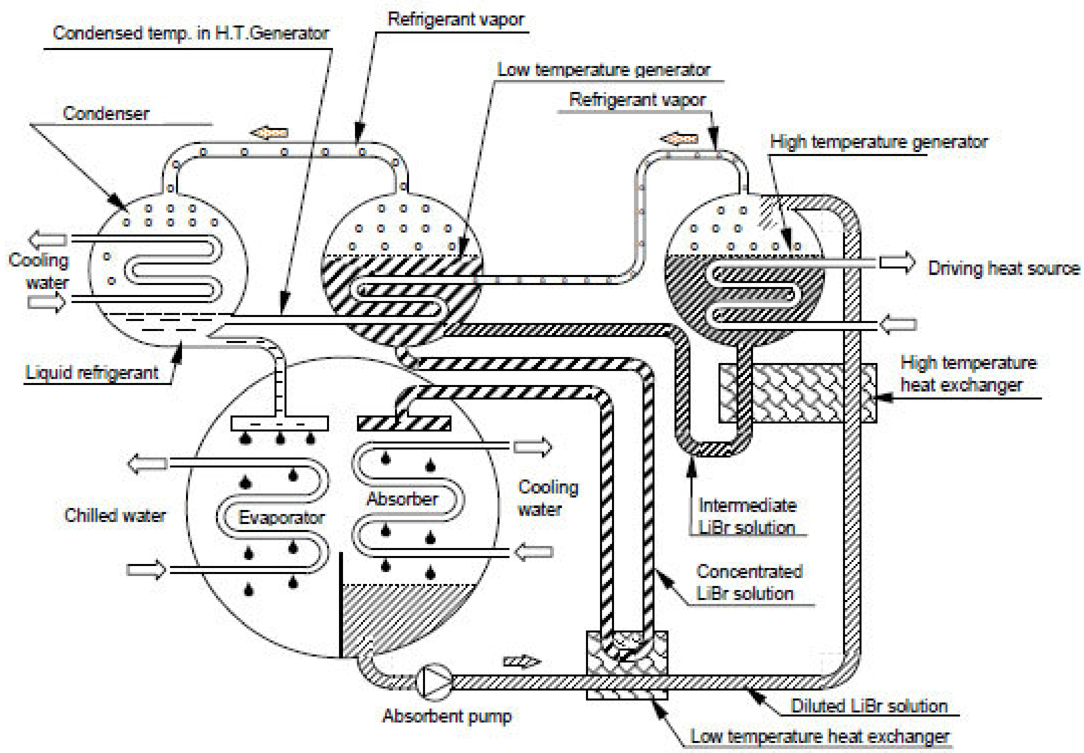


Figure 5.1 – Absorption Cooing Cycle

Sizing

In order to size the system it is important to first determine options for power generation. Nevada Power does not currently buy back power, however they do allow customers to feed the power to the grid and receive credit for the power. The option then exists that power can be over produced if necessary and it will not go to waste; however this is not the only choice, power could also be generated to the heating demand of the system. Power could also be produced to meet the electrical consumption in the building. In order to determine the scheme that the resort should use it is pertinent to look at the loads created.

Trane Trace was again used to model the building and central plant with the absorption refrigeration system as before with the electric chillers. It is imperative that the correct efficiencies and pump information be put in for the new systems. The boilers will remain the same for the simulation. Since the building loads are not going to change for the building the amount of cooling and heating the building requires will be the same, the change is only in the methods used to condition the building's environment. Therefore capacity must be available to meet the 3,900 tons of cooling and building heating loads.

Carrier markets absorption chillers made by Sanyo. They happen to make a double effect absorption chiller and it has capacities ranging from 98 to 1323 tons. This information was input into trace for the chiller plant. Absorption chillers are not used for energy efficiency; rather they are used in applications where there is a sufficient amount of waste heat to be used. These chillers have a low coefficient of performance, single effect around 0.5, double effect around 1.0, with regular centrifugal electric chillers having COP values around 4.0. This particular Carrier 16NK absorption chiller has a COP of 1.28. Therefore the result that could be attained from a simulation would result in a large additional boiler load from the heat source needed for the chiller and the electric usage would subsequently rise. Generating capacities must be added on order to compare the systems.

In order to design any type of mechanical system, the building's load profiles must be accurately predicted. With a combined heat and power system, this is imperative in order to determine the heating and especially the electrical loads needed to size equipment. Trane's Trace Energy Simulation Software along with Oak Ridge National Laboratory's BCHP Screening Tool and building data were used to estimate the building's electrical load profiles for the year.

Figure 5.2 shows the current electrical consumption of the resort with the electric chillers and grid energy, peak day hourly demand is about 7,300 kW. However, in order to determine the needs of the new system it is necessary to model the absorption system with the current parameters.

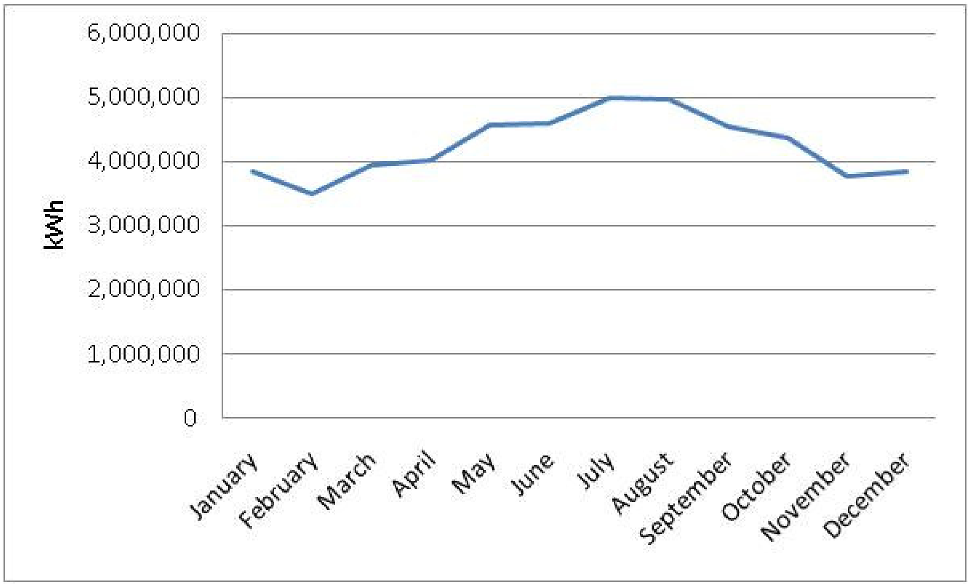


Figure 5.2 – Base model Electrical Consumption

As stated above, electricity can be produced for a variety of building load situations. Since Nevada power does not buy back power, it does not make sense to over produce. Therefore the system will be designed to meet the electrical consumption of the building. The electrical load profiles are essential to being able to design a system that accurately meets the building loads. This will help to determine the peak loads for a typical day and it will help determine how much a buildings electrical consumption varies through a day. Figure 5.3 shows the electrical demand for a typical day in February, it is important to note that this demand varies by 140 kW. Figure 5.4 show the same information for a typical July day, with a difference between the minimum and maximum being 97 kW. The importance of the difference of demand through a day helps to determine the amount of cycling that the equipment might have to do to meet the loads. To follow the load, the generating equipment will cycle on and off frequently, which will cause the efficiency to drop and the emissions will rise.

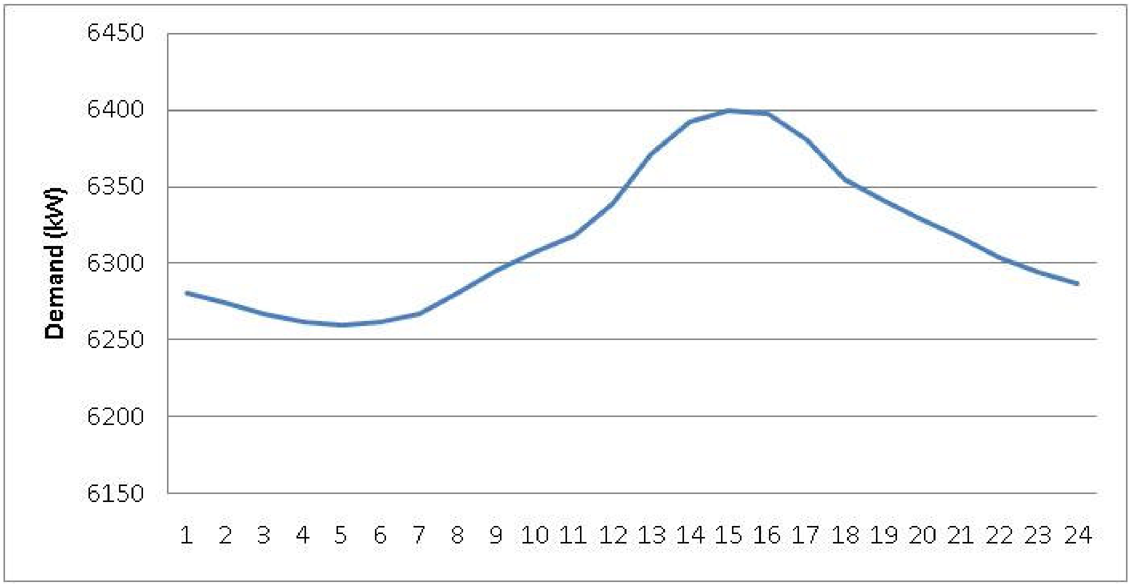


Figure 5.3 – Typical Day – February Electrical Demand

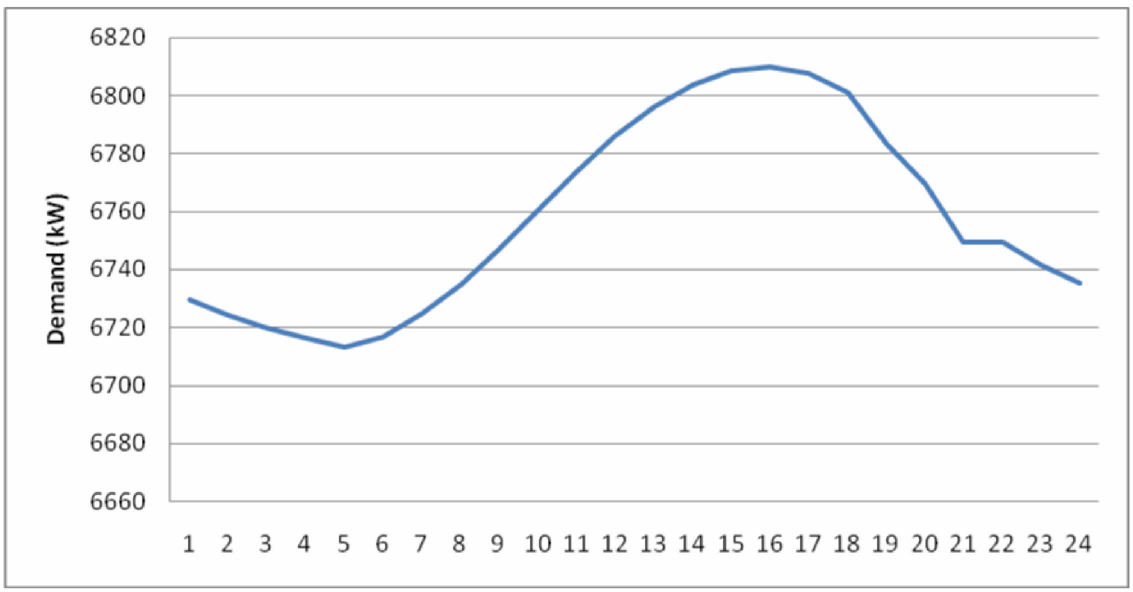


Figure 5.4 – Typical Day – July Electrical Demand

These figures help to determine the basic sizes of the generators needed to adequately meet the demand of the building. With the electricity rates higher in the summer as compared to the winter, it would be more advantageous to size to meet the demand in the summer which in this case is about 6810 kW. Upon examination of the Solar Gas Turbines, a generator this size does not exist. However with the loads varying it may be better to chose multiple generators to meet the loads. Mechanical equipment in general performs better when it is running at full load. Therefore the closer the generators are running to their designed capacities, the better the efficiency and emissions. Solar makes a 5700 kW, Taurus 60, model along with a 1200 kW, Saturn 20, model that will work well with the building. This will allow the 5700 kW generator to run most of the time at full load and when the load is exceeded the 1200 kW generator can fire.

Mechanical Redesign Results

Utilizing the modeling software previously mentioned, the loads and requirements of the redesign have been calculated and are first discussed, with conclusions following. The redesign will be examined by looking at the system sized for meeting the heating load as well as meeting the electrical load.

The first simulation will focus on meeting the electrical load and then using the waste heat for application processes such as the absorption cooling, general heating, hot water heating, and steam cooking needs. Figure 5.5 shows the existing electrical load on the building, with this amount of electricity needed the cogeneration electricity produced follows the same line on the graph. This results in nonexistent electricity consumption from the grid.

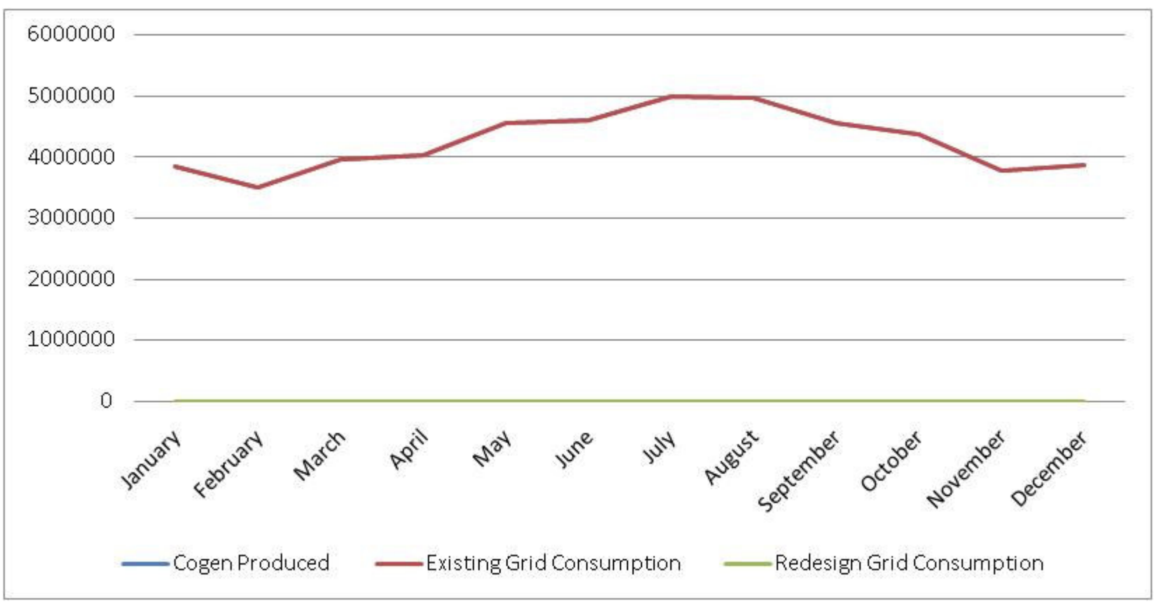


Figure 5.5 – Electrical consumption (kWh) Generating Full Electric

Figure 5.6 shows the heating requirements of the building as well as that produced from the electricity generating process. It is important to note that a large amount of waste heat is produced from the generating process as shown by the top most line on the graph. There is also a waste heat line in the figure. This represents tha amount of heat that is still avaliable after all of the building loads have been satisfied. During the winter especially there is a large amount of this wasted heat produced, while during the peak of the summer it is minimized.

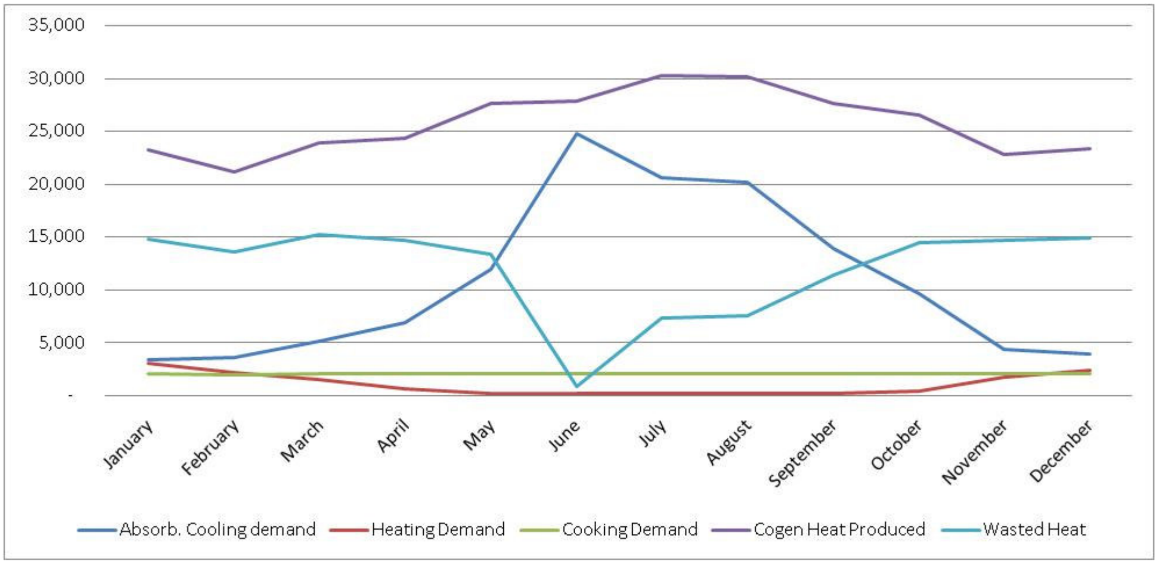


Figure 5.6 – Heating demands (MMBtu) Generating Full Electric

The absorption cooling process requires a certain amount of heat to create the desired cooling. The second simulation was executed to meet the heating requirement of this plant with other building heat needs included. Figure 5.7 includes the major sources of heat that are necessary for the building to run properly during a given year. These include the absorption cooling demand, the regular heating demand, and the heat that is necessary to produce the steam needed in the kitchens and restaurants. Also shown on the graph is the amount of heat produced by the Solar gas turbine generators.

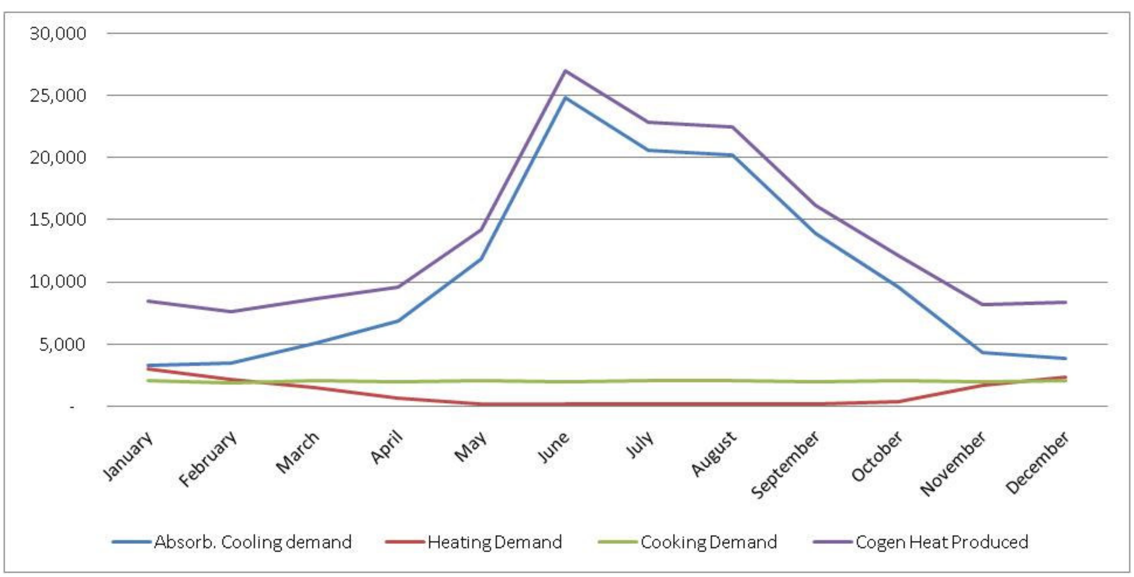


Figure 5.7 – Heating demand (MMBtu) Generating for heating demand

Figure 5.8 illustrates the electrical consumption of the building in the same simulation. This shows that the amount of heat needed for the building loads is not sufficient to generate the needed electricity. In the summer months the absorption chillers require more heat and thus the system can produce almost all of the electrical needs of the building, however it still falls just short. There is still a fairly large grid connected load for this simulation.

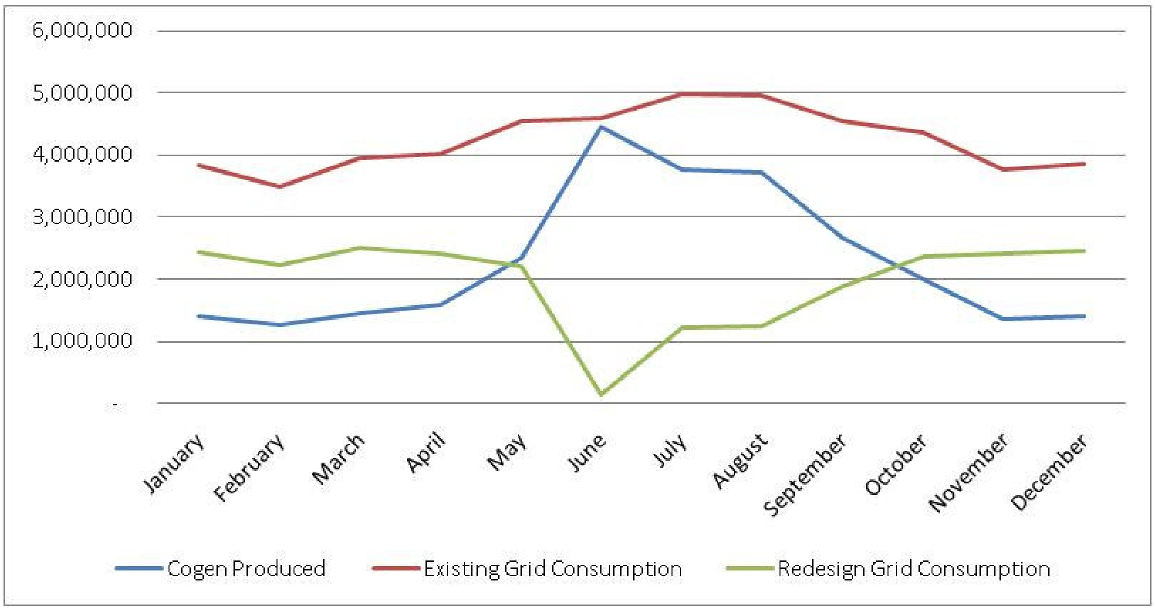


Figure 5.8 – Electrical Consumption (kWh) Generating for heating demand

In order to properly access the impact of each of the two simulations, it is important to determine the utility costs incurred. Figure 5.9 shows the three different systems in terms of natural gas cost per month. The existing system uses gas for building heating, hot water heating, and steam cooking. Figure 5.10 subsequently shows the electrical costs incurred through Southern Nevada Power per month. The gas consumption rises significantly in Figure 5.9 for the redesigned systems, however the electrical costs drops in response.

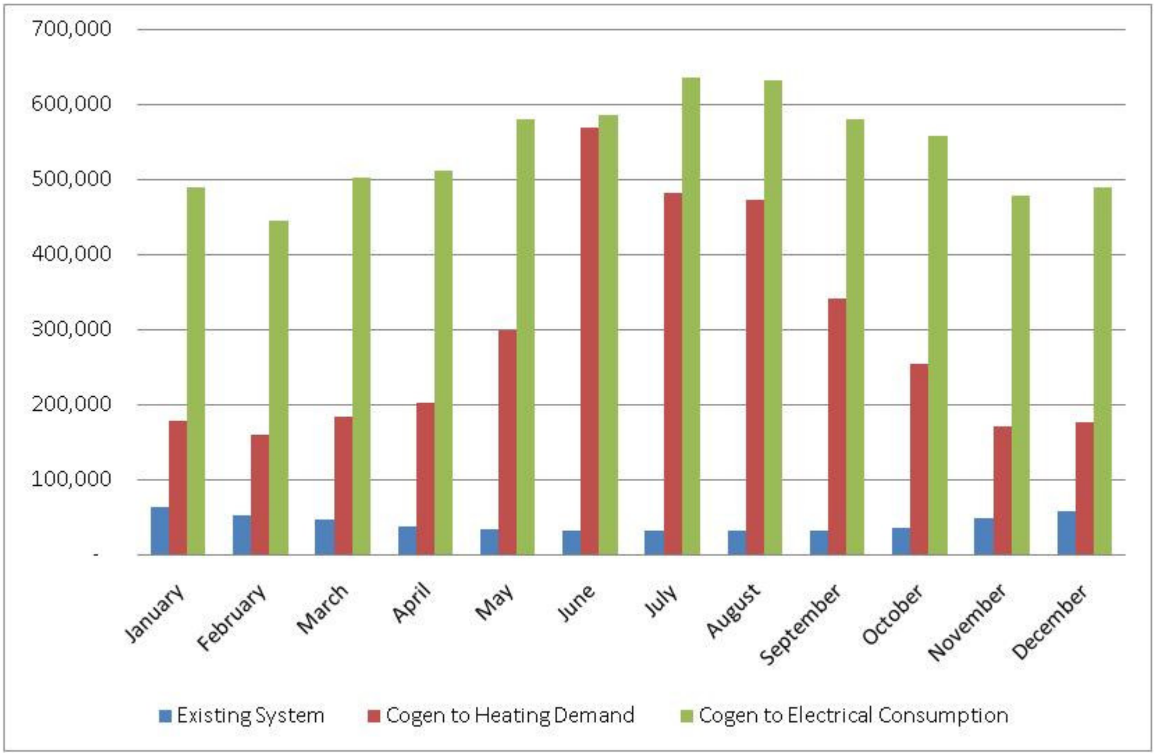


Figure 5.9 – Natural Gas Consumption ($)

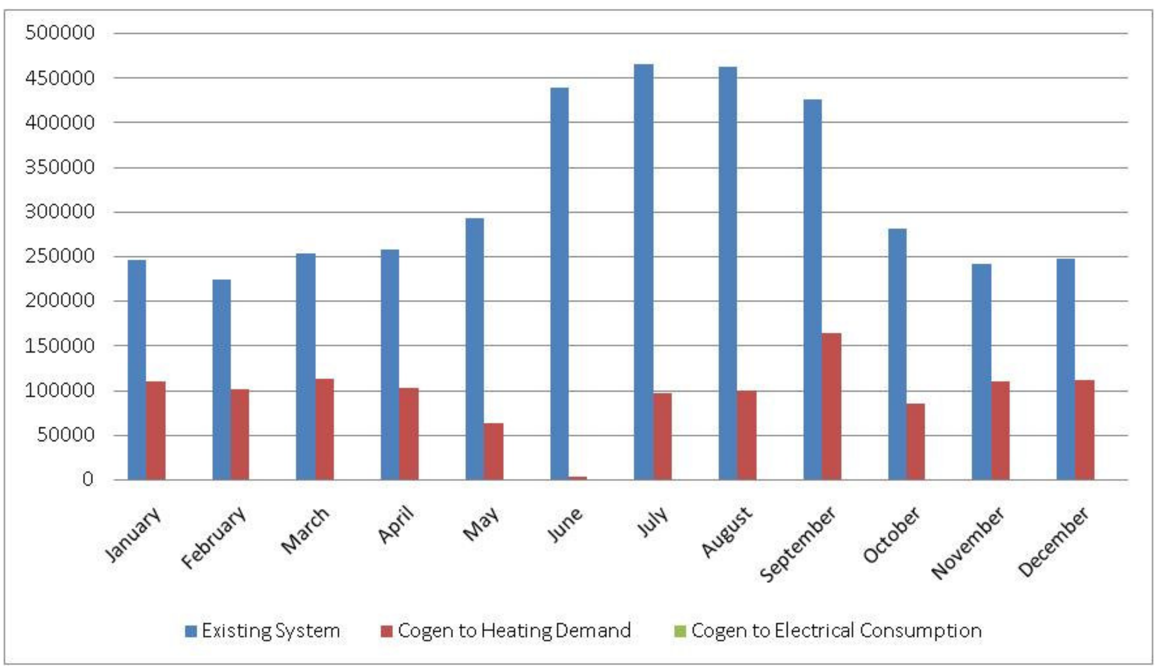


Figure 5.10 – Electrical Costs ($)

To properly compare the cost of each option in terms of utility cost a side by side comparison is needed. Figure 5.11 shows the three different systems and their respective yearly utility costs with the electrical and natural gas portions being of various colors to differentiate the portions of the costs. The existing system has electric and natural gas costs totaling $4,340,400, the cogeneration set to heating is $4,654,400, and the cogeneration set to electrical production is $6,496,300. Both of the redesign simulations resulted in higher yearly utility costs, however the life cycle cost must be used to determine if the savings exist in another part of the system life.

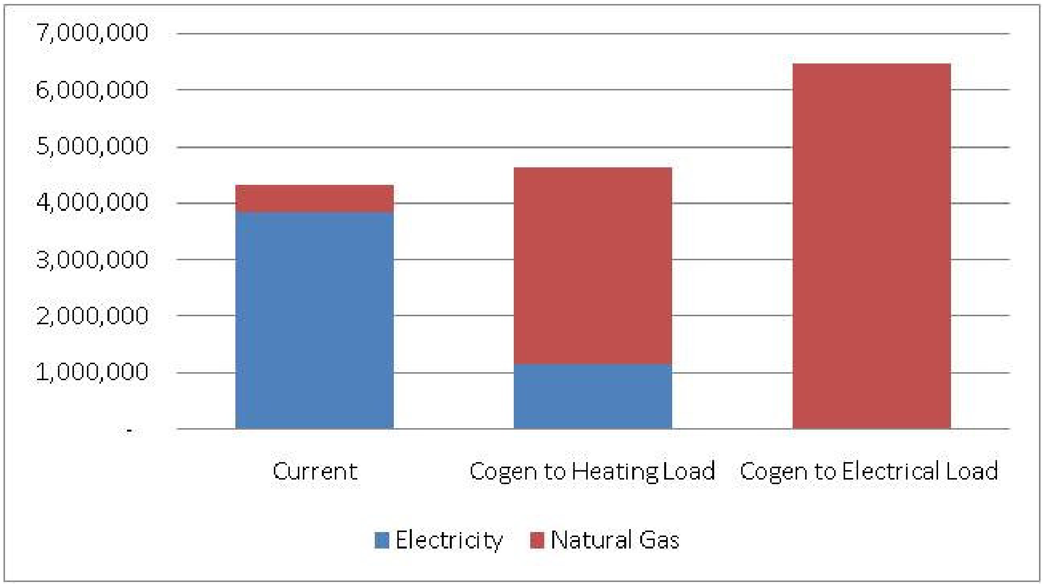
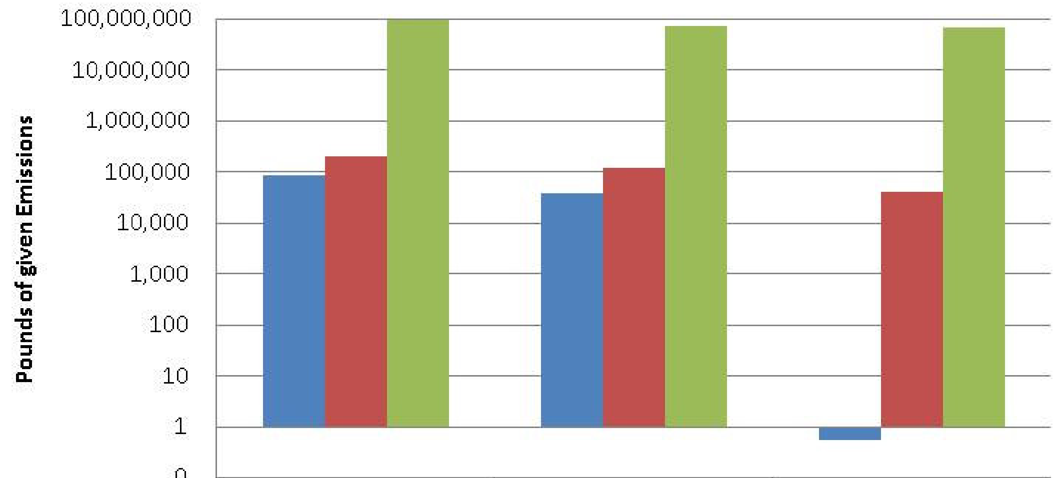


Figure 5.11 – Annual Utility Costs ($)

Environmental Analysis

Electricity from the grid comes from a variety of sources including coal, natural gas, wind, solar, hydro and many other forms. As a result The Energy Information Administration has developed average emission values for electricity in each state. Currently, southern Nevada attains a large amount of its power from the Hoover Dam, which does not have many emissions. This power is a direct factor of the water level in Lake Meade which is continually dropping and thus less and less power will be able to be generated in the future. Therefore for this analysis, emissions values from a more average state will be used with the assumption that in the future Nevada may have to use more emitting sources of power. New Mexico has emissions that are close to the average for the southwestern area and these values will be used for this analysis.

Figure 5.12 shows the emissions for the three different systems in terms of SO2, NOx, and CO2. From the given data it can be sent that the existing option has the most emissions with the subsequent systems having less and less emissions, since they are using natural gas which can generate electricity cleaner than what the grid can produce electricity.



	Existing System	Cogen to Heating Demand	Cogen to Electrical Consumption
SO2	86,601	40,033	0.55371
NOx	219,048	121,593	43,989
CO2	99,591,352	78,049,721	69,250,860

Figure 5.12 – Annual Emissions in Pounds (Log 10 Scale)

Life Cycle Cost Analysis

The Life Cycle cost is a great method to determining the value of the different redesign systems. This will take into account the firsts costs of each redesign as well as the utility costs for each. Table 5.3 breaks down this information for each system as well as various components of the system. The first costs are higher for the redesigned system as expected with the additional equipment needed. However the utility costs are higher for the redesigned systems as compared to the existing system. There will be no savings for the life cycle of the different systems because there are no utility savings from the redesign since this is the area which would make the system viable. The formula utilized to calculate the present value of the future utility costs is the uniform series present worth formula. The total savings after life cycle show values in parentheses which indicated that it is a negative amount.

PV= $A*(((1+I)^n – 1) / (i*(1+i)^n))$
n = number of years
i = Discount rate
A = annual utility bill
PV = present value of future utility bills

Table 5.3 – Life Cycle Cost Analysis

	Existing System	Redesign Cogen meeting Heating	Redesign Cogen meeting Electrical
Unchanged First Costs	$1,431,254	$1,431,254	$1,431,254
Central Plant First Costs	$1,624,000	$1,624,000	$1,624,000
Low Rise HVAC Cost	$20,850,428	$20,850,428	$20,850,428
High Rise HVAC Cost	$3,819,232	$3,819,232	$3,819,232
Chiller First Cost	$2,220,000	$2,008,500	$2,008,500
Generator First Cost	$0.00	$4,500,000	$4,500,000
Other Generator Costs	$0.00	$460,000	$460,000
TOTAL FIRST COSTS	**$29,944,914**	**$34,693,414**	**$34,693,414**
Natural Gas Cost	$499,029	$3,493,267	$6,496,278
Electrical Utility Cost	$3,841,339	$1,161,076	$0.00
TOTAL UTILITY COSTS	**$4,340,368**	**$4,654,344**	**$6,496,278**
Discount Rate	0.05	0.05	0.05
Life Cycle Length	20	20	20
Present Value of Utility Costs	$ 86,807,352	$ 93,086,875	$ 129,925,568
Total Savings After Life Cycle	-	$ (11,028,022)	$ (47,866,715)

Conclusions

For the M Resort it has been determined through the simulations and analysis that the existing system is the most cost effective both in terms of first cost as well as annual utility costs. Through the technical reports as well as the utility cost analysis completed in Section 5.0 of this report it was assumed that the cogeneration equipment along with the absorption chillers would be the most cost effective. The spark gap helped to support this finding as well. The data discussed in the above sections shows that the building does not have thermal and electrical loads that are in proportion to one another to support the use of a cogeneration system. The generating equipment produces a large amount of waste heat per kW. Therefore the building must have large heating demands to utilize the excess heat. In order to meet the electrical loads of the building a very large amount of heat is created with no outlet, therefore it does not make sense to generate to the electrical demands. Also, there is an extremely large increase in the utility bills with the natural gas turbine. The other option is to generate to the heating demand of the building and absorption chilling system. As discovered this also leads to an increase in utility costs. The generators selected do not efficiently generate electricity and waste heat in proportions to the needs of the M Resort. This type of system could work if a contract was created through the utility for a lower than market natural gas price that was fixed for a set number of years. For the breadth work the redesign to meet the heating loads will be used since it is the more reasonable of the two redesign options. Further conclusions can be found in section 8.0 of this report.

SECTION 6.0 - ELECTRICAL BREADTH

Purpose

The purpose of the electrical depth is to provide electrical service to the new mechanical equipment. In the mechanical depth it was calculated that the current system is the most cost effective, however with carbon credits and other environmental factors it may be economical to use the second alternative and generate electricity to meet the heating demand. The main equipment that changed is the chillers. This breadth will included and examination of the panel boards, resizing feeders and switchgear and breakers. Based on the manufacturer's data, the chillers have differing electrical needs and thus the system will need to be augmented.

Justification

In order for the equipment to run properly and safely, the electrical system must meet the specifications. Resizing one of the components and not the other could pose risks to the rest of the M Resort's systems.

Methods

The current electrical centrifugal chillers are supplied power directly from a transformer. They do not tie into the basic panel board system. According to the manufacturer's data, the new absorption chillers do not require such a large connected load. Therefore the new chillers can be placed on a panel board and subsequent switch gear that has enough room and meets the equipment's needs. Based on an analysis of the layout of the switch boards and their location in proximity to the chillers it was determined that switch board MSHAS1 would be the best alternative. This switch board currently has two spare sots as well.

Table 430.250 of the National Electrical Code 2005 was used to determine the full load amps of the new chillers. Table 310.16 and Table C.1 were then used to determine the wire size and associated conduit size. The breakers were sized based on a 250% of the full load current to allow for the start up current increases. The results can be seen in Table 6.1. All wires have been sized as copper THHN at 75°C.

Table 6.1 – Panel Board Equipment Sizing

Panel Board Equipment Sizing				
Equipment	Full Load Current	Wire Size	Conduit Size	Breaker Size
Absorption Chiller -1	65A	#4 AWG	1"	175A
Absorption Chiller -2	65A	#4 AWG	1"	175A
Absorption Chiller -3	65A	#4 AWG	1"	175A

The panel board is the next item to layout. Since the current system includes electrical connections for the future expansion as well as the immediate needs. Therefore in the design of the panel board for the chillers, space will be allotted for the future expansion. In a panel board lay out it is important to maintain as close to even loads on each of the three phases. Since this is a new board and the chillers are the only equipment on it, the loads are able to be kept even. After the board was completed, the panel circuit breaker could be sized based on the following equation. That amperage is then rounded up to the next standard breaker size.

$$\text{Breaker minimum} = 1.25*(\text{Highest Load}) + 1.0*(\text{Remaining Loads})$$

This calculation resulted in a 300A breaker. Table 310.16 and Table C.1 were again used to determine the feeder wire size and associated conduit size. For this panel board 500 kcm wire along with 3 inch conduit is needed. The new panel board, Figure 6.1, along with the original switchboard, Figure 6.2, and new switch board, Figure 6.3 can be seen below. Also included is an example of the existing chiller connection to the electrical system, Figure 6.4.

Voltage:	408/277V				300A						
Description	LOAD (VA)			Brk. Trip (A)	HPAS4		LOAD (VA)			Brk. Trip (A)	Description
	A	B	C		Ckt. #		A	B	C		
ABS Chiller-1	11265			175/3	1	2	11265			175/3	ABS Chiller-FUT
ABS Chiller-1		11265			3	4		11265			ABS Chiller-FUT
ABS Chiller-1			11265		5	6			11265		ABS Chiller-FUT
ABS Chiller-2	11265			175/3	7	8	11265			175/3	ABS Chiller-FUT
ABS Chiller-2		11265			9	10		11265			ABS Chiller-FUT
ABS Chiller-2			11265		11	12			11265		ABS Chiller-FUT
ABS Chiller-3	11265			175/3	13	14	11265			175/3	ABS Chiller-FUT
ABS Chiller-3		11265			15	16		11265			ABS Chiller-FUT
ABS Chiller-3			11265		17	18			11265		ABS Chiller-FUT
Spare				20	19	20				20	Spare
Spare				20	21	22				20	Spare
Spare				20	23	24				20	Spare
Spare				20	25	26				20	Spare
Spare				20	27	28				20	Spare
Spare				20	29	30				20	Spare
Spare				20	31	32				20	Spare
Spare				20	33	34				20	Spare
Spare				20	35	36				20	Spare
Spare				20	37	38				20	Spare
Spare				20	39	40				20	Spare
Spare				20	41	42				20	Spare
	33795	33795	33795				33795	33795	33795		

Total Load on Phase A:	67590	VA	Total Load			
Total Load on Phase B:	67590	VA	on Panel:	202.8	kVA Demand	
Total Load on Phase C:	67590	VA		254.0	A	Demand

Figure 6.1 – Panel Board HPSA4

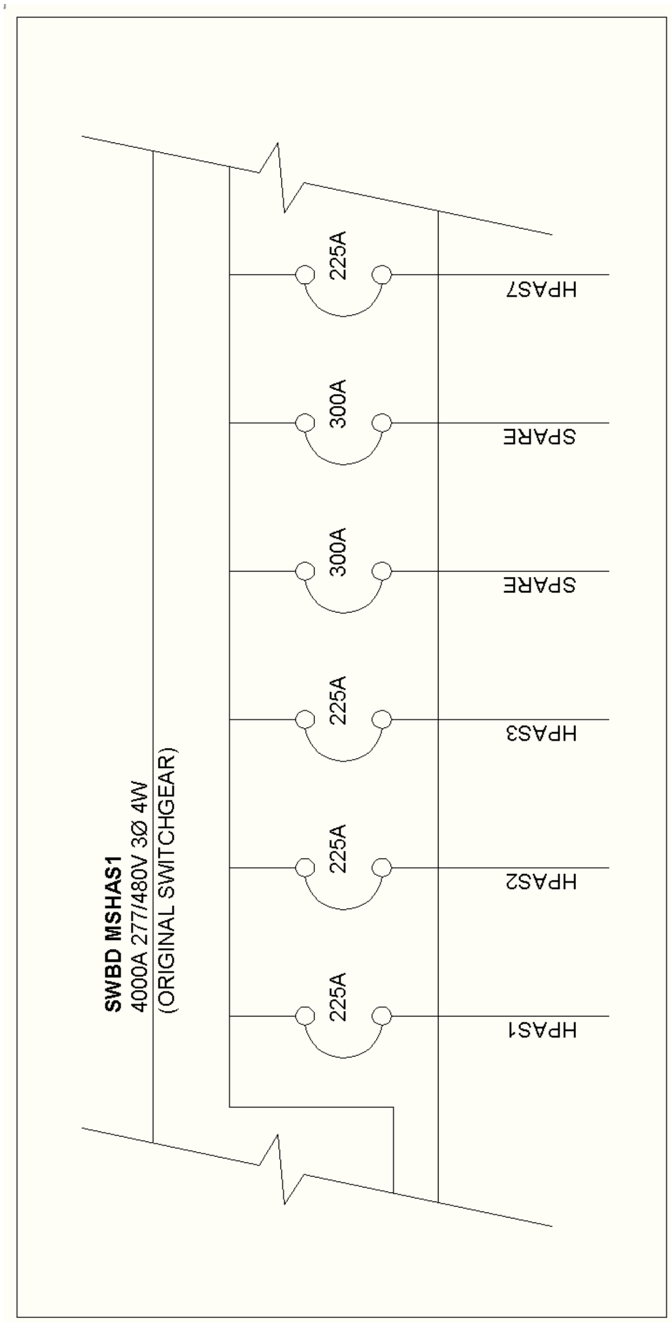


Figure 6.2 – Original Switchgear

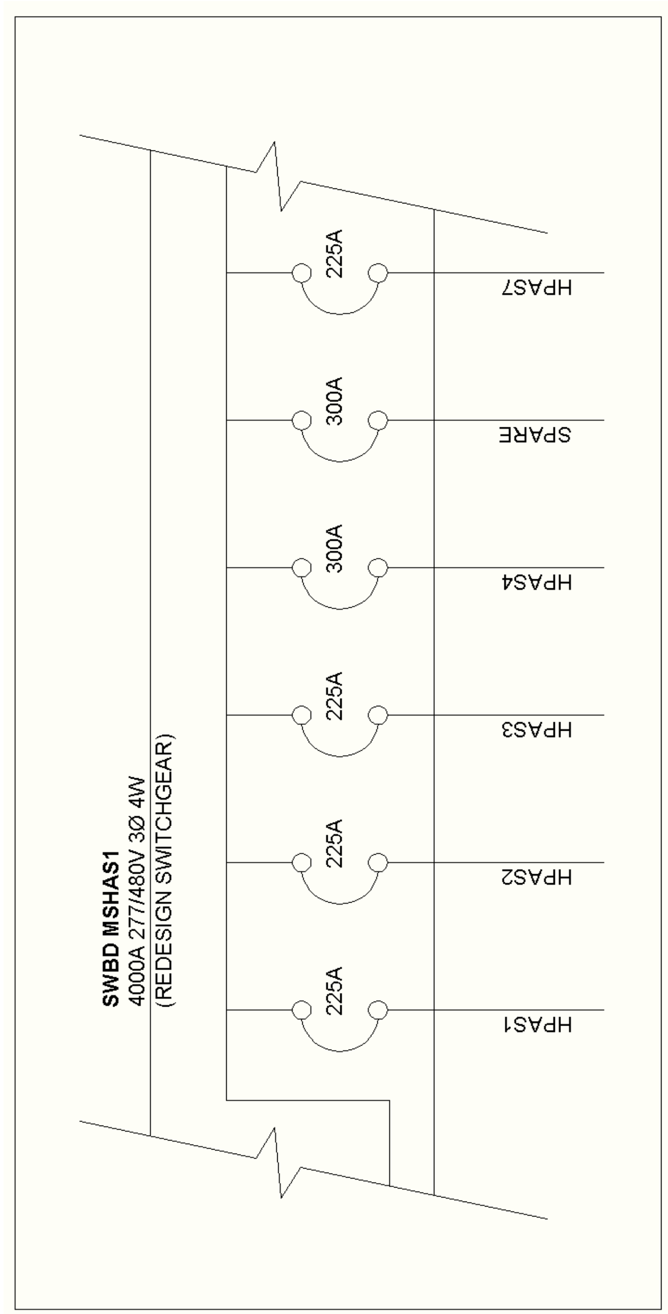


Figure 6.3 – Redesign Switchgear

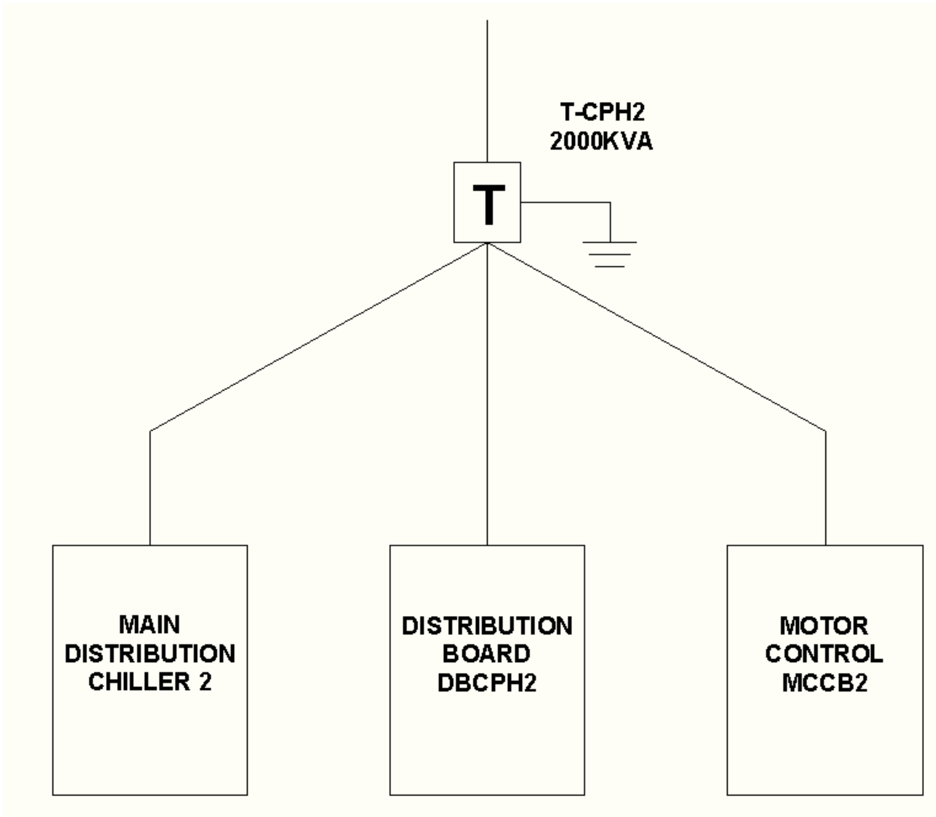


Figure 6.4 – Existing Chiller Connections

SECTION 7.0 - ACOUSTIC BREADTH

Purpose

The purpose of this acoustical breadth is to verify that the additional sound added to the mechanical equipment will not affect the surrounding M Resort spaces. Gas turbines for electrical generation inherently produce a large amount of noise as they run. The best option for a turbines location would be in a remote building without sound sensitive spaces in proximity. For this application, however, the mechanical, electrical and other main central services are located in the north east corner of the low rise section of the building. Currently this space includes room for future expansion as well as additional space for miscellaneous needs, which in this case includes the gas turbines. This area also has roof access which is necessary for the inlet combustion air as well as the exhaust air.

Methods

The diagram in Figure 7.1 shows where the cogeneration equipment will fit in the space, it is important to note that the drawing is not to scale. The actual drawings have not been included as per request of the owner. This sketch also illustrates the location of the new equipment to the surrounding spaces. Even though the central plant is located in the low rise section, there are no sound sensitive spaces directly neighboring the area. Therefore it is important to calculate the sound levels emanating from the equipment to the surrounding areas; however it is not as important as if it were next to a suite. Two of the sides of the new equipment room are parking areas, another being next to the current chillers, and the last being a corridor with access to other service areas.

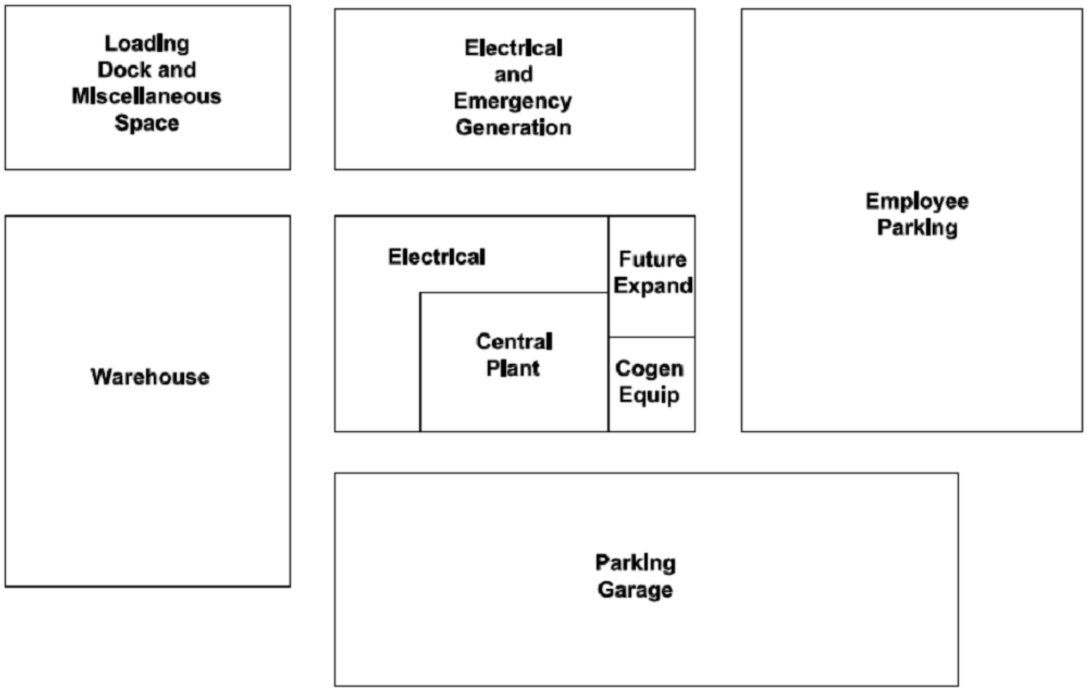


Figure 7.1 – Layout of Central Plant

Calculations

Sound levels have been taken from data given by the manufacturer of the Solar gas turbines. Tables D.1 and D.3 in Appendix D show the sound levels for un-silenced turbines without any type of sound enclosure. Tables D.2 and D.4 show the sound levels for silenced turbines including inlet and exhaust silencers. The sound levels have been calculated for a maximum radius of 50 feet; however the area most sensitive to these noise levels is located over 350 feet away from this location. It is also important to note that OSHA permits exposure in the range of 90dB for an eight hour work day. It is just as essential to protect workers in the area as it is the surrounding spaces.

The walls of the cogeneration room are made of 8" thick concrete masonry units which have a Sound transmission Class (STC) of 56db. Doors and other penetrations in the enclosure will also affect the sounds movement through the other spaces. Therefore three equations have been used to determine the sound levels moving into the other spaces.

Equation 1: $TL_{total} = 10*\text{Log}((\Sigma A)/\Sigma(A_i*(-0.1*TL_i))$

A_i = area of element

TL = Transmission Loss of element

Equation 2: $NR = TL + \text{Log}((\Sigma A)/S)$

A = total absorption in receiving room

S = Surface are of receiving room

Equation 3: $L_2 = L_1\text{-}NR$

L_1 = Noise level in source room

L_2 = Noise Level in receiving room

The Noise Reduction (NR) of the partition surrounding the cogeneration equipment was calculated to be 33dB. From Table 7.1 the A-weighted sound level is 81db which is L_1 in the equations. This results in $L_2 = 81\text{-}33 = 48$dB, which is more than an acceptable sound level for the spaces surrounding the equipment.

Table 7.1 – Taurus 60 and multiple turbines

Source	Octave Band Center Frequency, Hz									
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
Inlet Air	76	82	88	89	90	92	95	120	112	121
Inlet Air Silencer	0	-1	-2	-3	-15	-25	-48	-55	-37	
Net Inlet Air	76	81	86	86	75	67	47	65	75	
Exhaust Air	88	91	88	91	95	87	80	72	64	94
Exhaust Air Silencer	-3	-5	-11	-19	-22	-28	-26	-17	-14	
Net Exhaust Air	85	86	77	72	73	59	54	55	50	
Oil Cooler	63	70	67	60	55	52	48	44	39	58
Taurus 60	72	65	66	67	68	64	64	60	53	70
Multiple turbines	4	4	4	4	4	4	4	4	4	
Sum of Sources	86	87	87	86	78	69	65	67	75	
A-weighted correction	-39	-26	-16	-9	-3	0	1	1	-1	
A-weighted sound level	47	61	71	77	75	69	66	68	74	81

SECTION 8.0 – CONCLUSIONS AND FINAL RECOMMENDATIONS

Conclusions

This mechanical redesign explored the possibility of replacing the existing electric centrifugal chillers with a cogeneration system, producing electricity and using absorption chilling to cool the building with the waste heat. Originally this was thought to be a better alternative as discussed in the conclusion portion of Section 5.0. The high electricity prices as well as the spark gap analysis supported this theory.

However, when the efficiency of the cogeneration equipment is put into the equation, it becomes non cost effective. The turbines generate a large amount of waste heat per kW of electricity produced. In order to meet the electrical demands of the building the generators must be operating at full load, creating a large amount of heat. Figure 5.6 shows a large amount of excess heat is produced throughout most of the year, which is unusable by the building. Therefore the natural gas that is used to create some of the electricity goes to waste, since there is a larger portion of the energy burned going to the excess heat. Even meeting the heating demand be generating electricity does not result positively in terms of economics.

After examining the costs of the system a calculation was made to determine what a natural gas cost would have to be to be for the alternative to be effective. To meet the electrical consumption of the building and have a pay back period of 10 years the natural gas would have to be locked in at a cost of $0.673 per therm or less. The excess heat produced would still be wasted. Having a larger cooling demand that would offset the existing utility costs and create a demand for the extra heat could also result in an economical analysis. Another cost that could effect the decision in the future would be carbon taxes and emission requirements. Electricity from the grid generally has high emissions especially CO_2, using natural gas, as Figure 5.12 illustrates, has lower emissions in all of the categories shown. Depending on the cost per ton of carbon, this could make the system economically feasible.

Final Recommendations

It is clear from the data in this report that cogeneration is not a viable option for the M Resort. The designers of the mechanical system chose the best option for this application, which were the electric centrifugal chillers. These chillers have a longer history in the United States and there is a larger knowledge base in the contracting field. It is also important to realize that resorts in Las Vegas, Nevada do not have an extended life; most resorts have a shorter life cycle than most buildings at 20 to 30 years. Therefore any additional first cost would have to be paid back in this time preferably the first 5 to 10 years.

Currently the regulations on emissions are not strict enough to warrant a larger first cost. The M Resort was designed and built with first cost and reliability at the top of the list. Marnell Corrao Associates have built many of the top Las Vegas resorts and know and understand what works and what is truly cost effective for the life of the building. The saying, if it is not broke, don't fix it, fits this situation well, and is how the owners reacted.

The overall goal of this mechanical redesign was to improve the life cycle cost of the system in the M Resort. This means that the system with the lowest cost is the best option. Based on the data collected in this report and calculations conducted, the cogeneration turns out to be cost prohibitive. It is the final recommendation of this report that the current system be kept in place. It has a lower first cost and overall lower yearly utility costs. Environmentally the cogeneration system would be effective if the demand for heat was higher and if there was some incentive in terms of emissions.

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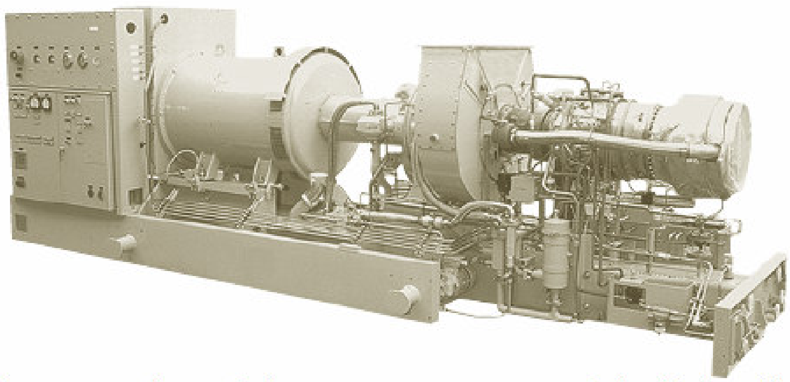
Chiller Information

Capacity	US Refrigeration Tons	Tons	1300
Chilled Water system	Flow Rate	m3/h	786
	Pressure Drop	mH20	9.6
		kPa	94
	Connection	inch	12
	Holding Water Volume	liter	1.61
Cooling Water System	Flow Rate	m3/h	1300
	Pressure Drop	mH20	14.5
		kPa	142
	Connection	inch	16
	Holding Water Volume	liter	4.11
Kind of Steam	Supply Pressure	kPa	784
	Steam Consumption	kg/h	5720
	Steam Connection	inch	6
	Drain Connection	inch	3
	Control Valve Connection	inch	3
Dimentions	Length	mm	7460
	Width	mm	3000
	Height	mm	3400
Tube Removal		mm	7000
Weights	Operation Weight	kgf	40500
	Max Shipping weight	kgf	27300
	Total Shipping Weight	kgf	27300
Shipping Method		1 Section	
Electric Power		V/PH/Hz	380/3/50
Total Electric Current		A	40.7
Apparent Power		kVA	33
Electrical Data	No. 1 ABS Pump	kW	7.5
		A	24
	No.2 ABS Pump	kW	3.7
		A	12
	REF Pump	kW	0.4
		A	1.8
	Purge Pump	kW	0.75
		A	1.9
	PD cell heater	W	76
	Control Circuit	W	300

Solar Turbines
A Caterpillar Company

SATURN 20
Gas Turbine Generator Set

POWER GENERATION



Package Arrangement

Gas Turbine
- *Saturn*® 20 Industrial, Single-Shaft
- Axial Compressor – 8 Stages
- Annular Combustion Chamber
  - 12 Fuel Injectors
- Coatings
  - Compressor: Inorganic Aluminum
  - Turbine and Nozzle Blades: Precious Metal Diffusion Aluminide
- Velocity Vibration Transducers

Main Reduction Drive
- Epicyclic
  - 1800 or 1500 rpm
  - Acceleration Vibration Transducers

Generator
- Salient Pole, 3 Phase, 6 Wire, Wye Connected, Synchronous with Brushless Exciter
- Open Drip-Proof Construction
- Sleeve Bearings
- Velocity Vibration Transducers
- Solid-State Voltage Regulation with Permanent Magnet Generator
- NEMA Class H Insulation with H Rise
- Continuous Duty Rating

Package
- Steel Base Frame with Drip Pans
- Direct-Drive AC or Pneumatic Start System
- Natural Gas Fuel System
- Control System
  - Microprocessor-Based PLC
  - Generator Control
  - Vibration and Temperature Monitoring
  - Auto Synchronizing
- Integrated Lube Oil System
  - Turbine-Driven Lube Pump
  - AC Pre/Post Lube Pump
  - Air/Oil Cooler
  - Integral Lube Oil Tank
  - Lube Oil Filter
- Documentation
  - Drawings
  - Quality Control Data Book
  - Inspection and Test Plan
  - Test Reports
  - O&M Manuals
- Factory Testing of Turbine and Package
- Weatherproof Acoustic Enclosure

Optional Equipment/Services
- Generator Options:
  - Standby Duty Rating
  - Standard Voltages: 380, 415, 3300 50 Hz; 240, 480 2400, 4160 60 Hz
- Fuel Systems
  - Liquid
  - Dual (Gas/Liquid)
  - Water Injection for NOx Control
  - Alternate Fuels (such as naphtha, propane, low Btu)
- Lube Oil System
  - Water/Oil Lube Cooler
  - Electrostatic Demister
  - Lube Oil Tank Heater
- Control System
  - Remote Display/Control Terminal
  - Heat Recovery Application Interface
  - Serial Link Supervisory Interface
  - KW Control
  - KVAR/Power Factor Control
- Accessory Equipment
  - Turbine Cleaning System: On-Crank and On-line
  - Package Lifting Kit
- Ancillary Equipment: Various Air Inlet and Exhaust Systems
  - Inlet and Exhaust Silencers
  - Self-Cleaning or Prefilter/Barrier Air Inlet Filter
  - Inlet Evaporative Cooler
  - Inlet Chiller Coils
  - Ancillary Support Frame

Solar Turbines
A Caterpillar Company

SATURN 20
Gas Turbine Generator Set

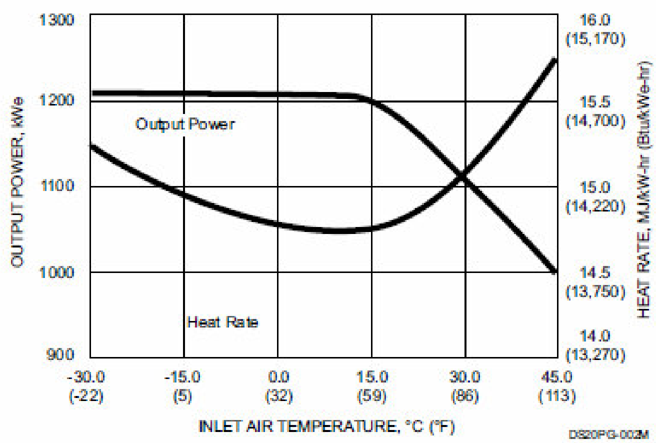
POWER GENERATION

Nominal Performance

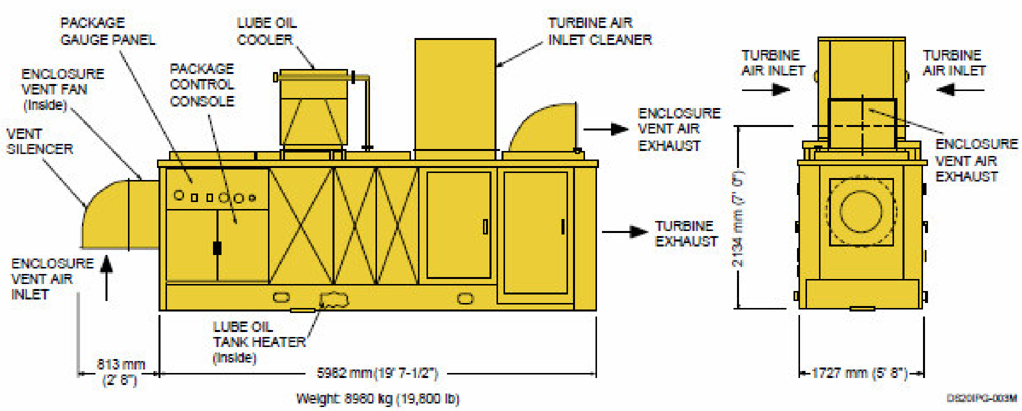
Output Power	1210 kWe
Heat Rate	14 795 kJ/kWe-hr (14,025 Btu/kWe-hr)
Exhaust Flow	23 540 kg/hr (51,890 lb/hr)
Exhaust Temp.	505°C (940°F)

Nominal Rating – per ISO
At 15°C (59°F), at sea level
No inlet/exhaust losses
Relative humidity 60%
Natural gas fuel with LHV = 31.5 to 43.3 MJ/nm³ (800 to 1100 Btu/scf)
AC-driven accessories
Engine efficiency: 24.3%

Available Power



Typical Package Configuration



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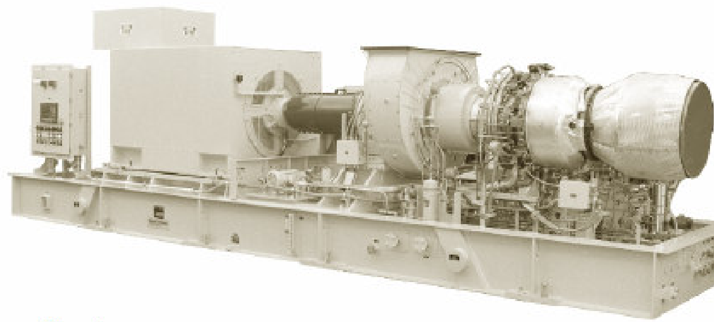
Solar Turbines
A Caterpillar Company

TAURUS 60
Gas Turbine Generator Set

POWER GENERATION



General Specifications

Gas Turbine
- *Taurus*™ 60
- Industrial, Single-Shaft
- Axial Compressor
  - 12-Stage
  - Compression Ratio: 12.5:1
- Combustion Chamber, Annular Type
  - Standard: Conventional Combustion
  - Available: Lean-Premixed, Dry, Low Emission (*SoLoNOx*™)
- Turbine
  - 3-Stage, Reaction

Main Reduction Drive
- Epicyclic
  - 1500 or 1800 rpm (50 or 60 Hz

Generator
- Salient Pole, 3 Phase, 6 Wire, Wye Connected, Synchronous with Brushless Exciter
- Permanent Magnet Generator
- Open Drip-Proof Construction
- Continuous Duty Rating
  - Standard: NEMA Class F Insulation with F Rise
  - Available: NEMA Class F Insulation with B Rise
- Standard Voltage Options:
  50 Hz: 3300, 6600, 11,000 V;
  60 Hz: 4160, 6900, 12,470, 13,200, 13,800 V

Key Package Features
- Steel Base Frame with Drip Pans
  - Standard: Open Unenclosed Package
  - Available: Weatherproof Acoustic Enclosure
- Direct-Drive AC Start System
- 120 VDC Battery/Charger System
- Standard Fuel System Options:
  - Natural Gas Fuel, Liquid Fuel, Dual Fuel (Gas/Liquid)
- Integrated Lube Oil System
  Standard Features:
  - Turbine-Driven Lube Pump
  - Pre/Post Lube Pump
  - Backup Post Lube Pump
  - Air/Oil Cooler
  - Integral Lube Oil Tank
  - Lube Oil Tank Heater
  - Simplex Lube Oil Filter
  - Oil Mist Eliminator
- Available Lube Oil System Options:
  - Water/Oil Cooler
  - Duplex Lube Oil Filters
- Factory Testing of Turbine and Package
- *Turbotronic*™ 4 On-skid Gas Turbine and Generator Control System
  System Features:
  - Combination Generator Control Module with Load Share, Auto Synchronization, Voltage Control
  - Standard Display with Discrete Event Log, Strip Chart, Historical Trend, Maintenance Screen
  - Vibration and Temperature Monitoring
  - English Display Text and Labels
- Available Control System Options:
  - Auxiliary and Remote Display/Control Terminals
  - Turbine Performance Map
  - Spanish, Portuguese, German, French, or Simplified Chinese Display Text and Labels
  - KW Control
  - KW Import Control
  - KVAR/Power Factor Control
  - ControlNet Redundant Media, Ethernet, Data Highway Plus, or Modbus RS232C/422/485 Supervisory Interface
  - Heat Recovery Application Interface
  - Unfired Waste Heat Recovery System Control
  - Multi-Unit Applications: Load Shed Control, Import/Export, or kW/KVAR Control Panels
  - Printer/Logger

Optional Pre-Engineered Solutions
- Multi-Unit Applications: Load Shed Control, Import/Export, or kW/KVAR Control Panels
- Operation with Alternate Fuels (such as naphtha, propane, low Btu gas)
- Liquid Fuel Monitoring, Boost Pump, and Coalescer Filter
- Ancillary Equipment
  - Inlet and Exhaust Silencers
  - Self-Cleaning or Prefilter/Barrier Air Inlet Filter
  - Inlet Evaporative Cooler or Chiller Coils
- Neutral Grounding Resistor or Transformer
- Switchgear and Generator Protective Relay
- Motor Control Center with Automatic Transfer Switch

Solar Turbines
A Caterpillar Company

TAURUS 60
Gas Turbine Generator Set

POWER GENERATION

Nominal Performance

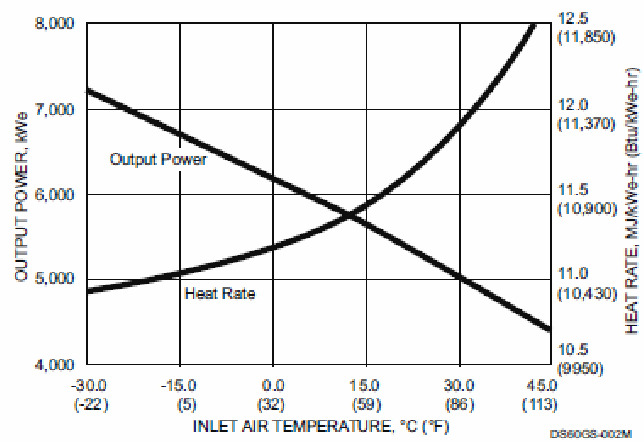
Output Power	5670 kWe
Heat Rate	11 425 kJ/kWe-hr (10,830 Btu/kWe-hr)
Exhaust Flow	78 385 kg/hr (172,810 lb/hr)
Exhaust Temp.	510°C (950°F)

Application Performance

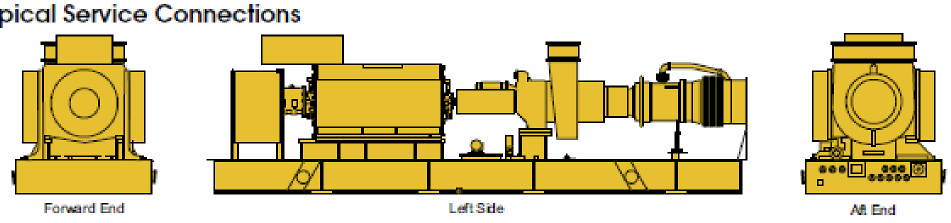
Steam (Unfired)	13.5 tonnes/hr (29,800 lb/hr)
Steam (Fired)	59.2 tonnes/hr (130,500 lb/hr)
Chilling (Absorption) 3.0 Refrigeration Tons	

Nominal rating – per ISO
At 15°C (59°F), sea level
No inlet/exhaust losses
Relative humidity 60%
Natural gas fuel with
LHV = 35 MJ/nm³ (940 Btu/scf)
No accessory losses
Standard and high ambient ratings available

Available Power



Typical Service Connections



Package Dimensions

Length:	9.8 m (32' 0")
Width:	2.7 m (8' 11")
Height:	3.0 m (9' 11")
Approx. Weight:	33 870 kg (74,650 lb)

Left Side
- Lube Oil Tank and Cooler Vents
- Lube Oil Cooler Connection
- Turbine Control Box
- Drains: Oil Tank, Oil Filter
- Turbine and Generator Drip Pan Drain

Right Side
- Generator Control Box
- Turbine and Generator Drip Pan Drain

Forward End
- AC Power, Lube Oil Tank Heater
- AC Power, Starter Motor
- Package Ground

Aft End
- DC Power, Backup Lube Pump
- Drains: Air Inlet Duct, Fuel, Combustor, Exhaust Diffuser
- Fuel Inlet
- Package Auxiliary Air Supply
- Turbine Cleaning
- AC Power: Pre/Post Lube Oil Pump, Liquid Fuel Pump (for Liquid Fuel Packages)
- Package Ground

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FOR MORE INFORMATION

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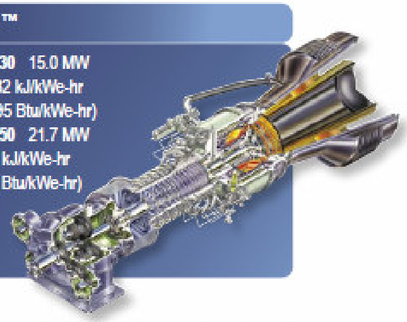
DNV

INDUSTRIAL GAS TURBINE PRODUCT LINE

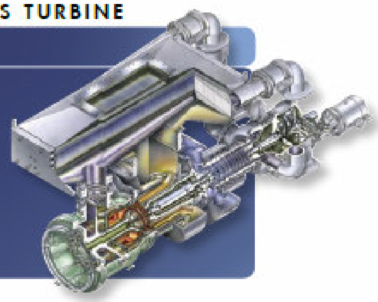
Saturn®	Centaur®	Taurus™	Mars®	Titan™
Saturn 20 1.2 MW 14 795 kJ/kWe-hr (14,025 Btu/kWe-hr)	*Centaur 40* 3.5 MW 12 910 kJ/kWe-hr (12,240 Btu/kWe-hr) *Centaur 50* 4.6 MW 12 270 kJ/kWe-hr (11,630 Btu/kWe-hr)	*Taurus 60* 5.7 MW 11 465 kJ/kWe-hr (10,860 Btu/kWe-hr) *Taurus 65* 6.3 MW 10 943 kJ/kWe-hr (10,373 Btu/kWe-hr) *Taurus 70* 7.5 MW 10 650 kJ/kWe-hr (10,100 Btu/kWe-hr)	*Mars 90* 9.5 MW 11 300 kJ/kWe-hr (10,710 Btu/kWe-hr) *Mars 100* 10.7 MW 11 090 kJ/kWe-hr (10,520 Btu/kWe-hr)	*Titan 130* 15.0 MW 10 232 kJ/kWe-hr (9695 Btu/kWe-hr) *Titan 250* 21.7 MW 9260 kJ/kWe-hr (8775 Btu/kWe-hr)



RECUPERATED GAS TURBINE

Mercury™

Mercury 50 4.6 MW
9351 kJ/kWe-hr
(8863 Btu/kWe-hr)



Mercury 50 Generator Set
Cogeneration and Distributed Energy
Austin, Texas

HEAT RECOVERY PERFORMANCE DATA

Specific Site Performance	Saturn 20	Centaur 40	Centaur 50	Mercury 50	Taurus 60	Taurus 65	Taurus 70	Mars 90	Mars 100	Titan 130	Titan 250
Exhaust Temperature, °C	511	446	513	377	516	555	490	468	491	500	465
Exhaust Temperature, (°F)	(952)	(835)	(956)	(710)	(960)	(1032)	(914)	(875)	(915)	(932)	(865)
Exhaust Mass Flow, thousand kg/hr	23.4	67.9	68.2	63.7	77.7	74.1	96.3	143.4	149.3	177.9	245.7
Exhaust Mass Flow, (thousand lbs/hr)	(51.5)	(149.6)	(150.3)	(140.4)	(171.3)	(163.4)	(212.3)	(316.2)	(329.1)	(392.2)	(541.6)
Turbine Fuel Input, GJ/hr	17.7	45.1	56.0	42.7	64.4	67.3	79.4	105.9	117.6	152.2	199.7
Turbine Fuel Input, (MMBtu/hr)	(16.8)	(42.7)	(53.1)	(40.5)	(61.0)	(63.8)	(75.3)	(100.4)	(111.5)	(144.3)	(189.3)
Process Steam Production (Unfired)											
Steam Output, tonnes/hr	4.0	8.9	11.5	6.3	13.5	14.6	15.6	21.2	23.5	29.3	35.8
Steam Output, (thousand lbs/hr)	(8.9)	(19.6)	(25.3)	(13.8)	(29.8)	(32.1)	(34.4)	(46.8)	(51.8)	(64.5)	(78.7)
Process Steam Production with Supplemental Firing, 871°C (1600°F)											
Steam Output, tonnes/hr	8.4	24.2	24.0	22.4	28.1	27.4	34.5	51.4	53.3	64.1	88.7
Steam Output, (thousand lbs/hr)	(18.5)	(53.3)	(53.0)	(49.4)	(62.0)	(60.5)	(76.1)	(113.3)	(117.4)	(141.4)	(195.2)
Additional Fuel to Burner, GJ/hr	10.4	35.7	30.5	38.7	34.6	30.2	45.9	71.7	70.9	82.8	124.8
Additional Fuel to Burner, (MMBtu/hr)	(9.9)	(33.8)	(28.9)	(36.7)	(32.8)	(28.6)	(43.5)	(68.0)	(67.2)	(78.5)	(118.3)
Process Steam Production with Supplemental Firing, 1538°C (2800°F)											
Steam Output, tonnes/hr	18.1	51.3	51.0	47.4	59.2	58.2	71.9	108.6	112.5	135.3	188.9
Steam Output, (thousand lbs/hr)	(40.0)	(113.2)	(112.4)	(104.4)	(130.5)	(128.3)	(158.6)	(239.5)	(248.1)	(298.2)	(415.9)
Additional Fuel to Burner, GJ/hr	33.2	100.4	95.9	100.0	109.2	105.4	138.5	209.2	214.2	254.1	371.9
Additional Fuel to Burner, (MMBtu/hr)	(31.5)	(95.2)	(90.9)	(94.8)	(103.5)	(99.9)	(131.3)	(198.3)	(203.0)	(240.8)	(352.6)

Emissions Data – Saturn 20

```
SOLAR TURBINES INCORPORATED                DATE RUN: 24-Mar-09
ENGINE PERFORMANCE CODE  REV. 3.41         RUN BY: Leslie Witherspoon
CUSTOMER: PSU Student Project
JOB ID: Saturn 20 with WI

          --- SUMMARY OF ENGINE EXHAUST ANALYSIS ---
                      POINT NUMBER 1

GENERAL INPUT SPECIFICATIONS

ENGINE FUEL:  SD NATURAL GAS
             27.83  in Hg        AMBIENT PRESSURE
              60.0  percent      RELATIVE HUMIDITY
            0.0069  ---          SP. HUMIDITY (LBM H2O/LBM DRY AIR)

FUEL GAS COMPOSITION (VOLUME PERCENT)
LHV (Btu/Scf) =   939.2   SG = 0.5970   W.I. @60F (Btu/Scf) = 1215.6

 Methane (CH4)               = 92.7899
 Ethane (C2H6)               =  4.1600
 Propane (C3H8)              =  0.8400
 N-Butane (C4H10)            =  0.1800
 N-Pentane (C5H12)           =  0.0400
 Hexane (C6H14)              =  0.0400
 Carbon Dioxide (CO2)        =  0.4400
 Hydrogen Sulfide (H2S)      =  0.0001
 Nitrogen (N2)               =  1.5100

GENERAL OUTPUT DATA

              627.  lbm/hr         INJECTED WATER FLOW
              813.  lbm/hr       FUEL FLOW
            20612.  Btu/lbm      LOWER HEATING VALUE
              939.  Btu/Scf      LOWER HEATING VALUE
            10826.  Scfm         EXHAUST FLOW @ 14.7 PSIA & 60F
            31613.  Acfm         ACTUAL EXHAUST FLOW CFm
            48403.  lbm/hr       EXHAUST GAS FLOW
            4466.4  deg R        ADIA STOICH FLAME TEMP, SDNG
             28.28  ---          MOLECULAR WEIGHT OF EXHAUST GAS
             57.91  ---          AIR/FUEL RATIO

EXHAUST GAS ANALYSIS

  ARGON        CO2       H2O        N2         O2

   0.88        2.89      8.58      73.54      14.10  VOLUME PERCENT WET
   0.96        3.17      0.00      80.44      15.43  VOLUME PERCENT DRY
   601.       2180.     2646.     35253.      7722.  lbm/hr
   0.74        2.68      3.25      43.35       9.50  G/(G FUEL)
```

Emissions Data – Taurus 60

```
SOLAR TURBINES INCORPORATED                    DATE RUN: 24-Mar-2009
ENGINE PERFORMANCE CODE  REV 3.41              RUN BY: Leslie Witherspoon
CUSTOMER: PSU Student Project
JOB ID: Taurus 60

          --- SUMMARY OF ENGINE EXHAUST ANALYSIS ---
                     POINT NUMBER 1

GENERAL INPUT SPECIFICATIONS

ENGINE FUEL:  SD NATURAL GAS
            27.82   in Hg         AMBIENT PRESSURE
             60.0   percent       RELATIVE HUMIDITY
           0.0068   ---           SP. HUMIDITY (LBM H2O/LBM DRY AIR)

FUEL GAS COMPOSITION (VOLUME PERCENT)
LHV (Btu/Scf) =   939.2   SG = 0.5970   W.I. @60F (Btu/Scf) = 1215.6

 Methane (CH4)                =  92.7899
 Ethane (C2H6)                =   4.1600
 Propane (C3H8)               =   0.8400
 N-Butane (C4H10)             =   0.1800
 N-Pentane (C5H12)            =   0.0400
 Hexane (C6H14)               =   0.0400
 Carbon Dioxide (CO2)         =   0.4400
 Hydrogen Sulfide (H2S)       =   0.0001
 Nitrogen (N2)                =   1.5100


GENERAL INPUT DATA

             2742   lbm/hr       FUEL FLOW
            20612   Btu/lbm      LOWER HEATING VALUE
              939   Btu/Scf      LOWER HEATING VALUE
            35291   Scfm         EXHAUST FLOW @ 14.7 PSIA & 60F
           103782   Acfm         ACTUAL EXHAUST FLOW CFm
           158953   lbm/hr       EXHAUST GAS FLOW
            28.49   ---          MOLECULAR WEIGHT OF EXHAUST GAS
            57.12   ---          AIR/FUEL RATIO

EXHAUST GAS ANALYSIS

  ARGON       CO2        H2O        N2         O2

   0.90       2.99       6.77      75.03      14.31  VOLUME PERCENT WET
   0.96       3.21       0.00      80.47      15.35  VOLUME PERCENT DRY
   1999       7349       6800     117256      25546  lbm/hr
   0.73       2.68       2.48      42.77       9.32  G/(G FUEL)
```

Table B.1 – Un-silenced Saturn 20

Source	Octave Band Center Frequency, Hz									
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
Inlet Air	72	74	76	80	84	91	93	97	108	108
Exhaust Air	94	94	93	92	90	89	85	81	70	93
Oil Cooler	63	70	67	60	55	52	48	44	39	58
Saturn 20	72	77	78	81	81	77	77	78	80	85
Sum of Sources	94	94	93	92	91	93	94	97	108	109
A-weighted correction	-39	-26	-16	-9	-3	0	1	1	-1	
A-weighted sound level	55	68	77	83	88	93	95	98	107	108

Table B.2 – Silenced Saturn 20

Source	Octave Band Center Frequency, Hz									
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
Inlet Air	72	74	76	80	84	91	93	97	108	108
Inlet Air Silencer	0	-1	-2	-3	-15	-25	-48	-55	-37	
Net Inlet Air	72	73	74	77	69	66	45	42	71	
Exhaust Air	94	94	93	92	90	89	85	81	70	93
Exhaust Air Silencer	-3	-5	-11	-19	-22	-28	-26	-17	-14	
Net Exhaust Air	91	89	82	73	68	61	59	64	56	
Oil Cooler	63	70	67	60	55	52	48	44	39	58
Saturn 20	70	69	72	68	66	62	61	75	62	76
Sum of Sources	91	89	83	79	73	68	63	75	72	94
A-weighted correction	-39	-26	-16	-9	-3	0	1	1	-1	
A-weighted sound level	52	63	67	70	70	68	64	76	71	79

Table B.3 – Un-silenced Taurus 60

Source	Octave Band Center Frequency, Hz									
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
Inlet Air	76	82	88	89	90	92	95	120	112	121
Exhaust Air	88	91	88	91	95	87	80	72	64	94
Oil Cooler	63	70	67	60	55	52	48	44	39	58
Taurus 60	81	81	84	86	86	81	79	78	79	88
Sum of Sources	89	92	92	94	97	93	95	120	112	
A-weighted correction	-39	-26	-16	-9	-3	0	1	1	-1	
A-weighted sound level	50	66	76	85	94	93	96	121	111	121

Table B.4 – Silence Taurus 60

Source	Octave Band Center Frequency, Hz									
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
Inlet Air	76	82	88	89	90	92	95	120	112	121
Inlet Air Silencer	0	-1	-2	-3	-15	-25	-48	-55	-37	
Net Inlet Air	76	81	86	86	75	67	47	65	75	
Exhaust Air	88	91	88	91	95	87	80	72	64	94
Exhaust Air Silencer	-3	-5	-11	-19	-22	-28	-26	-17	-14	
Net Exhaust Air	85	86	77	72	73	59	54	55	50	
Oil Cooler	63	70	67	60	55	52	48	44	39	58
Taurus 60	72	65	66	67	68	64	64	60	53	70
Sum of Sources	86	87	87	86	78	69	65	67	75	
A-weighted correction	-39	-26	-16	-9	-3	0	1	1	-1	
A-weighted sound level	47	61	71	77	75	69	66	68	74	81

Solar Turbine Manufacturer Acoustical Data

Table 5. Sound Pressure Levels – Unsilenced Combustion Air Inlet at 15 m (50 ft), Full Load

Model	Octave Band Center Frequency, Hz									
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
Saturn 10	70	72	74	78	82	89	91	95	106	106
Saturn 20	72	74	76	80	84	91	93	97	108	108
Centaur 40	75	81	87	88	89	91	94	117	109	118
Centaur 50	75	81	87	88	89	91	94	118	110	119
Mercury 50	74	80	86	87	88	90	93	116	108	117
Taurus 60	76	82	88	89	90	92	95	120	112	121
Taurus 70	79	85	91	92	93	95	98	126	118	127
Mars 90	81	87	93	94	95	97	100	125	117	126
Mars 100	81	87	93	94	95	97	100	129	121	130
Titan 130	82	88	94	95	96	98	101	131	123	132

Sound pressure levels (Re 20 µPa) for *SoLoNOx* and conventional gas turbine packages

Table 6. Sound Pressure Levels – Unsilenced Combustion Exhaust at 15 m (50 ft), Full Load

Model	Octave Band Center Frequency, Hz									
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
Saturn 10	93	93	92	91	89	88	84	80	69	92
Saturn 20	94	94	93	92	90	89	85	81	70	93
Centaur 40	84	92	88	90	89	88	81	72	66	91
Centaur 50	86	88	88	87	94	88	82	70	61	93
Mercury 50	63	75	71	69	75	65	54	41	32	73
Taurus 60	88	91	88	91	95	87	80	72	64	94
Taurus 70	91	94	91	95	97	93	87	80	67	98
Mars 90	91	95	93	96	100	95	87	77	67	100
Mars 100	91	95	93	96	100	95	87	77	67	100
Titan 130	92	96	94	97	101	96	88	78	68	101

Sound pressure levels (Re 20 µPa) for *SoLoNOx* and conventional gas turbine packages

Table 7. Sound Pressure Levels – Unenclosed Package at 1 m (3 ft), Full Load

Model	Octave Band Center Frequency, Hz								
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
Saturn 10	85	90	91	94	94	90	90	91	93	98
Saturn 20	86	91	92	95	95	91	91	92	94	99
Centaur 40	94	94	97	99	99	94	92	91	92	101
Centaur 50	94	94	97	99	99	94	92	91	92	101
Mercury 50	79	81	84	88	84	84	86	90	83	94
Taurus 60	94	94	97	99	99	94	92	91	92	101
Taurus 70	94	94	97	99	99	94	102	100	95	106
Mars 90	87	86	92	92	95	93	102	100	95	106
Mars 100	87	86	92	92	95	93	102	100	95	106
Titan 130	93	92	100	97	94	90	91	103	96	105

Package average sound pressure levels (Re 20 µPa) for *SoLoNOx* and conventional gas turbine packages

Table 8. Sound Pressure Levels – Unenclosed Package at 15 m (50 ft), Full Load

Model	Octave Band Center Frequency, Hz								
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
Saturn 10	71	76	77	80	80	76	76	77	79	84
Saturn 20	72	77	78	81	81	77	77	78	80	85
Centaur 40	81	81	84	86	86	81	79	78	79	88
Centaur 50	81	81	84	86	86	81	79	78	79	88
Mercury 50	67	69	72	76	72	72	74	78	71	82
Taurus 60	81	81	84	86	86	81	79	78	79	88
Taurus 70	82	82	85	87	87	82	90	88	83	94
Mars 90	75	74	80	80	83	81	90	88	83	94
Mars 100	75	74	80	80	83	81	90	88	83	94
Titan 130	82	81	89	86	83	79	80	92	85	94

Package average sound pressure levels (Re 20 µPa) for *SoLoNOx* and conventional gas turbine packages

Table 9. Sound Pressure Levels – Enclosed Package at 1 m (3 ft), Full Load

Model	Octave Band Center Frequency, Hz									
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
Saturn 10	84	83	86	82	80	76	75	89	76	90
Saturn 20	84	83	86	82	80	76	75	89	76	90
Centaur 40	84	86	86	88	83	78	74	76	71	85
Centaur 50	85	86	84	85	81	77	73	72	64	83
Mercury 50	86	87	84	83	78	76	70	74	68	82
Taurus 60	85	78	79	80	81	77	77	73	66	83
Taurus 70	96	89	86	83	81	80	78	75	70	85
Mars 90	96	89	86	83	81	80	78	75	70	85
Mars 100	96	89	86	83	81	80	78	75	70	85
Titan 130	96	89	86	83	81	80	78	75	70	85

Package average sound pressure levels (Re 20 µPa) for *SoLoNOx* and conventional gas turbine packages
For enclosed *Mars* and *Titan* gas turbine packages, acoustical lagging is required for the combustion air inlet ducting.

Table 10. Sound Pressure Levels – Enclosed Package at 15 m (50 ft), Full Load

Model	Octave Band Center Frequency, Hz									
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
Saturn 10	70	69	72	68	66	62	61	75	62	76
Saturn 20	70	69	72	68	66	62	61	75	62	76
Centaur 40	71	73	73	75	70	65	61	63	58	72
Centaur 50	72	73	71	72	68	64	60	59	51	70
Mercury 50	73	74	72	73	69	65	61	60	52	71
Taurus 60	72	65	66	67	68	64	64	60	53	70
Taurus 70	84	77	74	71	69	68	66	63	58	73
Mars 90	84	77	74	71	69	68	66	63	58	73
Mars 100	84	77	74	71	69	68	66	63	58	73
Titan 130	85	78	75	72	70	69	67	64	59	74

Package average sound pressure levels (Re 20 µPa) for *SoLoNOx* and conventional gas turbine packages
For enclosed *Mars* and *Titan* gas turbine packages, acoustical lagging is required for the combustion air inlet ducting.

Table 11. Sound Pressure Levels – Unsilenced Combustion Inlet at 15 m (50 ft), Less than Full Load

Model	Octave Band Center Frequency, Hz									
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
Taurus 60	78	83	93	95	95	98	103	133	116	134
Taurus 70	81	86	96	98	98	101	106	139	122	140
Mars 90	81	87	94	94	95	98	101	135	126	136
Mars 100	81	87	94	94	95	98	101	135	126	136
Titan 130	82	88	95	95	96	99	102	137	129	138

Sound pressure levels (Re 20 µPa) for *Taurus*, *Mars* and *Titan* two-shaft gas turbine packages only

Table 12. Sound Pressure Levels – Unsilenced Combustion Exhaust at 15 m (50 ft), Less than Full Load

Model	Octave Band Center Frequency, Hz									
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
Taurus 60	89	91	88	91	98	93	82	73	64	97
Taurus 70	96	98	96	98	102	107	106	95	80	111
Mars 90	92	95	93	96	102	101	89	78	67	103
Mars 100	92	95	93	96	102	101	89	78	67	103
Titan 130	93	96	94	97	104	102	90	79	68	105

Sound pressure levels (Re 20 µPa) for *Taurus*, *Mars* and *Titan* two-shaft gas turbine packages only

Table 13. Sound Pressure Levels – Unenclosed Package at 1 m (3 ft), Less than Full Load

Model	Octave Band Center Frequency, Hz									
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
Taurus 60	98	92	95	100	109	98	92	90	95	107
Taurus 70	91	91	100	100	104	102	110	104	101	113
Mars 90	84	83	95	93	100	100	110	104	101	113
Mars 100	84	83	95	93	100	100	110	104	101	113
Titan 130	84	83	95	93	100	100	110	104	101	113

Sound pressure levels (Re 20 µPa) for *Taurus*, *Mars* and *Titan* two-shaft gas turbine packages only

Table 14. Sound Pressure Levels – Unenclosed Package at 15 m (50 ft), Less than Full Load

Model	Octave Band Center Frequency, Hz									
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
Taurus 60	85	79	82	87	96	85	79	77	82	94
Taurus 70	72	71	83	81	88	88	98	92	89	101
Mars 90	72	71	83	81	88	88	98	92	89	101
Mars 100	72	71	83	81	88	88	98	92	89	101
Titan 130	73	72	84	82	89	89	99	93	90	102

Sound pressure levels (Re 20 µPa) for *Taurus*, *Mars* and *Titan* two-shaft gas turbine packages only

Table 15. Sound Pressure Levels – Lube Oil Cooler at 15 m (50 ft)

Lube Oil Cooler, Sound Power Level	Octave Band Center Frequency, Hz									
	31.5	63	125	250	500	1000	2000	4000	8000	dBA
90 dBA (special)	63	70	67	60	55	52	48	44	39	58
100 dBA (standard)	73	80	77	70	65	62	58	54	49	68

Table 16. Combustion Air Inlet Air Filter Insertion Losses

Air Filter	Octave Band Center Frequency, Hz								
	31.5	63	125	250	500	1000	2000	4000	8000
Pulse Cleaning, Up-Draft, dBA	2	4	8	9	13	26	27	27	33
Pulse Cleaning, Cross-Flow, dBA	0	3	5	7	12	9	18	17	24
Barrier, dBA	0	2	3	4	4	5	8	3	18
Marine, dBA	0	1	2	1	2	5	6	9	8

Table 17. Inlet Silencer Insertion Losses for Oil & Gas Applications

Model	Octave Band Center Frequency, Hz								
	31.5	63	125	250	500	1000	2000	4000	8000
Saturn	0	1	2	3	15	25	48	55	37
Centaur & Taurus 60	1	2	3	4	17	32	46	47	31
Taurus 70	1	2	4	6	22	43	47	55	52
Mars	2	4	7	16	40	50	51	55	55
Titan	3	7	13	23	40	54	57	59	48

Table 18. Inlet Silencer Insertion Losses for Power Generation Applications

Model	Octave Band Center Frequency, Hz								
	31.5	63	125	250	500	1000	2000	4000	8000
Saturn	0	1	2	3	15	25	48	55	37
Centaur & Taurus 60	1	2	3	4	18	38	46	54	50
Taurus 70	1	3	7	11	20	40	55	53	41
Mars	3	6	15	24	35	55	55	55	45
Titan	0	1	5	8	27	48	55	61	60

Table 19. Exhaust Silencer Insertion Losses for Oil & Gas Applications

Model	Octave Band Center Frequency, Hz								
	31.5	63	125	250	500	1000	2000	4000	8000
Saturn	0	1	4	8	13	19	17	14	8
Centaur & Taurus 60	1	2	6	12	17	21	19	14	10
Centaur & Taurus 60 (floor standing)	2	4	9	19	26	29	23	20	13
Taurus 70	2	4	8	16	22	26	22	19	12
Taurus 70 (floor standing)	2	5	11	22	30	36	34	29	13
Mars (1.5 m (5 ft) long)	1	3	6	11	16	18	19	19	17
Mars (3 m (10 ft) long)	3	5	10	19	28	34	34	33	22
Titan	1	6	10	20	35	38	36	24	16

Table 20. Exhaust Silencer Insertion Losses for Power Generation Applications

Model	Octave Band Center Frequency, Hz								
	31.5	63	125	250	500	1000	2000	4000	8000
Saturn (floor standing)	3	5	11	19	22	28	26	17	14
Centaur & Taurus 60	1	2	6	12	17	21	19	14	10
Centaur & Taurus 60 (floor standing)	2	4	9	19	26	29	23	20	13
Taurus 70	2	5	10	16	21	26	26	24	17
Taurus 70 (floor standing)	2	5	11	22	30	36	34	29	13
Mars	1	3	6	11	16	18	19	19	17
Titan	1	6	10	20	35	38	36	24	16

Acceptable Work Environment Sound Levels

7.1 OCCUPATIONAL NOISE EXPOSURE STANDARDS

The workplace noise standards in most countries specify an 8-hour exposure limit of 85 dBA, although some have a 90 dBA limit. In these standards, the 8-hour exposure limit above which engineering or administrative controls must be applied is also 85 to 90 dBA. Some countries apply an 8-hour exposure limit of 80 dBA while utilizing an 85 dBA 8-hour exposure limit for the application of engineering and administrative controls. Discussed below are three workplace noise standards: The United States Occupational Noise Exposure Standard and the European Union's Council Directives 86/188/EEC and 2003/10/EC. Please refer to these for a complete explanation of the their requirements.

These standards apply to the sound levels in the workplace. They do not establish limits for the sound levels from machinery or other equipment. It is popularly assumed that a requirement for machinery to meet the OSHA or the EU standards means that workplace sound levels from the machinery will not exceed 85 dBA. However, the sound level in the workplace is a function of the workplace environment and the number and sound levels of the other machines in the workplace. Workplace sound levels equal to or greater than 85 dBA can be in compliance with these standards, if the provisions of the standards are met.

7.1.1 United States Occupational Noise Exposure Standard

The U.S. Department of Labor Occupational Safety and Health Administration (OSHA) Noise Exposure Standard of May 29, 1971 established a standard for noise exposure in the workplace. This standard was amended on March 8, 1983 with the addition of the Hearing Conservation Amendment, which described conditions and requirements for hearing conservation programs and lowered the 8-hour exposure action level from a sound level exceeding 90 dBA to a sound level equal to or greater than 85 dBA. This standard is known as the OSHA standard.

OSHA Standard

The OSHA standard requires protection against the effects of noise exposure when the sound levels exceed those shown in Table 21 (Table G-16 from the standard), when measured on the A scale of a sound level meter at slow response. The standard also allows determination of the A-weighted sound level from octave bands and it defines a procedure for this.

Table 21 defines permissible exposure levels. When these exposure levels are exceeded, the standard requires employers to implement feasible administrative or engineering controls. If these controls do not reduce the sound levels within the levels of Table 21, the employer must provide personal protective equipment to employees and employees must wear them to reduce the exposure levels to the levels in the table.

Table 21. Permissible Noise Exposures

Duration per Day, hours	Sound Level, dBA slow response
8	90
6	92
4	95
3	97
2	100
1½	102
1	105
½	110
¼ or less	115

When the daily noise exposure is composed of two or more periods of noise exposure of different levels, their combined effect should be considered, rather than the individual effect of each. If the sum of the following fractions: $C_1/T_1 + C_2/T_2 + \dots + C_n/T_n$ exceeds unity, then the mixed exposure should be considered to exceed the limit value. C_n indicates the total time of exposure permitted at that level and T_n indicates the total time of exposure permitted at that level.

Exposure to impulsive or impact noise should not exceed 140 dBA sound pressure level.

Hearing Conservation Program

The hearing conservation program is intended to minimize employee hearing loss from exposure to noise. The amendment to the OSHA standard requires implementation of a hearing conservation program when an employee's noise exposure equals or exceeds an 8-hour time-weighted